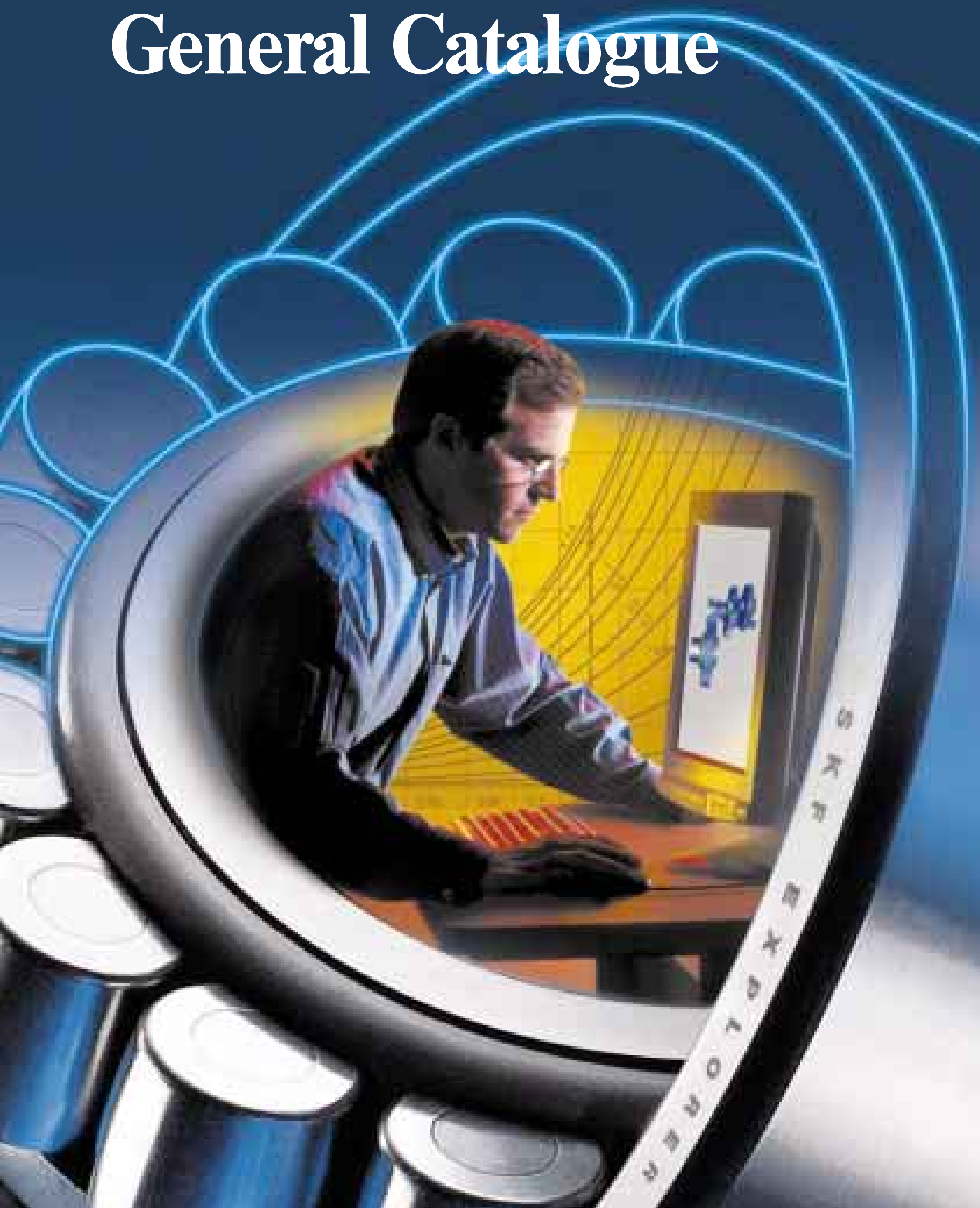
SKF

General Catalogue



Catalogue **5000 E** · June 2003

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Deep groove ball bearings

Angular contact ball bearings

Self-aligning ball bearings

Cylindrical roller bearings

Taper roller bearings

Spherical roller bearings

CARB® toroidal roller bearings

Thrust ball bearings

Cylindrical roller thrust bearings

Spherical roller thrust bearings

Engineering products

Mechatronics

Bearing accessories

Bearing housings

Maintenance and lubrication products

Other SKF products

General Catalogue

The SKF brand now stands for more than ever before, and means more to you as a valued customer.

While SKF maintains its leadership as the hallmark of quality bearings throughout the world, new dimensions in technical advances, product support and services have evolved SKF into a truly solutions-oriented supplier, creating greater value for customers.

These solutions encompass ways to bring greater productivity to customers, not only with breakthrough application-specific products, but also through leading-edge design simulation tools and consultancy services, plant asset efficiency maintenance programs, and the industry's most advanced supply management techniques.

The SKF brand still stands for the very best in rolling bearings, but it now stands for much more.

SKF – The knowledge engineering company

General

Quantity	Unit	Conversion			
Length	inch	1 mm	0,039 inch	1 in	25,40 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 mile	1 mile	1,609 km
Area	square inch	1 mm^2	0,00155 sq.in	1 sq.in	645,16 mm^2
	square foot	1 m^2	10,76 sq.ft	1 sq.ft	0,0929 m^2
Volume	cubic inch	1 cm^3	0,061 cub.in	1 cub.in	16,387 cm^3
	cubic foot	1 m^3	35 cub.ft	1 cub.ft	0,02832 m^3
	imperial gallon	1 l	0,22 gallon	1 gallon	4,5461 l
	U.S. gallon	1 l	0,2642 U.S. gallon	1 U.S. gallon	3,7854 l
Velocity, speed	foot per second	1 m/s	3,28 ft/s	1 ft/s	0,30480 m/s
	mile per hour	1 km/h	0,6214 mile/h (mph)	1 mile/h (mph)	1,609 km/h
Mass	ounce	1 g	0,03527 oz	1 oz	28,350 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/cub.in	1 lb/cub.in	27,680 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
Moment	inch pound-force	1 Nm	8,85 in.lbf	1 in.lbf	0,113 Nm
Power	foot-pound per second	1 W	0,7376 ft lbf/s	1 ft lbf/s	1,3558 W
	horsepower	1 kW	1,36 HP	1 HP	0,736 kW
Temperature	degree	Celcius	$t_C = 0{,}555\ (t_F - 32)$	Fahrenheit	$t_F = 1{,}8\ t_C + 32$

Foreword

The previous edition of the SKF General Catalogue was originally published in 1989. Since that time it has been translated into 16 languages and over 1 million copies have been distributed worldwide. In the last edition SKF introduced the “New Life Theory”, which since has become a major new technical standard for the bearing industry. With this broad usage and professional acceptance, the SKF General Catalogue is generally considered to be the authoritative reference source of its type throughout industry.

The General Catalogue was subsequently converted to electronic formats for added accessibility and convenience – available in a CD-ROM version, titled “SKF Interactive Engineering Catalogue”, and online at www.skf.com.

This new edition of the General Catalogue is also available electronically as well as in print and includes many revisions, additions and enhancements to make the catalogue even more useful and valuable. A familiarization with the information in this foreword, as well as other annotated sections, enables the user to obtain the maximum benefit from this comprehensive tool.

This foreword discusses the main sections of the catalogue with reference to major technical and product information as well as other available information of importance in understanding the overall capabilities of SKF.

General Catalogue overview

This new SKF General Catalogue contains almost all of the standard rolling bearings and accessories required by industrial equipment manufacturers, as well as those used for replacement purposes. To provide the highest levels of service, SKF endeavours to have its standard assortment match virtually all customer needs and to have products available on a worldwide basis.

The data in this catalogue relate to SKF’s state-of-the-art technology and production capabilities as of early 2003. The data may differ from that shown in earlier catalogues because of redesign, technological developments, or revised methods of calculation.

SKF reserves the right to make continuing improvements to SKF products with respect to materials, design and manufacturing methods, as well as changes necessitated by technological developments.

The units used in this catalogue are in accordance with ISO (International Organization for Standardization) standard 1000:1992, and SI (Système International d’Unités).

Technical section – principles of bearing selection and application

The technical section has a simplified index allowing for easy identification of the separate chapters, which cover the basics of bearing technology as required for the design of a bearing arrangement. These chapters are arranged in the order normally followed by a design engineer.

Significant innovations in the technical section

- a new model for the determination of friction in rolling bearings.
- revised speed ratings for the thermally permissible reference speeds based on the new friction model.
- a new model for the determination of the requisite lubricant viscosity based on more accurate knowledge of the influence of raceway surface roughness and of the elastic deformation of the lubricant film.
- a new method to determine lubricant service life as well as the optimum relubrication intervals for grease-lubricated bearings.
- the progress made in rolling bearing technology and the experience accumulated in practical applications in recent years have been taken into consideration and are referenced where appropriate in the various sections of the technical part.
- SKF technical services related to the bearing, the machine, or even the complete manufacturing plant – from bearing selection and calculation, to installation, monitoring, maintenance and replacement, are covered in a section referred to as Asset Efficiency Optimization.

Product section – bearing designations, descriptions and data

The product section has a thumb index for easy identification of bearing types. The product tables contain all of the engineering data required for the selection of a bearing and its application. Information relating to the specific types of bearings covered is arranged in front of the relevant product table(s).

Notable innovations in the product section

Significant products included for the first time are:

- CARB® toroidal roller bearings
- stainless steel deep groove ball bearings
- hybrid deep groove ball bearings
- ICOS™ bearing/seal units
- sealed spherical roller bearings
- high-temperature ball bearings and Y-bearing units
- INSOCOAT® bearings
- NoWear® bearings
- Solid Oil bearings

Investigations have shown that factors such as mounting, lubrication and maintenance have a much greater influence on bearing life than previously assumed. For this reason, new information has been included on the following:

- SKF products for maintenance and lubrication
- SKF condition monitoring equipment and systems

Several SKF innovations are worthy of particular note as they offer many advantages for enhanced performance or greater productivity.

For example, some of SKF product enhancements make it possible to build smaller machines having the same or even better performance. Smaller size also implies lighter weight; meaning less friction, lower operating temperatures, reduced lubricant requirements and power consumption, and as a result, improved economy and added value.

To facilitate easy reference to the various product innovations among the volume of information contained in this catalogue, the specific products are identified as follows:

- SKF Explorer bearings – the new performance class from SKF,
- application optimized bearings – standard bearings tailored for specialized requirements,
- mechatronic components – combinations of bearings and electronic sensors.

These innovations represent the most important new products in this catalogue and will be explained in more detail under their appropriate headings in the product section. For convenience, a summary description of these products is included hereafter.

SKF Explorer bearings
– the new performance class from SKF
SKF Explorer is a new performance class of rolling bearings, of the types angular contact ball bearings, cylindrical roller bearings, spherical roller bearings, CARB toroidal roller bearings and spherical roller thrust bearings, which provide a substantial improvement in key operational parameters relevant to the bearing type and its typical applications. This new level of superior performance represents the blending of SKF's applications knowledge with its expertize in tribology, materials development, design optimization and manufacturing.

Using advanced analytical and modelling techniques and supporting testing, SKF engineers were able to confirm that SKF Explorer bearings provide a significant improvement in key operational parameters. These parameters, according to bearing type and application, include noise, vibration, service life, dimensional stability, dynamic load carrying ability and heat generation (friction torque). Because these parameters are not adequately factored into standardized life calculations, SKF Explorer bearing life is calculated with modified factors, which takes key operational parameters into account.

SKF Explorer bearings are interchangeable with previously specified SKF standard bearings of the same type and size. These bearings are included in the relevant product tables and are easily identified by an asterisk (*) in front of the bearing designation.

The making of an SKF Explorer bearing

Achieving the outstanding levels of SKF Explorer bearings has only been possible due to the basic sound engineering design of SKF products and by further improving the manufacturing of bearings to these designs. By studying the interrelationship between each bearing component, SKF engineers were able to maximize the effects of lubrication and minimize the effects of friction, wear and contamination. To do this, an international research team looked at each component at micro level and then developed new procedures to consistently manufacture this new standard of excellence.

SKF Explorer bearings are characterized by a number of technical improvements some of which are listed below. Depending on the type of the SKF Explorer bearing one or several of these below given examples have been applied:

- Improved bearing steel
  SKF Explorer bearings feature an extremely clean and homogenous steel with a minimum number of inclusions. This improved steel is so much cleaner than the highest grades covered by present classification methods that SKF has developed new calculation methods to take this factor into account.

- The unique SKF heat treatment procedures
  To maximize the benefits of SKF's ultra-clean steel, engineers incorporated unique heat treatment procedures. These new procedures optimize the bearing's resistance to operational damage without affecting heat stabilization. Wear resistance was improved so dramatically that SKF engineers were not able to accurately predict life expectancy using existing life factors for calculation methods.

- Improved surface finish
  The surface finish on all contact surfaces (rolling elements and raceways) has been improved to maximize the effects of the lubricant and reduce vibrations and noise. This has lead to a smoother, cooler running bearing that uses less lubricant and consequently the arrangement, including the seals, requires less maintenance.

Deep groove ball bearings and taper roller bearings

For the rolling bearing types deep groove ball bearings and taper roller bearings there have been many performance improvements since the last SKF General Catalogue. In line with the SKF product strategy, improvements for deep groove ball bearings and taper roller bearings have now sufficiently been implemented for certain sizes to qualify as SKF Explorer class bearings. For these selected deep groove ball bearings sizes improved sealing, precision and surface finish, give reduced noise and vibration levels and improved running accuracy. Similarly for selected taper roller bearing sizes, improved surfaces for better lubrication and significantly reduced noise and vibration levels, cleaner steel in combination with improved heat treatment give significantly longer life. Because all these parameters are not adequately factored into standardized life calculations, the bearing life of selected sizes SKF deep groove ball bearings and taper roller bearings is calculated with modified factors in line with all SKF Explorer class bearings.

Application optimized bearings – tailored bearings
These bearings have standardized dimensions but incorporate special features for specific applications. Properly applied, these bearings make costly customized bearings unnecessary, and they can also greatly reduce lead times since they are generally available from stock. This group of SKF bearings includes the following:

- Hybrid deep groove ball bearings with ceramic balls and rings of bearing steel. These bearings have good emergency running properties and can cope with extreme conditions and high speeds. Their inherent resistance to the passage of electric current means that they are very suitable for electric motors and electrically powered tools.
- INSOCOAT bearings have an insulating coating of aluminium oxide on the external surfaces of the inner or outer ring. These bearings can be used in difficult electrical applications with no additional design requirements and they can also be a drop-in replacement for conventional bearings in existing applications.
- Bearings and bearing units for extreme temperatures. Their operating temperature range covers –150 to +350 °C, making these products ideal for kiln trucks, roller hearth furnaces, bakery plants and refrigeration rooms.
- NoWear bearings. These bearings have been surface treated to withstand arduous operating conditions such as smearing, zero load or boundary lubrication conditions.
- Solid Oil bearings for applications where conventional grease or oil lubrication methods are not adequate or practical.

Mechatronic components – bearings and sensors in combination
SKF "plug and play" mechatronic bearing units can be used to monitor or control operating sequences, motion or steering systems. Information in this catalogue provides a brief overview of the mechatronic components and developments engineered by SKF, which have already been well proven in a variety of industrial and automotive applications. More in-depth information on SKF mechatronics products and capabilities can be obtained through your SKF representative.

Detailed information on Sensor-Bearing Units, which are part of the SKF standard line, will be found together with the appropriate product data.

Other SKF products
In this section, all rolling bearings, plain bearings, linear bearings, and seals etc., not listed in the product section of the catalogue are listed with a brief description. Where further information is available, reference is made to appropriate SKF printed and/or electronic media.

SKF system solutions
SKF has applied its extensive knowledge of particular industrial applications and their demanding requirements, and developed system solutions that yield cost-effective results.

Some of these solutions do not even incorporate bearings. This underscores SKF's continuing efforts to expand its offerings beyond traditional bearing applications to other technologies from the fields of mechatronics and electronics. Some of the more important system solutions currently available are listed here:

- Copperhead system solution for vibrating screens
- system solution for continuous casting plants
- system solution for paper machines
- system solution for printing machines

- system solution for automotive transmissions
- system solution for railway vehicles
- system solution for wind power plants

Other SKF catalogues
Even though this General Catalogue contains more than 1 100 pages of core products and related information, it is by no means all-inclusive of the total SKF product offering. Detailed information on many of the other SKF products not covered in this General Catalogue is also available in separate, individual printed catalogues which include:

- Needle roller bearings
- High-precision bearings
- Y-bearings and Y-bearing units
- Spherical plain bearings and rod ends
- Bearing accessories
- Bearing housings
- CR seals

A brief description of these products can be found in the General Catalogue under the heading “Other SKF products” or online at www.skf.com.

Information on the comprehensive assortment of SKF linear bearing products, ball and roller screws, and linear actuators will be found in the separate SKF catalogue “Linear Motion Product Range”, available from your SKF Linear Motion representative.

The SKF Interactive Engineering Catalogue
SKF provides this catalogue in electronic formats available on CD-ROM or online at www.skf.com. The SKF Interactive Engineering Catalogue contains comprehensive technical information on the following products:

- SKF rolling bearings inclusive accessories
- SKF bearing units
- SKF bearing housings
- SKF plain bearings
- CR seals

The electronic catalogue formats allow for easy navigation and also provide calculations for critical design factors such as:

- basic and modified rating lives (L_{10} and L_{nm})
- requisite lubricant viscosity
- equivalent bearing load
- minimum bearing load
- dynamic axial load carrying capacity of cylindrical roller bearings
- friction
- bearing frequencies

In addition, 2 or 3-dimensional drawings can be supplied in some 50 CAD formats via the SKF internet site.

SKF – the supplier of choice
The SKF General Catalogue – as comprehensive as it is – is just one of the many advantages provided to our customers. There are many other capabilities that contribute to the overall value customers receive in making SKF their supplier of choice:

- simplified bearing selection,
- short delivery times,
- worldwide availability,
- commitment to product innovation,
- state-of-the-art application solutions,
- extensive engineering and technology knowledge in virtually every industry.

SKF – The knowledge engineering company

The business of the SKF Group consists of the design, manufacture and marketing of the world's leading brand of rolling bearings, with a global leadership position in complementary products such as radial seals. SKF also holds an increasingly important position in the market for linear motion products, high precision aerospace bearings, machine tool spindles, plant maintenance services and is an established producer of high-quality bearing steel.

The SKF Group maintains specialized business operations to meet the needs of the global marketplace. SKF supports specific market segments with ongoing research and development efforts that have led to a growing number of innovations, new standards and new products.

The Group has global ISO 14001 environmental certification. Individual divisions have been approved for quality certification in accordance with either ISO 9000 or appropriate industry specific standards.

Some 80 manufacturing sites worldwide and sales companies in 70 countries make SKF a truly international corporation. In addition, our 7 000 distributor and dealer partners around the world, e-business marketplace and global distribution system put SKF close to customers for the supply of both products and services. In essence, SKF solutions are available wherever and whenever our customers need them.

Overall, the SKF brand now stands for more than ever before. It stands for the knowledge engineering company ready to serve you with world-class product competences, intellectual resources and the vision to help you succeed.

Evolving by-wire technology

SKF has unique expertize and knowledge in fast-growing by-wire technology, from fly-by-wire, to drive-by-wire, to work-by-wire. SKF pioneered practical fly-by-wire technology and is a close working partner with all aerospace industry leaders. As an example, virtually all aircraft of the Airbus design use SKF by-wire systems for cockpit flight control.

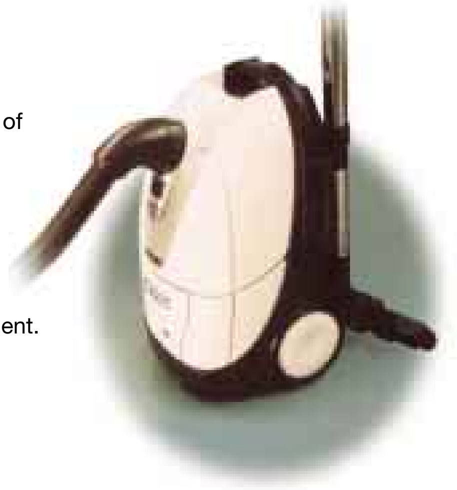
SKF is also a leader in automotive drive-by-wire, having jointly developed the revolutionary Filo and Novanta concept cars which employ SKF mechatronics for steering and braking. Further by-wire development has led SKF to produce an all-electric forklift truck which uses mechatronics rather than hydraulics for all controls.

Delivering asset efficiency optimization
To optimize efficiency and boost productivity, many industrial facilities outsource some or all of their maintenance services to SKF, often with guaranteed performance contracts. Through the specialized capabilities and knowledge available from SKF Reliability Systems, SKF provides a comprehensive range of asset efficiency services, from maintenance strategies and engineering assistance, to operator-driven reliability and machine maintenance programs.

Planning for sustainable growth
By their very nature, bearings make a positive contribution to the natural environment. Reduced friction enables machinery to operate more efficiently, consume less power and require less lubrication. SKF is continually raising the performance bar, enabling a new generation of high-efficiency products and equipment. With an eye to the future, SKF's global policies and manufacturing techniques are planned and implemented to help protect and preserve the earth's limited natural resources. We remain committed to sustainable, environmentally responsible growth.

Maintaining a 320 km/h R&D lab
In addition to SKF's renowned research and development facilities in Europe and the United States, Formula One car racing provides a unique environment for SKF to push the limits of bearing technology. For over 50 years, SKF products, engineering and knowledge have helped make Scuderia Ferrari a formidable force in F1 racing. (The average racing Ferrari utilizes more than 150 SKF components.) Lessons learned here are applied to the products we provide to automakers and the aftermarket worldwide.

Developing a cleaner cleaner
The electric motor and its bearings are the heart of many household appliances. SKF works closely with appliance manufacturers to improve their product's performance, cut costs, reduce weight, etc. A recent example produced a new generation of vacuum cleaners with substantially more suction. SKF's knowledge in small bearing technology is also applied to manufacturers of power tools and office equipment.



Creating a new "cold remedy"
In the frigid winters of northern China, sub-zero temperatures can cause rail car wheel assemblies and their bearings to seize due to lubrication starvation. SKF created a new family of synthetic lubricants formulated to retain their lubrication viscosity even at these extreme bearing temperatures. SKF's knowledge of lubricants and friction are unmatched in the world.

Harnessing wind power
The growing industry of wind-generated electric power provides an environmentally compatible source of electricity. SKF is working closely with global industry leaders to develop efficient and trouble-free turbines, using SKF knowledge to provide highly specialized bearings and condition monitoring systems to extend equipment life in the extreme and often remote environments of wind farms.

Contents

Principles of bearing selection and application

A bearing arrangement consists of more than just the bearings. Associated items such as the shaft and housing are integral parts of the overall assembly arrangement. The importance of lubricant and sealing elements cannot be overestimated. Deploying a full bearing's performance relies on the presence of appropriate lubrication and adequate protection from corrosion and the ingress into the assembly of foreign matter. Cleanliness has a profound effect on bearing service life – which is why lubricants and seals have become part of the SKF business.

To design a rolling bearing arrangement it is necessary

- to select a suitable bearing type and
- to determine a suitable bearing size,

but that is not all. Several other aspects have to be considered:

- a suitable form and design of other components of the arrangement,
- appropriate fits and bearing internal clearance or preload,
- holding devices,
- adequate seals,
- the type and quantity of lubricant,
- installation and removal methods, etc.

Each individual decision affects the performance, reliability and economy of the bearing arrangement.

The amount of work entailed depends on whether experience is already available about similar arrangements. When experience is lacking, when extraordinary demands are made or, when the costs of the bearing arrangement and any subsequent outline have to be given special consideration, then much more work is needed including, for example, more accurate calculations and/or testing.

As the leading bearing supplier SKF manufactures a large number of bearing types, series, designs, variants and sizes. The most common of them are introduced in the section "Product index", starting on **page 1114**. There are also bearings which are not included in this catalogue. Information about most of these bearings will be found in special catalogues or in the "SKF Interactive Engineering Catalogue" on CD-ROM or online at www.skf.com.

In the following sections of this general technical introduction, the designer of a bearing arrangement will find the necessary basic information presented in the order in which it is generally required. Obviously it is impossible to include all the information needed to cover every conceivable bearing application. For this reason, in many places, reference is made to the comprehensive SKF application engineering service, which includes technical support regarding the selection of the right bearing as well as calculations of the complete bearing arrangement. The higher the technical demands placed on a bearing arrangement and the more limited the available experience of using bearings for particular applications, the more advisable it is to make use of this service.

The information contained in the general technical section generally applies to rolling bearings, or at least to a group of bearings. Special information specific to one bearing type only will be found in the text preceding the appropriate individual product section. Additional special catalogues and brochures covering specific application areas are available on request. Detailed information on almost all SKF rolling bearings, bearing units, bearing housings, plain bearings, seals etc. can also be found in the "SKF Interactive Engineering Catalogue" on CD-ROM or online at www.skf.com.

It should be noted that the values given in the product tables for load and speed ratings as well as for the fatigue load limit are heavily rounded.

Bearing terminology

To better understand frequently used bearing terms, definitions are provided on **pages 20** and **21** and explained with help of drawings. A detailed collection of bearing specific terms and definitions is to be found in ISO 5593:1997:Rolling bearings – Vocabulary.

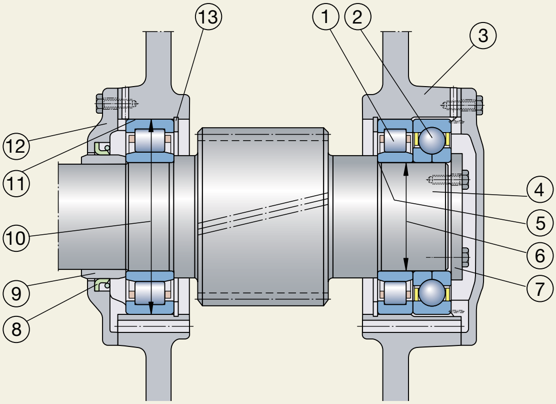
Product index
The product range shown in this General Catalogue comprises approximately 10 000 bearings, bearing accessories and bearing housings. So that users can find the technical data for a product known only by its designation, e.g. 6208-2RS1, the series designations are listed in the index starting on **page 1114**. In this case 62-2RS1 is used. Designations in this index are in alphanumerical order. The page listing for each designation is the start of the introductory text for that particular product type.

Bearing terminology

Bearing arrangement (➔ fig 1)

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

Fig 1



Radial bearings (➔ figs 2 and 3)

1 Inner ring
2 Outer ring
3 Rolling element: ball, cylindrical roller, needle roller, tapered roller, spherical roller
4 Cage
5 Capping device
Seal – made of elastomer, contacting (shown in figure) or non-contacting
Shield – made of sheet steel, non-contacting
6 Outer ring outside diameter
7 Inner ring bore
8 Inner ring shoulder diameter
9 Outer ring shoulder diameter
10 Snap ring groove
11 Snap ring
12 Outer ring side face
13 Seal anchorage groove

Fig 2

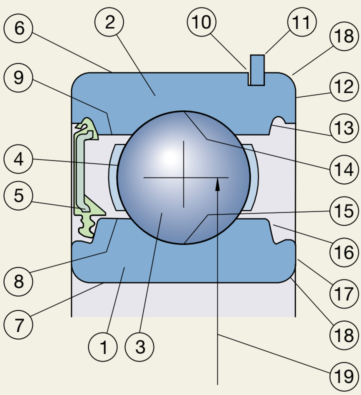
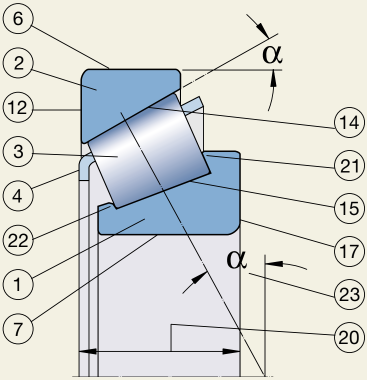
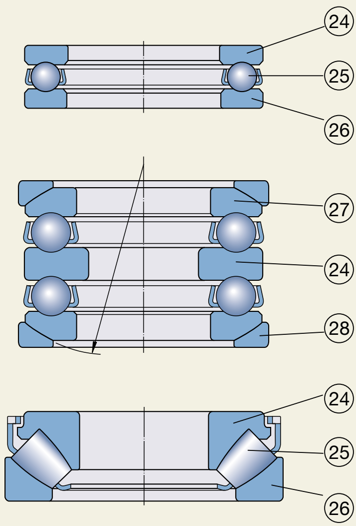


Fig 3



14 Outer ring raceway
15 Inner ring raceway
16 Sealing groove
17 Inner ring side face
18 Chamfer
19 Bearing mean diameter
20 Total bearing width
21 Guiding flange
22 Retaining flange
23 Contact angle

Fig 4



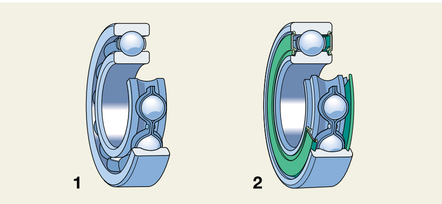
Thrust bearings (→ **fig** 4)

24 Shaft washer
25 Rolling element and cage assembly
26 Housing washer
27 Housing washer with sphered seating surface
28 Seating support washer


SKF

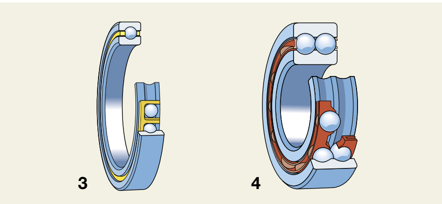
Bearing types

Radial bearings



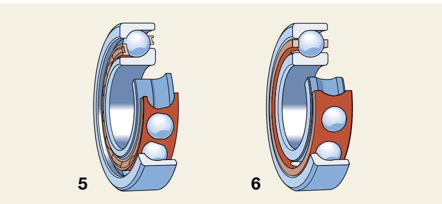
Deep groove ball bearings

single row, with or without filling slots
- open basic design (**1**)
- with shields
- with contact seals (**2**)
- with snap ring groove, with or without snap ring



single row with fixed section
- open basic design (**3**)
- with contact seals

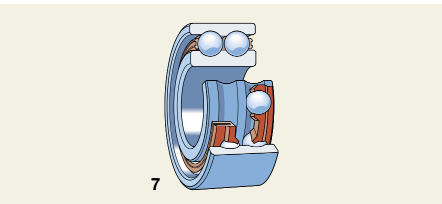
double row (**4**)



Angular contact ball bearings

single row
- design for universal matching (**5**)
- basic design for single mounting

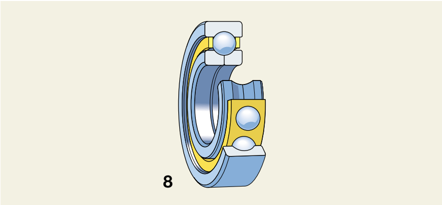
single row high precision[1)]
- standard design for single mounting (**6**)
- design for universal matching
- matched bearing sets



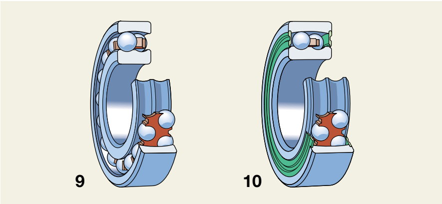
double row
- with one-piece inner ring (**7**)
  - open basic design
  - with shields
  - with contact seals
- with two-piece inner ring

Footnote → **page 31**

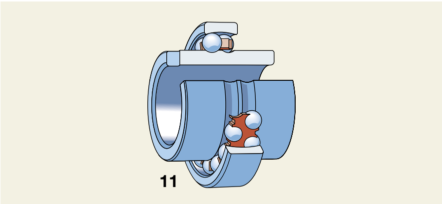
Radial bearings



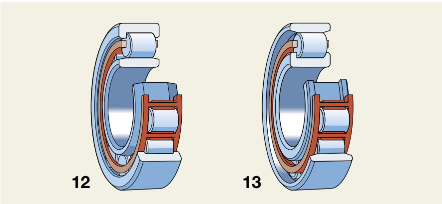
Four-point contact ball bearings (**8**)



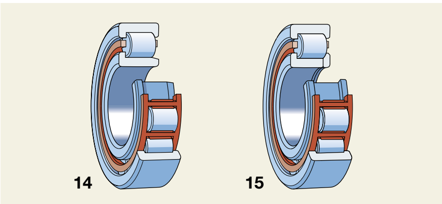
Self-aligning ball bearings
with cylindrical or tapered bore
open basic design (**9**)
with contact seals (**10**)



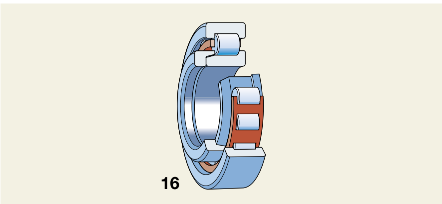
with extended inner ring (**11**)



Cylindrical roller bearings
single row
NU type (**12**)
N type (**13**)

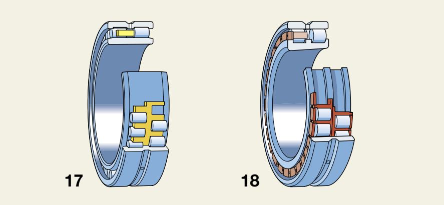


NJ type (**14**)
NUP type (**15**)

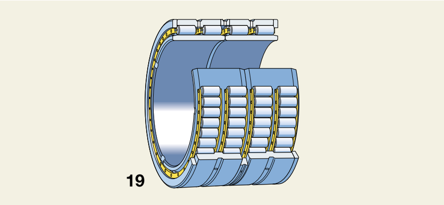


angle ring (**16**)
for NU and NJ type bearings

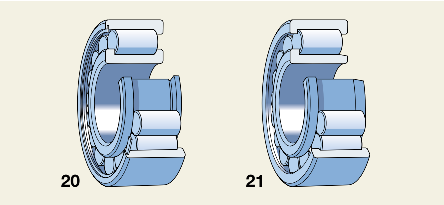
Radial bearings



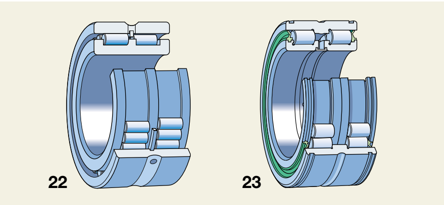
Cylindrical roller bearings
double row[1)]
- with cylindrical or tapered bore
  - NNU type (**17**)
  - NN type (**18**)
  - NNUP type



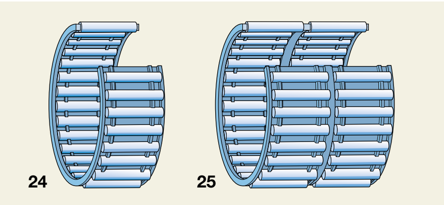
four-row[2)]
- with cylindrical or tapered bore
  - open design (**19**)
  - with contact seals



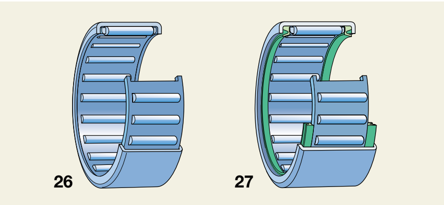
Full complement cylindrical roller bearings
single row
- NCF type (**20**)
- NJG type (**21**)



double row
- with integral flanges on the inner ring (**22**)
- with integral flanges on the inner and outer rings
- with contact seals (**23**)



Needle roller and cage assemblies[3)]
single row (**24**)
double row (**25**)



Drawn cup needle roller bearings, open ends[3)]
single and double row
- open basic design (**26**)
- with contact seals (**27**)

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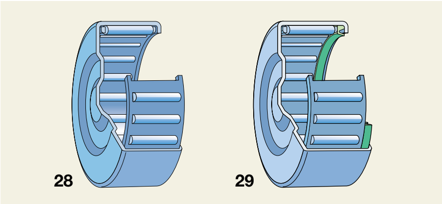
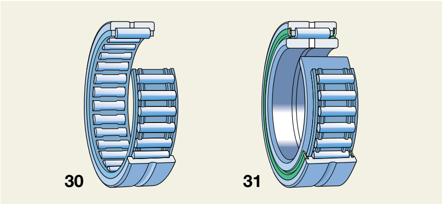
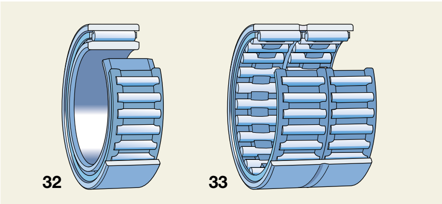
Radial bearings



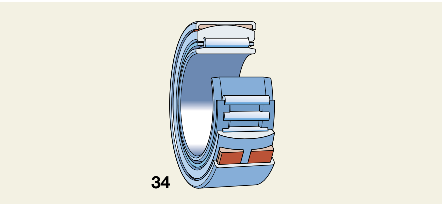
Drawn cup needle roller bearings, closed end[3)]
single and double row
open basic design (**28**)
with contact seal (**29**)



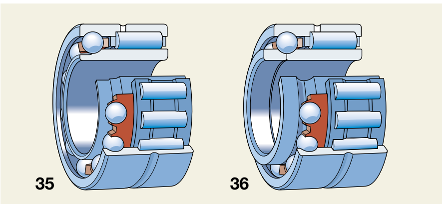
Needle roller bearings with flanges[3)]
single and double row
without inner ring (**30**)
with inner ring
open basic design
with contact seals (**31**)



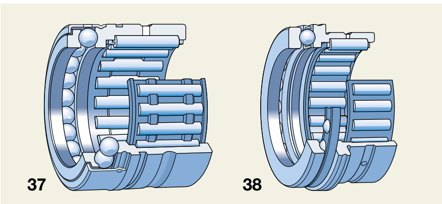
Needle roller bearings without flanges[3)]
single and double row
with inner ring (**32**)
without inner ring (**33**)



Alignment needle roller bearings[3)]
without inner ring
with inner ring (**34**)



Combined needle roller bearings[3)]
Needle roller/angular contact ball bearings
single direction (**35**)
double direction (**36**)



Needle roller/thrust ball bearings
with full complement thrust ball bearing (**37**)
with cage-guided ball set
with or without (**38**) cover

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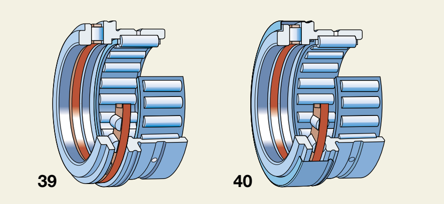
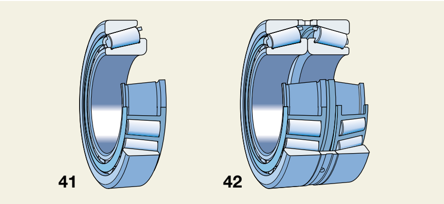
Radial bearings

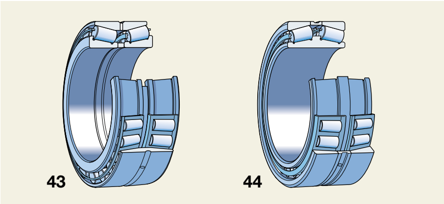


Needle roller/cylindrical roller thrust bearings
- without cover (**39**)
- with cover (**40**)

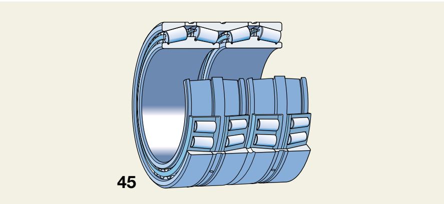


Taper roller bearings

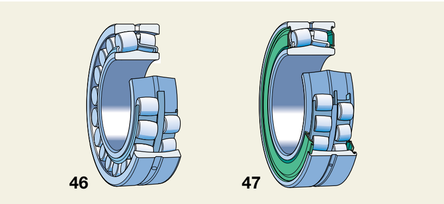
single row
- single bearings (**41**)
- matched bearing sets
  - face-to-face (**42**)
  - back-to-back
  - in tandem



double row[2)]
- TDO configuration (back-to-back) (**43**)
- TDI configuration (face-to-face) (**44**)

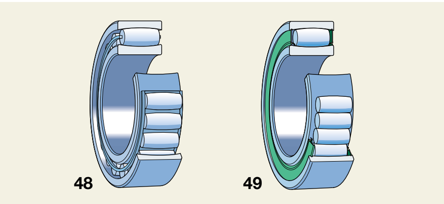


four-row[2)]
- TQO configuration (**45**)
- TQI configuration



Spherical roller bearings

with cylindrical or tapered bore
- open basic designs (**46**)
- with contact seals (**47**)



CARB toroidal roller bearings

with cylindrical or tapered bore
- open basic designs
  - with cage-guided roller set (**48**)
- with full complement roller set
  - with contact seals (**49**)

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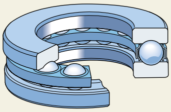
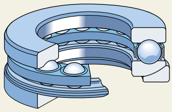
Thrust bearings



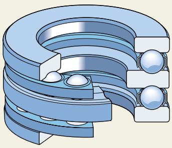
50



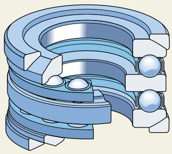
51

Thrust ball bearings

single direction
- with flat housing washer (**50**)
- with sphered housing washer
  - with (**51**) or without seating washer

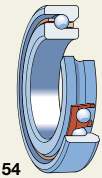


52

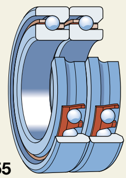


53

double direction
- with flat housing washers (**52**)
- with sphered housing washers
  - with (**53**) or without seating washers



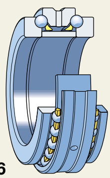
54



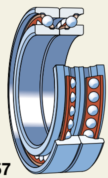
55

Angular contact thrust ball bearings[1]

high precision bearings
- single direction
  - standard design for single mounting (**54**)
  - design for universal matching
  - matched bearing sets (**55**)

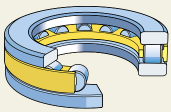


56

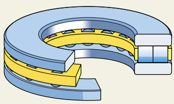


57

- double direction
  - standard design (**56**)
  - high-speed design (**57**)



58

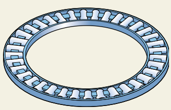


59

Cylindrical roller thrust bearings

single direction
- single row (**58**)
- double row (**59**)

components
- cylindrical roller and cage thrust assemblies
- shaft and housing washers



60

Needle roller thrust bearings[3]

single direction (**60**)
- needle roller and cage thrust assemblies
- raceway washers
- thrust washers

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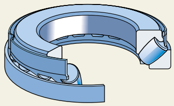
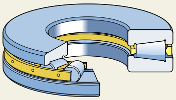
SKF

Thrust bearings

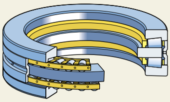


61

Spherical roller thrust bearings
single direction (**61**)



62

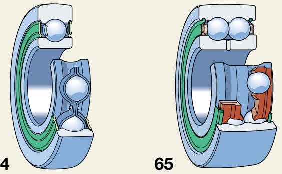


63

Taper roller thrust bearings[2)]
single direction
  with or without (**62**) cover
  screw down bearings
double direction (**63**)

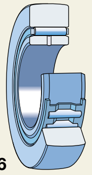
Footnote ➔ **page 31**

Track runner bearings



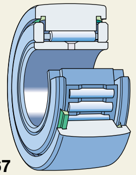
64 65

Cam rollers
single row ball bearing cam roller (**64**)
double row ball bearing cam roller (**65**)



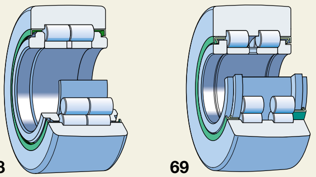
66

Support rollers[3)]
without axial guidance
with or without contact seals
without inner ring
with inner ring (**66**)



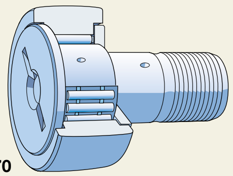
67

with axial guidance by thrust washers
with or without contact seals
with cage-guided needle roller set (**67**)
full complement



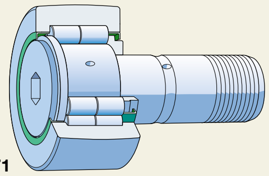
68 69

with axial guidance by cylindrical rollers
with labyrinth seals (**68**)
with contact seals (**69**)
with lamellar seals



70

Cam followers[3)]
with axial guidance by thrust plate
with or without contact seals
with concentric seating (**70**)
with eccentric seating collar
with cage-guided needle roller set
with full complement needle roller set



71

with axial guidance by cylindrical rollers
with labyrinth seals (**71**)
with contact seals
with concentric seating
with eccentric seating collar

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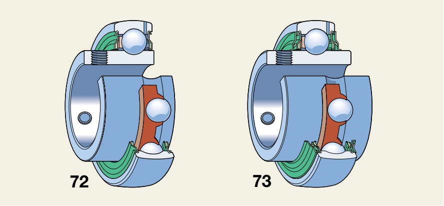
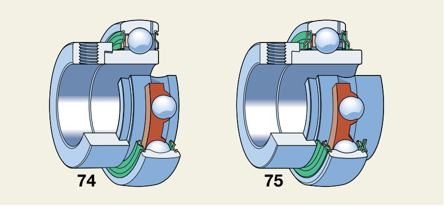
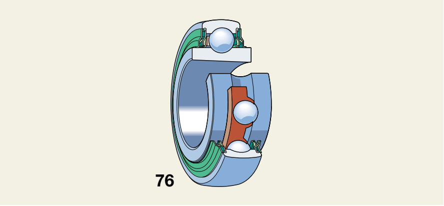
Y-bearings



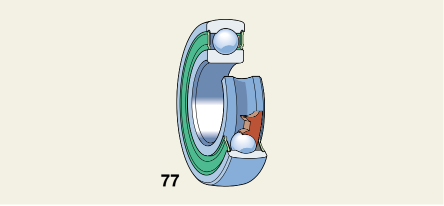
Y-bearings (insert bearings)[4)]
with grub screw locking
inner ring extended on one side (**72**)
inner ring extended on both sides (**73**)



with eccentric locking collar
inner ring extended on one side (**74**)
inner ring extended on both sides (**75**)



with tapered bore
inner ring extended on both sides (**76**)
for adapter sleeve mounting



with standard inner ring
for locating by interference fit on the shaft (**77**)

1) See SKF catalogue “High-precision bearings” or “SKF Interactive Engineering Catalogue”
2) See “SKF Interactive Engineering Catalogue”
3) See SKF catalogue “Needle roller bearings” or “SKF Interactive Engineering Catalogue”
4) See SKF catalogue “Y-bearing units” or “SKF Interactive Engineering Catalogue”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Selection of bearing type

Each bearing type displays characteristic properties, based on its design, which makes it more, or less, appropriate for a given application. For example, deep groove ball bearings can accommodate moderate radial loads as well as axial loads. They have low friction and can be produced with high precision and in quiet running variants. Therefore they are preferred for small and medium-sized electric motors.

Spherical and toroidal roller bearings can carry very heavy loads and are self-aligning. These properties make them popular for example for heavy engineering applications, where there are heavy loads, shaft deflections and misalignments.

In many cases, however, several factors have to be considered and weighed against each other when selecting a bearing type, so that no general rules can be given.

The information provided here should serve to indicate which are the most important factors to be considered when selecting a standard bearing type and thus facilitate an appropriate choice:

- available space
- loads
- misalignment
- precision
- speed
- quiet running
- stiffness
- axial displacement
- mounting and dismounting
- integral seals

A comprehensive overview of the standard bearing types, their design characteristics and their suitability for the demands placed on a given application will be found in the matrix on **pages 46** and **47**. Detailed information on the individual bearing types, including their characteristics and the available designs, will be found in the sections dealing with individual bearing types. Bearing types that are not included in the matrix are generally only used for a few well-defined applications.

The matrix permits only a relatively superficial classification of bearing types. The limited number of symbols does not allow an exact differentiation and some properties do not depend solely on bearing design. For example, the stiffness of an arrangement incorporating angular contact ball bearings or taper roller bearings also depends on the applied preload and the operating speed which is influenced by the precision of the bearing and its associated components as well as by the cage design. In spite of its limitations, the matrix on **pages 46** and **47** should enable an appropriate choice of bearing type to be made. It should also be considered that the total cost of a bearing arrangement and inventory considerations could also influence the final choice.

Other important criteria to be observed when designing a bearing arrangement including load carrying capacity and life, friction, permissible speeds, bearing internal clearance or preload, lubrication and sealing are dealt with in depth in separate sections of this catalogue. The complete SKF product range is not shown in this General Catalogue. Specific catalogues and brochures are available for bearings not covered here – please consult SKF.

Available space

In many cases, one of the principal dimensions of a bearing – the bore diameter – is predetermined by the machine's design and the shaft diameter.

For small-diameter shafts all types of ball bearings can be used, the most popular being deep groove ball bearings; needle roller bearings are also suitable (➔ **fig** 1). For large-diameter shafts, cylindrical, taper, spherical and toroidal roller bearings are available, as well as deep groove ball bearings (➔ **fig** 2).

When radial space is limited, bearings with a small cross section, particularly those with a low cross-sectional height, should be chosen, i.e. bearings in the 8 or 9 Diameter Series. Needle roller and cage assemblies, drawn cup needle roller bearings and needle roller bearings without or even with inner ring (➔ **fig** 3) are very appropriate as well as certain series of deep groove and angular contact ball bearings, cylindrical, taper, spherical and toroidal bearings.

Fig 1

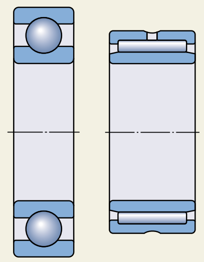


Fig 2

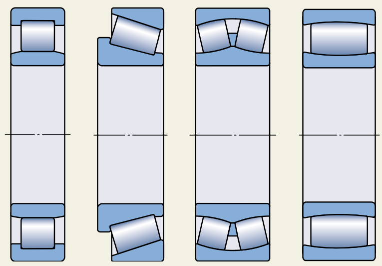
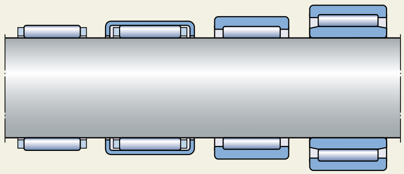


Fig 3



When axial space is limited, certain series of cylindrical roller bearings and deep groove ball bearings can be used for radial or combined loads respectively (➔ **fig** 4) as well as the various types of combined needle roller bearings (➔ **fig** 5). For purely axial loads, needle roller and cage thrust assemblies (with or without washers) as well as thrust ball bearings and cylindrical roller thrust bearings can be used (➔ **fig** 6).

Fig 4

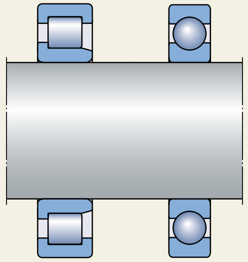


Fig 5

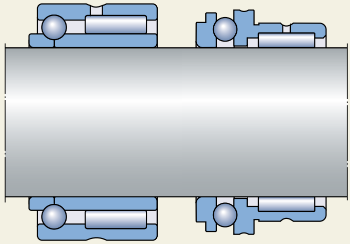


Fig 6

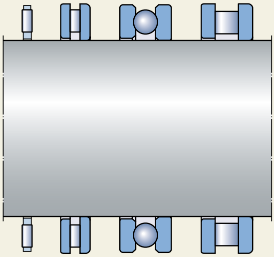


Fig 7

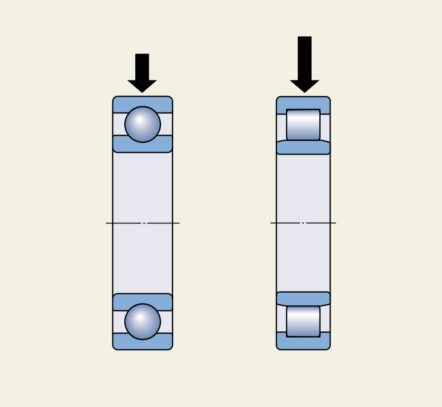


Fig 8

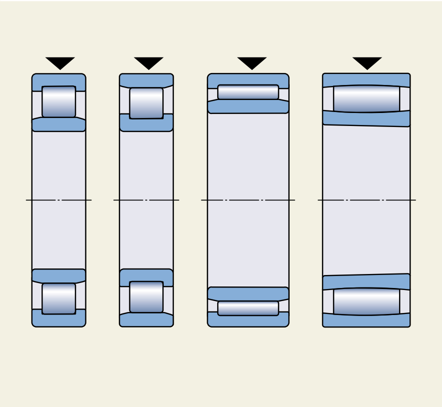
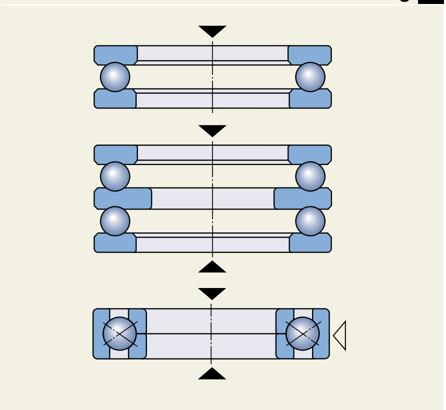


Fig 9



Loads

Magnitude of load

The magnitude of the load is one of the factors that usually determines the size of the bearing to be used. Generally, roller bearings are able to support heavier loads than similar sized ball bearings (➔ **fig** 7) and bearings having a full complement of rolling elements can accommodate heavier loads than the corresponding caged bearings. Ball bearings are mostly used where loads are light or moderate. For heavy loads and where shaft diameters are large, roller bearings are usually the more appropriate choice.

Direction of load

Radial load

NU and N types cylindrical roller bearings, needle roller bearings and toroidal roller bearings can only support pure radial loads (➔ **fig** 8). All other radial bearings can accommodate some axial loads in addition to radial loads (➔ "Combined loads").

Axial load

Thrust ball bearings and four-point contact ball bearings (➔ **fig** 9) are suitable for light or moderate loads that are purely axial. Single direction thrust ball bearings can only accommodate axial loads acting in one direction; for axial loads acting in both directions, double direction thrust ball bearings are needed.

Fig 10

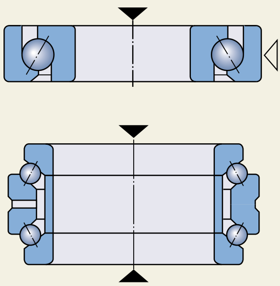
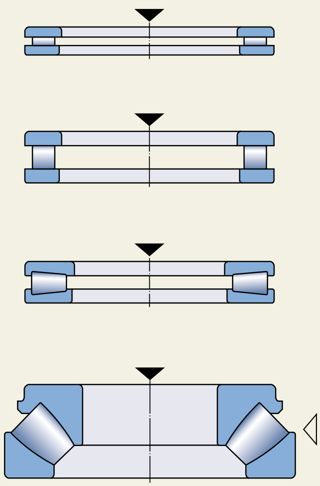


Fig 11



Angular contact thrust ball bearings can support moderate axial loads at high speeds; here the single direction bearings can also accommodate simultaneously acting radial loads, while double direction bearings are normally used only for purely axial loads (➔ **fig** 10).

For moderate and heavy axial loads acting in one direction, needle roller thrust bearings, cylindrical and taper roller thrust bearings are suitable, as are spherical roller thrust bearings (➔ **fig** 11). Spherical roller thrust bearings can also accommodate simultaneously acting radial loads. For heavy alternating axial loads, two cylindrical roller thrust bearings or two spherical roller thrust bearings can be mounted adjacent to each other.

Combined load

A combined load comprises a radial and an axial load acting simultaneously. The ability of a bearing to carry an axial load is determined by the angle of contact α – the greater the angle, the more suitable the bearing for axial loads. An indication of this is given by the calculation factor Y, which becomes smaller as the contact α increases. The values of this factor for a bearing type or for individual bearings will be found in the introductory text of the product table sections, or in the actual product tables. The axial load carrying capacity of a deep groove ball bearing depends on its internal design and the internal clearance in the bearing (➔ section "Deep groove ball bearings", starting on **page 287**).

For combined loads, single and double row angular contact ball bearings and single row taper roller bearings are most commonly used although deep groove ball bearings and spherical roller bearings are suitable (➔ **fig** 12). In addition, self-aligning ball bearings and NJ and NUP type cylindrical roller bearings as well as NJ and NU type cylindrical roller bearings with HJ angle rings can be used for combined loads where the axial component is relatively small (➔ **fig** 13).

Single row angular contact ball bearings and taper roller bearings, NJ and NU+HJ type cylindrical roller bearings and spherical roller thrust bearings can only accommodate axial loads acting in one direction. For axial

Fig 12

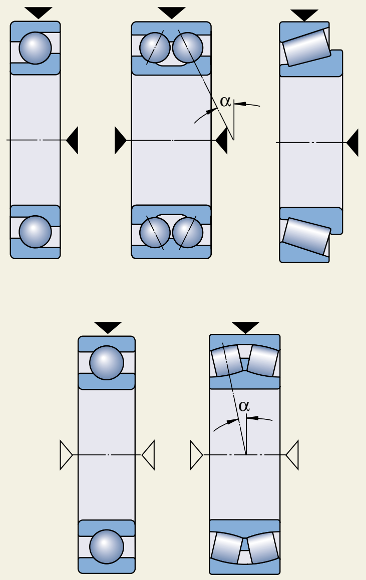
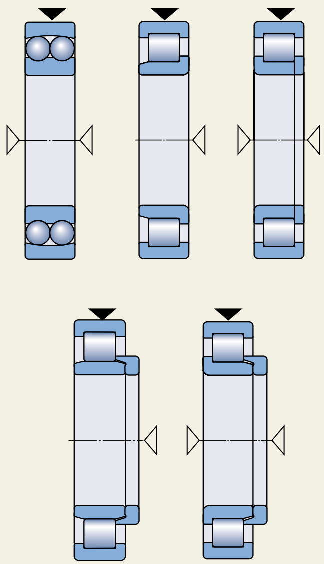


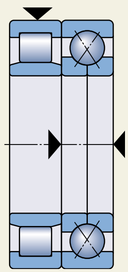
Fig 13



loads of alternating direction these bearings must be combined with a second bearing. For this reason, single row angular contact ball bearings are available as “universal bearings” for paired mounting and single row taper roller bearings can be supplied as matched sets comprising two single row bearings (➔ sections “Single row angular contact ball bearings”, starting on **page 407**, and “Paired single row taper roller bearings”, starting on **page 667**).

When the axial component of combined loads is large, it may be supported independently from the radial load by a separate bearing. In addition to the thrust bearings, some radial bearings, e.g. deep groove ball bearings or four-point contact ball bearings (➔ **fig** 14) are suitable for this task. To make sure that the bearing is only subjected to the axial load in such cases, the bearing outer ring must be mounted with radial clearance.

Fig 14



Moment load

When a load acts eccentrically on a bearing, a tilting moment will occur. Double row bearings, e.g. deep groove or angular contact ball bearings, can accommodate tilting moments, but paired single row angular contact ball bearings or taper roller bearings arranged face-to-face, or better still back-to-back, are more suitable (➔ **fig** 15).

Misalignment

Angular misalignments between the shaft and housing occur, for example, when the shaft bends (flexes) under the operating load, when the bearing seatings in the housing are not machined to the same height or when shafts are supported by bearings in separate housings that are too far apart.

Rigid bearings, i.e. deep groove ball bearings and cylindrical roller bearings cannot accommodate any misalignment, or can only accommodate very minor misalignments, unless by force. Self-aligning bearings, i.e. self-aligning ball bearings, spherical roller bearings, toroidal roller bearings and spherical roller thrust bearings (➔ **fig** 16), on the other hand, can accommodate misalignment produced under operating loads and can also compensate for initial errors of misalignment resulting from machining or mounting errors. Values for the permissible misalignments are given in the introductory text of the table section. If the expected misalignment exceeds the permissible values, please contact the SKF application engineering service.

Thrust ball bearings with sphered housing washers and seating rings, Y-bearing units and alignment needle roller bearings (➔ **fig** 17) can compensate for initial misalignment arising from machining or mounting errors.

Precision

Bearings with higher precision than Normal are required for arrangements that must have high running accuracy (e.g. machine tool spindle arrangements) as well as those applications where very high speeds are required.

The introductory text to each table section contains information regarding the tolerance classes to which the bearings in that section are produced. SKF produces a comprehensive range of high precision bearings, including single row angular contact ball bearings, single and double row cylindrical roller bearings and single and double direction angular contact thrust ball bearings (➔ the SKF catalogue “High-precision bearings”).

Fig 15

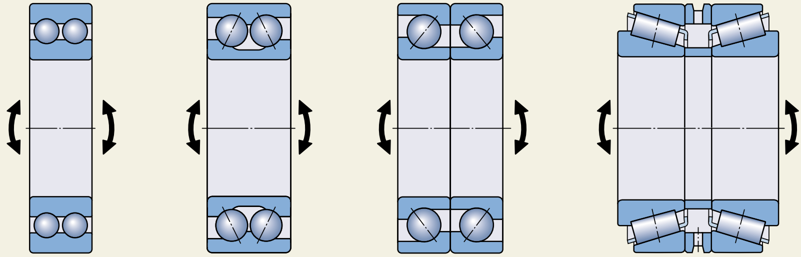


Fig 16

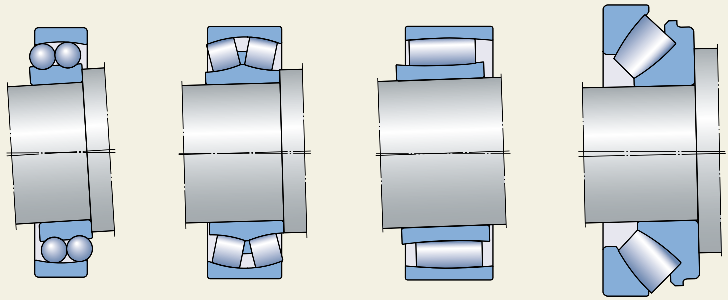
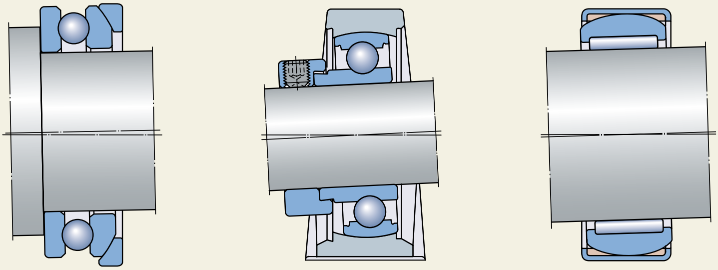


Fig 17



Speed

The permissible operating temperature limits the speed at which rolling bearings can be operated. Bearing types with low friction and correspondingly low heat generation inside the bearing are therefore the most suitable for high-speed operation.

The highest speeds can be achieved with deep groove ball bearings and self-aligning ball bearings (➔ **fig** 18) when loads are purely radial and with angular contact ball bearings (➔ **fig** 19) for combined loads. This is particularly true of high precision angular contact ball bearings or deep groove ball bearings with ceramic rolling elements.

Because of their design, thrust bearings cannot accommodate as high speeds as radial bearings.

Fig 18

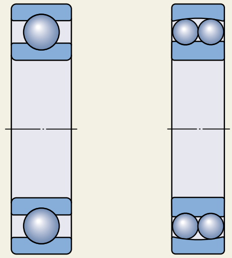
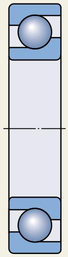


Fig 19



Quiet running

In certain applications, e.g. small electric motors for household appliances or office machinery, the noise produced in operation is an important factor and can influence the bearing choice. SKF deep groove ball bearings are produced specifically for these applications (➔ **fig** 18).

Stiffness

The stiffness of a rolling bearing is characterised by the magnitude of the elastic deformation (resilience) in the bearing under load. Generally this deformation is very small and can be neglected. In some cases, however, e.g. spindle bearing arrangements for machine tools or pinion bearing arrangements, stiffness is important.

Because of the contact conditions between the rolling elements and raceways, roller bearings e.g. cylindrical or taper roller bearings (➔ **fig** 20), have a higher degree of stiffness than ball bearings. Bearing stiffness can be further enhanced by applying a preload (➔ section "Bearing preload", starting on **page 206**) .

Fig 20

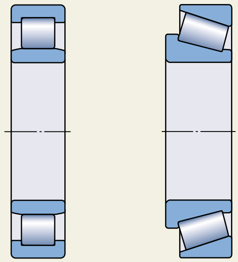


Fig 21

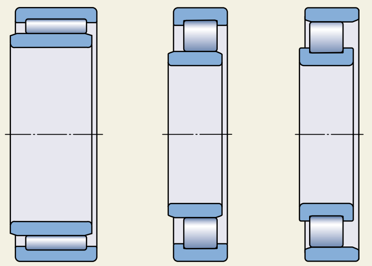


Fig 22

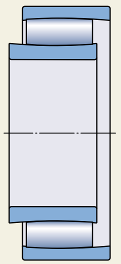
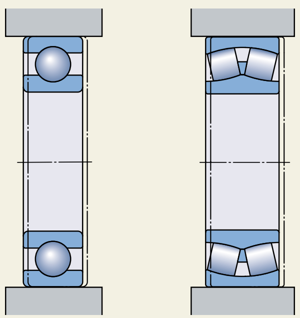


Fig 23



Axial displacement

Shafts, or other rotating machine components, are generally supported by a locating and a non-locating bearing (➔ section "Bearing arrangements", starting on **page 160**).

Locating bearings provide axial location for the machine component in both directions. The most suitable bearings for this are those that can accommodate combined loads, or can provide axial guidance in combination with a second bearing (➔ matrix on **page 46** and **47**).

Non-locating bearings must permit shaft movement in the axial direction, so that the bearings are not overloaded when, for example, thermal expansion of the shaft occurs. The most suitable bearings for the non-locating position include needle roller bearings and NU and N design cylindrical roller bearings (➔ **fig** 21). NJ design cylindrical roller bearings and some full complement design cylindrical roller bearings can also be used.

In applications where the required axial displacement is relatively large and also the shaft may be misaligned, the CARB toroidal roller bearing is the ideal non-locating bearing (➔ **fig** 22).

All of these bearings permit axial displacement of the shaft with respect to the housing within the bearing. Values for the permissible axial displacement within the bearing are given in the relevant product tables.

If non-separable bearings, e.g. deep groove ball bearings or spherical roller bearings (➔ **fig** 23) are used as non-locating bearings, one of the bearing rings must have a loose fit (➔ section "Radial location of bearings", starting on **page 164**).

Mounting and dismounting

Cylindrical bore
Bearings with a cylindrical bore are easier to mount and dismount if they are of separable design rather than non-separable, particularly if interference fits are required for both rings. Separable bearings are also preferable if frequent mounting and dismounting are required, because the ring with rolling element and cage assembly of these separable bearings can be fitted independently of the other ring, e.g. four-point contact ball bearings, cylindrical, needle and taper roller bearings (➔ **fig** 24) as well as ball and roller thrust bearings.

Tapered bore
Bearings with a tapered bore (➔ **fig** 25) can easily be mounted on a tapered journal or on a cylindrical shaft seating using an adapter or withdrawal sleeve (➔ **fig** 26) or a stepped sleeve.

Fig 24

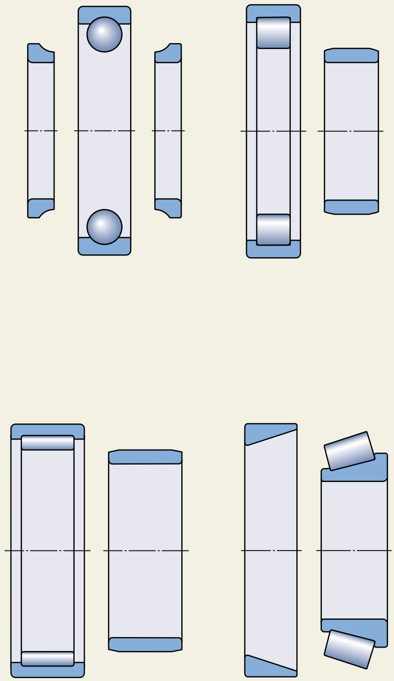


Fig 25

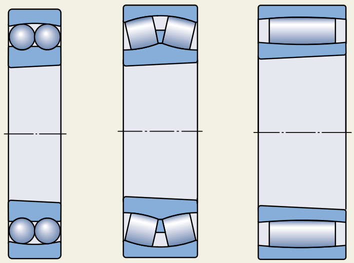
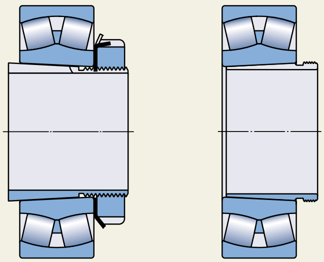


Fig 26



Integral seals

The selection of a seal is of vital importance to the proper performance of the bearing. SKF supplies bearings with integral

- shields (➔ **fig** 27),
- low-friction seals (➔ **fig** 28),
- contact seals (➔ **fig** 29).

that can provide an economic and space-saving solution for many applications. A large number of sizes are available for

- deep groove ball bearings
- angular contact ball bearings
- self-aligning ball bearings
- cylindrical roller bearings
- needle roller bearings
- spherical roller bearings
- CARB toroidal roller bearings
- cam rollers
- Y-bearings and Y-bearing units

All bearings with integral seals on both sides are filled with a grease of appropriate quality and quantity.

Fig 27

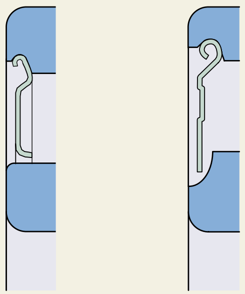


Fig 28

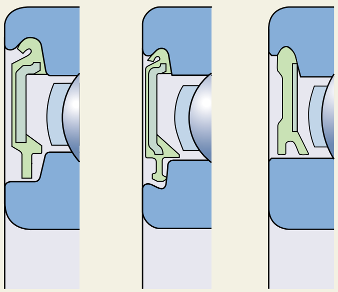
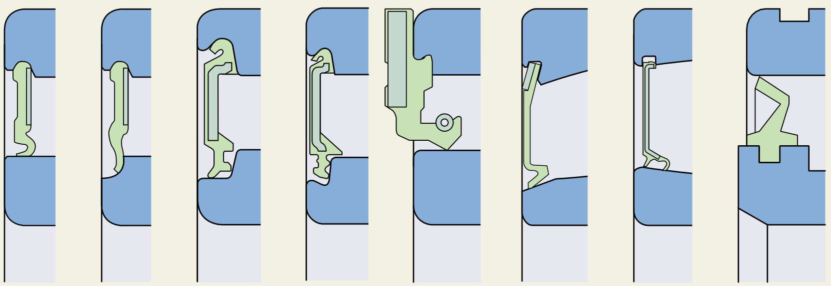


Fig 29



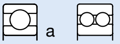
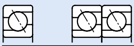
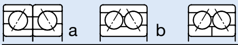
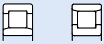
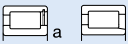
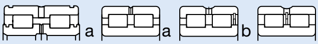
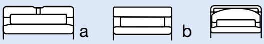
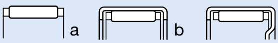
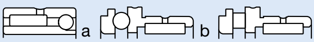
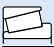
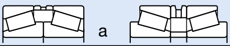
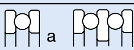
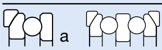
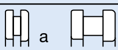
The matrix can only provide a rough guide so that in each individual case it is necessary to make a more qualified selection referring to the information given on the preceding pages or the detailed information in the text preceding each table section. If several designs of the bearing type are shown adjacent to each other, the relevant information is indicated by the same small letter used to identify the individual design.

Symbols:

+ + + excellent	– poor
+ + good	– – unsuitable
+ fair	← single direction
	←→ double direction

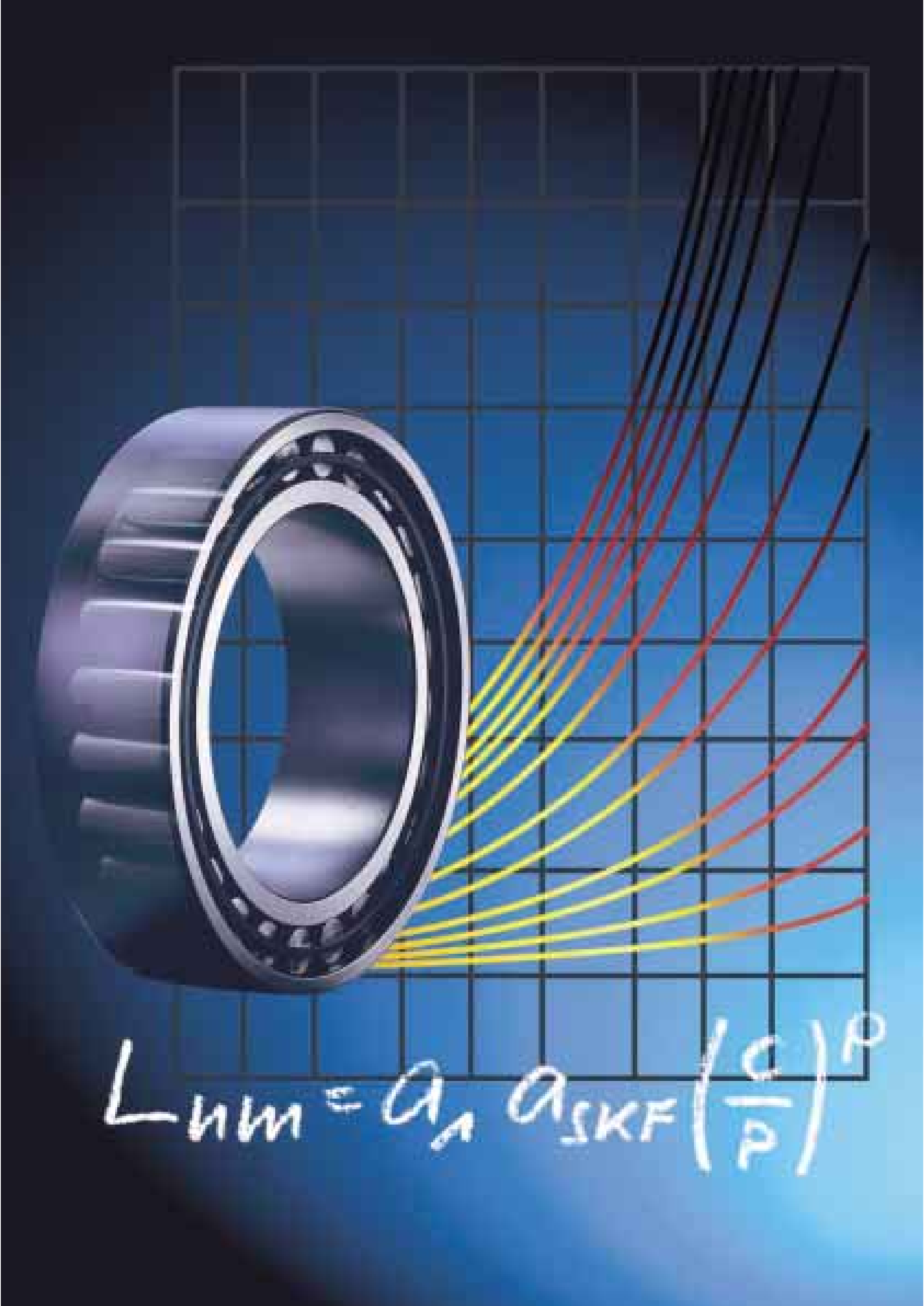
Bearing types – design and characteristics

Design

Bearing type		tapered bore	shields or seals	self-aligning	non-seperable	separable
Deep groove ball bearings	 a b		a			
Angular contact ball bearings, single row						
matched single row, double row	 a b c		b		a, b	c
four-point contact						
Self-aligning ball bearings						
Cylindrical roller bearings, with cage						
	 a a b b					
full complement, single row	 a b				a	b
full complement, double row	 a a b c		a			
Needle roller bearings, with steel rings	 a b c		a	c		
assemblies/drawn cups	 a b c		b, c			
combined bearings	 a b c		b, c			
Taper roller bearings, single row						
matched single row	 a b					
Spherical roller bearings						
CARB toroidal roller bearings, with cage						
full complement						
Thrust ball bearings	 a b					
with sphered housing washer	 a b					
Needle roller thrust bearings Cylindrical roller thrust bearings	 a b					
Spherical roller thrust bearings						

Characteristics
Suitability of bearings for

purely radial load	purely axial load	combined load	moment load	high speed	high running accuracy	high stiffness	quiet running	low friction	compensation for misalignment in operation	compensation for errors of alignment (initial)	locating bearing arrangements	non-locating bearing arrangements	axial displacement possible in bearing
+	+ ←→	+ ←→	– b+	+++ b+	+++ b+	+	+++	+++	–	–	++ ←→	+	––
+	+ ←	++ ←	–	++	+++	+	++	++	–	–	++ ←	––	––
++	+ ←→	++ ←→	+	+	++	++	+	+	––	––	++ ←→	+	––
–	++ ←→	+ ←→	+	++	+	+	+	+	––	––	++ ←→	–	––
+	–	–	––	+++	++	–	++	+++	+++	+++	+ ←→	+	––
++	––	––	––	++	++	++	++	++	–	–	––	+++	+++
++	+ a← b←→	+ a← b←→	––	++	++	++	+	++	–	–	++ a← b←→	+ a←	+ a←
+++	–	+ ←	––	–	+	+++	–	–	–	–	+ ←	+	+
+++	–	+ a←→ b←	+	–	+	+++	–	–	––	––	+ a←→ b←	+ c	+ b← c←→
++	––	––	––	+	+ a++	a++ b++	+	–	––	–– c++	––	+++	+++
++	––	––	––	+	+	++	+	–	––	––	––	+++	+++
+	+ c++ ←	+ ←	–	+	+	++	+	–	––	––	+ ←	––	––
++	++ ←	+++ ←	–	+	+	++	+	+	–	–	+++ ←	––	––
+++	++ ←→	+++ ←→	+	+	+	+++	+	+	–	––	+++ ←→	–	––
+++	+ ←→	+++ ←→	––	+	+	++	+	+	+++	+++	++ ←→	+	––
+++	––	––	––	+	+	++	+	+	+++	+++	––	+++	+++
+++	––	––	––	–	+	+++	+	+	+++	+++	––	+++	+++
––	+ a← b←→	––	––	–	++ a	+	–	+	–	––	++ a← b←→	––	––
––	+ a← b←→	––	––	–	+	+	–	+	–	++	++ a← b←→	––	––
––	++ ←	––	––	–	a+ b++	++	–	–	––	––	++ ←	––	––
––	+++ ←	+ ←	––	–	+	++	–	+	+++	+++	+++ ←	––	––


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

Selection of bearing size

The bearing size to be used for an application can be initially selected on the basis of its load ratings in relation to the applied loads and the requirements regarding service life and reliability. Values for the basic dynamic load rating C and the basic static load rating C_0 are quoted in the product tables. Both static and dynamic bearing load conditions have to be independently verified. Static loads are not only those that are applied with the bearing at rest or at very low rotational speeds (n < 10 r/min) but should include checking the static safety of heavy shock loads (very short duration loads). Dynamic loads should also be checked using a representative spectrum of load conditions on the bearing. The load spectrum should include any peak (heavy) loads that may occur on rare occasions.

System approach and bearing reliability

In the SKF life rating equation the stress resulting from the external loads is considered together with stresses originated by the surface topography, lubrication and kinematics of the rolling contact surfaces. The influence on bearing life of this combined stress system provides a better prediction of the actual performance of the bearing in a particular application.

Due to its complexity, a detailed description of the theory is beyond the scope of this catalogue. Therefore, a simplified "catalogue" approach is presented under the heading "SKF rating life". This enables users to fully exploit bearing life potential, to undertake controlled downsizing, and to recognise the influence of lubrication and contamination on bearing service life.

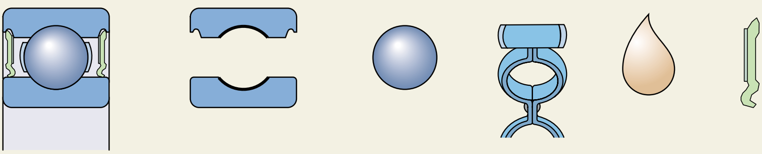
Metal fatigue of the rolling contact surfaces is generally the dominant failure mechanism in rolling bearings. Therefore, a criterion based on raceway fatigue is generally sufficient for the selection and sizing of a rolling bearing for a given application. International standards such as ISO 281 are based on metal fatigue of the rolling contact surfaces. Nevertheless, it is important to remember that the complete bearing can be viewed as a system in which the life of each component, i.e. cage, lubricant and seal (→ **fig 1**), when present, equally contributes and in some cases dominates the effective endurance of the bearing. In theory the optimum service life is achieved when all the components reach the same life.

In other words, the calculated life will correspond to the actual service life of the bearing when the service life of other contributing mechanisms is at least as long as the calculated bearing life. Contributing mechanisms can include the cage, seal and lubricant. In practice metal fatigue is most often the dominating factor.

Fig 1

Bearing system life

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



Load ratings and life

Static bearing loads

The basic static load rating C_0 is used in calculations when the bearings are to

- rotate at very slow speeds ($n < 10$ r/min),
- perform very slow oscillating movements,
- be stationary under load for certain extended periods.

It is also most important to check the safety factor of short duration loads, such as shock or heavy peak loads that act on a rotating (dynamically stressed) bearing or with the bearing at rest.

The basic static load rating as defined in ISO 76:1987 corresponds to a calculated contact stress at the centre of the most heavily loaded rolling element/raceway contact of

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

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

Verification of the static bearing loads is performed checking the static safety factor of the application, which is defined as

$$s_0 = C_0/P_0$$

where
C_0 = basic static load rating, kN
P_0 = equivalent static bearing load, kN
s_0 = static safety factor

The maximum load that can occur on a bearing should be used in the calculation of the equivalent static bearing load. Further information about the advised values for the safety factor and its calculation can be found in the section “Selecting bearing size using the static load carrying capacity”, starting on **page 76**.

Dynamic bearing loads and life

The basic dynamic load rating C is used for calculations involving dynamically stressed bearings, i.e. a bearing, that rotates under load. It expresses the bearing load that will give an ISO 281:1990 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 281:1990. The load ratings given in this catalogue apply to chromium steel bearings, heat-treated to a minimum hardness of 58 HRC, and operating under normal conditions.

The SKF Explorer class bearings account among others, for improvements in material and manufacturing techniques applied by SKF and apply update factors to calculate the basic dynamic load ratings according to ISO 281:1990.

The life of a rolling bearing is defined as

- the number of revolutions or
- the number of operating hours at a given speed,

which the bearing is capable of enduring before the first sign of metal fatigue (flaking, spalling) occurs on one of its rings or rolling elements.

Practical experience shows that seemingly identical bearings operating under identical conditions have different individual endurance lives. A clearer definition of the term “life” is therefore essential for the calculation of the bearing size. All information presented by SKF on dynamic load ratings is based on the life that 90 % of a sufficiently large group of apparently identical bearings can be expected to attain or exceed.

There are several other types of bearing life. One of these is “service life” which represents the actual life of a bearing in real operating conditions before it fails. Note that individual bearing life can only be predicted statistically. Life calculations refer only to a bearing population and a given degree of reliability, i.e. 90 %, furthermore field failures are not generally caused by fatigue, but are more often caused by contamination, wear, misalignment, corrosion, or as a result of cage, lubrication or seal failure.

Another “life” is the “specification life”. This is the life specified by an authority, for example, based on hypothetical load and speed data supplied by the same authority. It is generally a requisite L_{10} basic rating life and based on experience gained from similar applications.

Selecting bearing size using the life equations

Basic rating life

The basic rating life of a bearing according to ISO 281:1990 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 the equation

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

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 basic rating life can deviate significantly from the actual service life in a given application. Service life in a particular application depends on a variety of influencing factors including lubrication, the degree of contamination, misalignment, proper installation and environmental conditions.

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

ISO 281:1990/Amd 2:2000 also makes provisions for bearing manufacturers to recommend a suitable method for calculating the life modification factor to be applied to a bearing based on operating conditions. The

SKF life modification factor a_{SKF} applies the concept of a fatigue load limit P_u analogous to that used when calculating other machine components. The values of the fatigue load limit are given in the product tables. Furthermore, the SKF life modification factor a_{SKF} makes use of the lubrication conditions (viscosity ratio κ) and a factor η_c for contamination level to reflect the application's operating conditions.

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

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

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

$$L_{nmh} = \frac{10^6}{60\, n} L_{nm}$$

where

L_{nm} = SKF rating life (at 100 – n % reliability), millions of revolutions

L_{nmh} = SKF rating life (at 100 – n % reliability), operating hours

L_{10} = basic rating life (at 90 % reliability), millions of revolutions

a_1 = life adjustment factor for reliability (➔ **table 1**)

a_{SKF} = SKF life modification factor (➔ **diagrams 1** to **4**)

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

In some cases it is preferable to express bearing life in units other than millions of revolutions or hours. For example, bearing life for axle bearings used in road and rail vehicles is commonly expressed in terms of kilometres travelled. To facilitate the calculation of bearing life into different units, **table 2**, **page 58**, provides the conversion factors commonly used.

SKF life modification factor a_{SKF}

As mentioned, this factor represents the relationship between the fatigue load limit ratio (P_u/P), the lubrication condition (viscosity ratio κ) and the contamination level in the bearing (η_c). Values for the factor a_{SKF} can be obtained from four diagrams, depending on bearing type, as a function of η_c (P_u/P) for SKF standard and SKF Explorer bearings and different values of the viscosity ratio κ:

Diagram 1: Radial ball bearings, **page 54**
Diagram 2: Radial roller bearings, **page 55**
Diagram 3: Thrust ball bearings, **page 56**
Diagram 4: Thrust roller bearings, **page 57**

The diagrams are drawn for typical values and safety factors of the type normally associated with fatigue load limits for other mechanical components. Considering the simplifications inherent of 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.

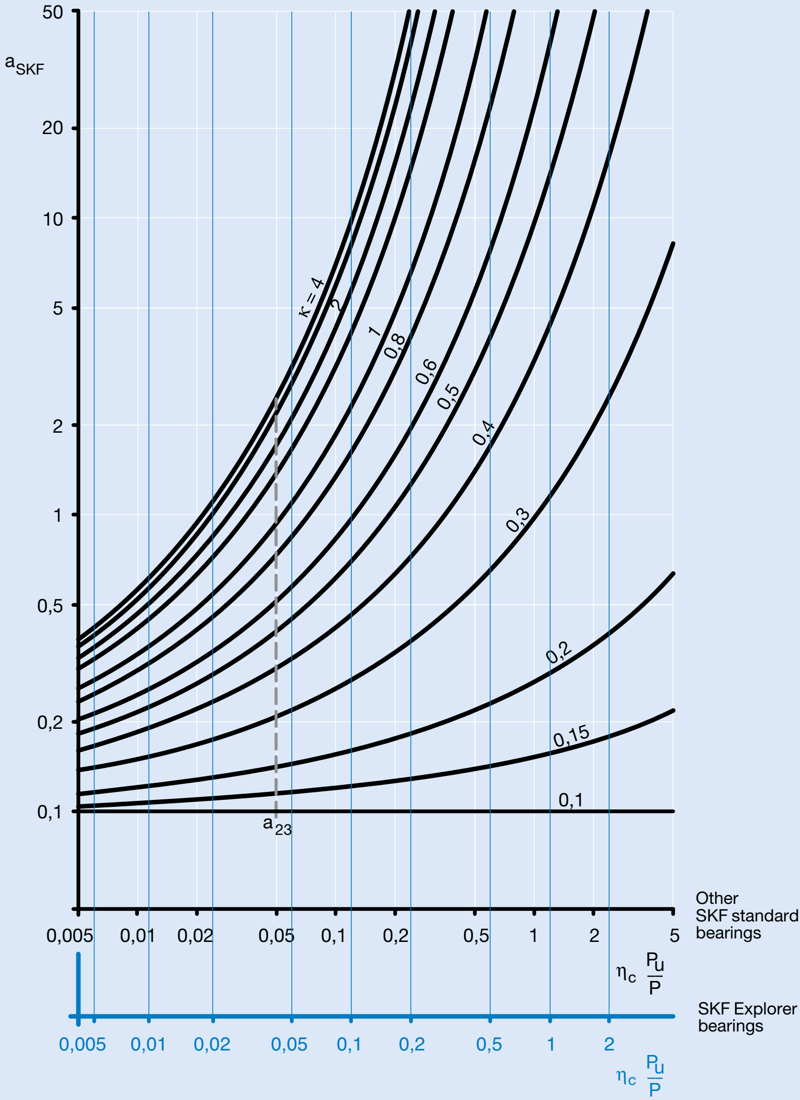
Table 1

Values for life adjustment factor a_1

Reliability %	**Failure probability** %	**Rating life** L_{nm}	**Factor** a_1
90	10	L_{10m}	1
95	5	L_{5m}	0,62
96	4	L_{4m}	0,53
97	3	L_{3m}	0,44
98	2	L_{2m}	0,33
99	1	L_{1m}	0,21

Diagram 1

Factor a_{SKF} for radial ball bearings

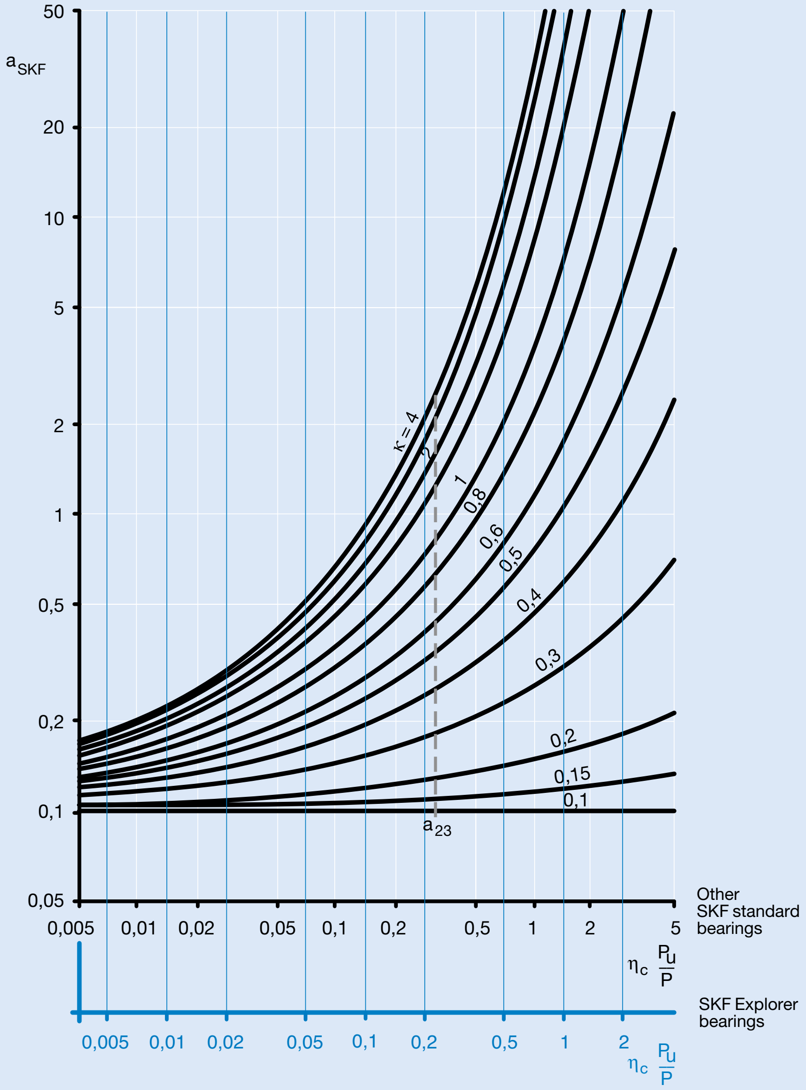


If $\kappa > 4$, use curve for $\kappa = 4$

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

The dotted line marks the position of the old a_{23} (κ) scale, where $a_{SKF} = a_{23}$

Factor a_{SKF} for radial roller bearings



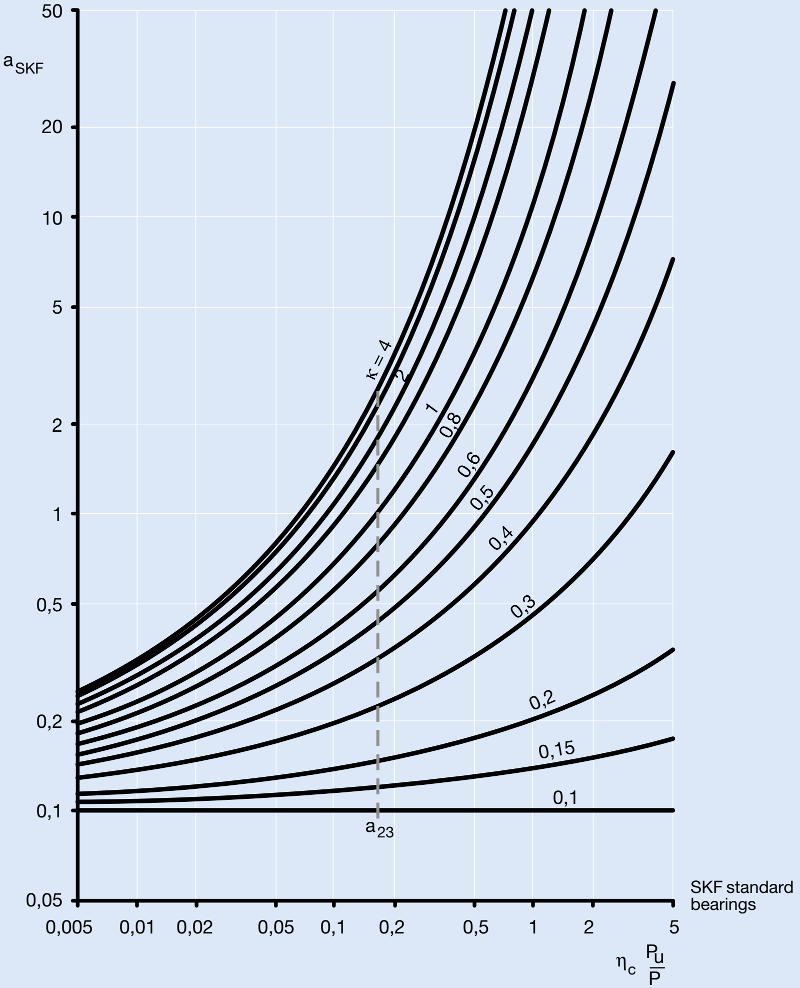
If $\kappa > 4$, use curve for $\kappa = 4$

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

The dotted line marks the position of the old a_{23} (κ) scale, where $a_{SKF} = a_{23}$

Diagram 3

Factor a_{SKF} for thrust ball bearings

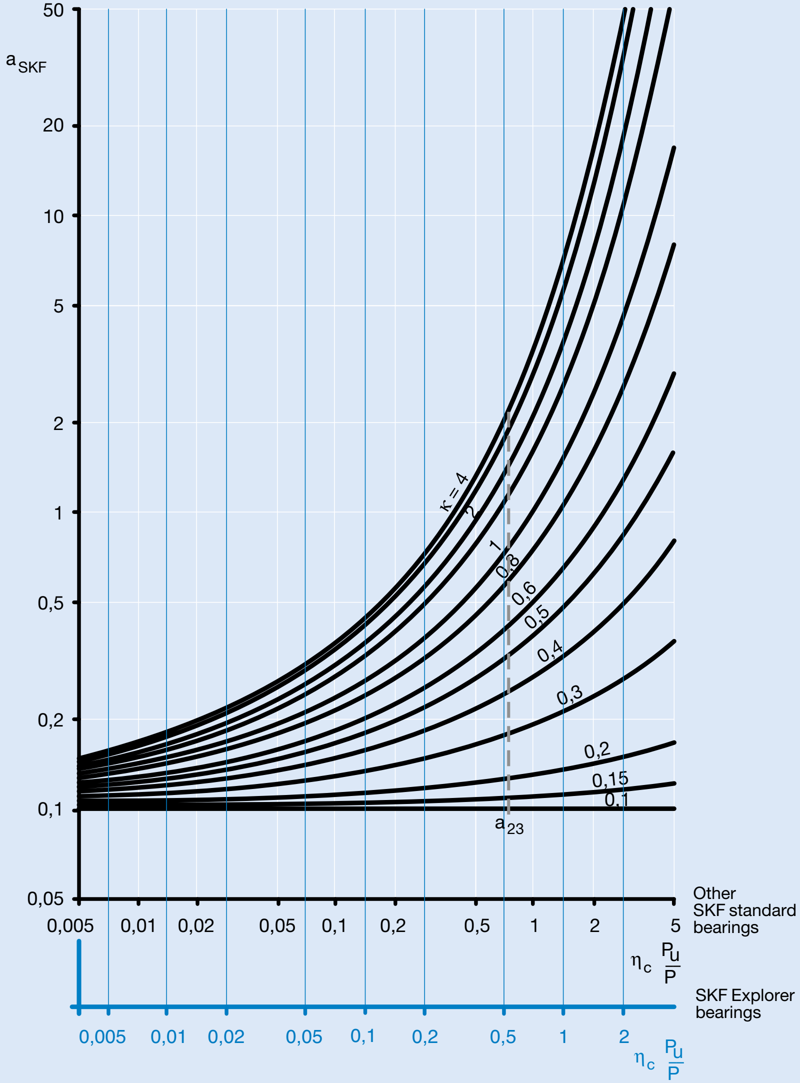


If κ > 4, use curve for κ = 4

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

The dotted line marks the position of the old a_{23} (κ) scale, where $a_{SKF} = a_{23}$

Factor a_{SKF} for thrust roller bearings



If κ > 4, use curve for κ = 4

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

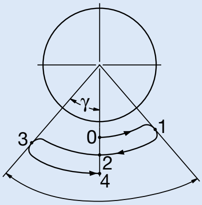
The dotted line marks the position of the old a_{23} (κ) scale, where $a_{SKF} = a_{23}$

Calculation of the life modification factors a_{SKF}

SKF engineering programs – CADalog, or the “SKF Interactive Engineering Catalogue”, available on CD-ROM or online at www.skf.com – can also be used to facilitate the calculation of the factor a_{SKF}. Furthermore, SKF has also developed sophisticated computer programs incorporating the SKF rating life equation directly at the rolling contact stress level thus permitting other factors influencing bearing life, such as misalignment, shaft deflection and housing deformation to be taken into account (➔ section “SKF calculation tools”, starting on **page 82**).

Table 2

Units conversion factors for bearing life



The complete oscillation = 4 γ,
i.e. from point 0 to point 4

Basic units	**Conversion factor** Millions of revolutions	Operating hours	Kilometres travelled	Millions of oscillation cycles
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 kilometre	$\frac{10^3}{\pi\,D}$	$\frac{10^9}{60\,n\,\pi\,D}$	1	$\frac{180 \times 10^3}{2\,\gamma\,\pi\,D}$
1 million oscillation cycles	$\frac{2\,\gamma}{180}$	$\frac{2\,\gamma\,10^6}{180 \times 60\,n}$	$\frac{2\,\gamma\,\pi\,D}{180 \times 10^3}$	1

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

Lubrication conditions – the viscosity ratio κ
The effectiveness of the lubricant is primarily determined by the degree of surface separation between the rolling contact surfaces. If an adequate lubricant film is to be formed, the lubricant must have a given minimum viscosity when the application has reached its normal operating temperature. The condition of the lubricant is described by the viscosity ratio κ as the ratio of the actual viscosity ν to the rated viscosity ν_1 for adequate lubrication, both values being considered when the lubricant is at normal operating temperature (➔ section "Selection of lubricating oil", starting on **page 252**).

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

where
κ = viscosity ratio
ν = actual operating viscosity of the lubricant, mm^2/s
ν_1 = rated viscosity depending on the bearing mean diameter and rotational speed, mm^2/s

In order to form an adequate lubricant film between the rolling contact surfaces, the lubricant must retain a certain minimum viscosity when the lubricant is at operating temperature. The rated viscosity ν_1, required for adequate lubrication, can be determined from **diagram 5**, **page 60**, using the bearing mean diameter $d_m = 0,5\,(d + D)$, mm, and the rotational speed of the bearing n, r/min. This diagram has been revised taking the latest findings of tribology in rolling bearings into account.

When the operating temperature is known from experience or can otherwise be determined, the corresponding viscosity at the internationally standardized reference temperature of 40 °C can be obtained from **diagram 6**, **page 61**, or can be calculated. The diagram is compiled for a viscosity index of 95. **Table 3** lists the viscosity grades according to ISO 3448:1992 showing the range of viscosity for each class at 40 °C. Certain bearing types, e.g. spherical roller bearings, taper roller bearings, and spherical roller thrust bearings, normally have a higher operating temperature than other bearing types, e.g. deep groove ball bearings and cylindrical roller bearings, under comparable operating conditions.

Consideration of EP additives
It is known that some EP additives in the lubricant can extend bearing service life where lubrication might otherwise be poor, e.g. when $\kappa < 1$ and if the factor for the contamination level $\eta_c \geq 0,2$ according to DIN ISO 281 Addendum 1:2003, a value of $\kappa = 1$ can be used in the calculation if a lubricant with proven effective EP additives is used. In this case the life modification factor a_{SKF} has to be limited to ≤ 3, but not less than a_{SKF} for normal lubricants.

For the remaining range, the life modification factor a_{SKF} can be determined using the actual κ of the application. In case of severe contamination, i.e. contamination factor $\eta_c < 0,2$, the possible benefit of an

Table 3

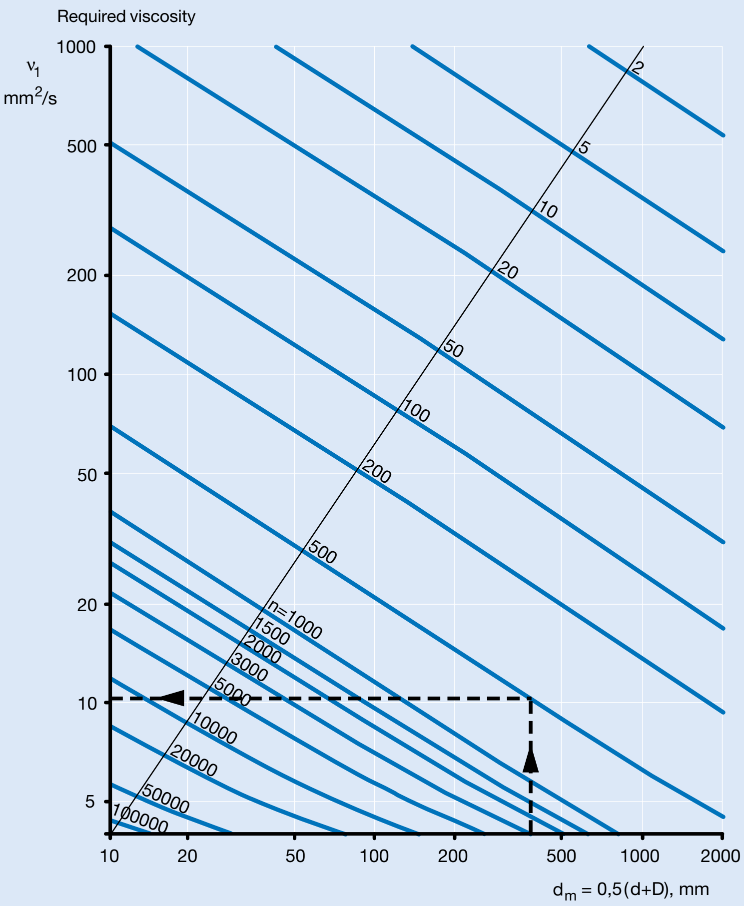
ISO viscosity classification to ISO 3448

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

EP additive has to be proved by testing. Reference should also be made to the information regarding EP additives presented in the section “Lubrication”, starting on **page 229**.

Diagram **5**

Required kinematic viscosity at operating temperature

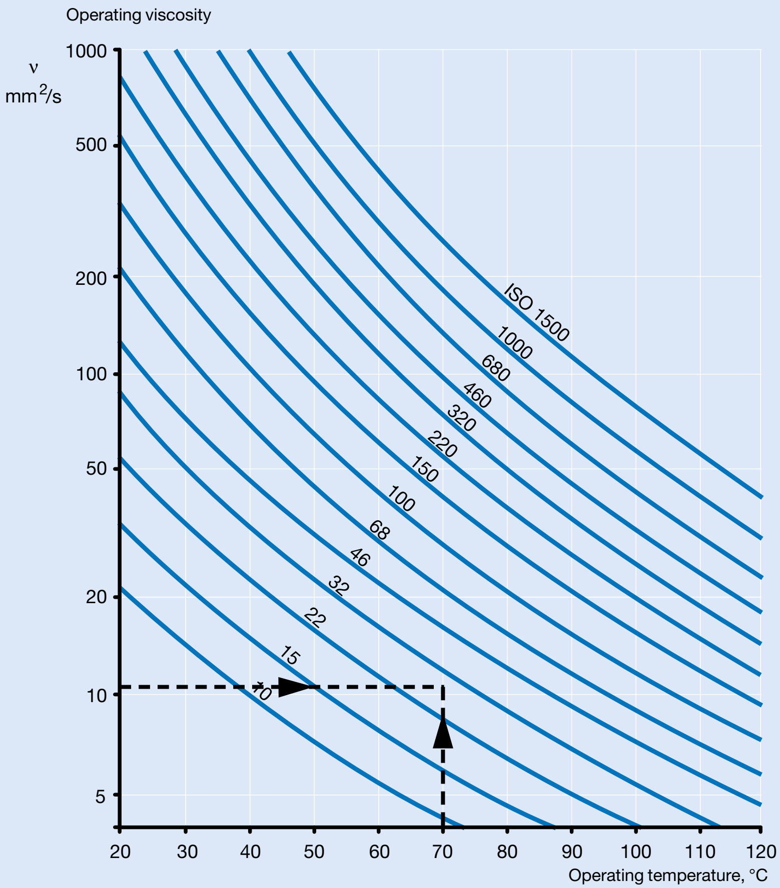


Calculation example

A bearing having a bore diameter d = 340 mm and an outside diameter D = 420 mm is required to operate at a speed n = 500 r/min. Since $d_m = 0{,}5\,(d + D)$, $d_m = 380$ mm, from **diagram 5**, the minimum rated viscosity ν_1 required to give adequate lubrication at the operating temperature is approximately 11 mm^2/s. From **diagram 6**, assuming that the operating temperature of the bearing is 70 °C, it is found that a lubricant to an ISO VG 32 viscosity class, with an actual viscosity ν of at least 32 mm^2/s at the reference temperature of 40 °C will be required.

Diagram 6

Required kinematic viscosity at reference temperature



Factor η_c for contamination level

This factor was introduced to consider the contamination level of the lubricant in the bearing life calculation. The influence of contamination on bearing fatigue depends on a number of parameters including bearing size, relative lubricant film thickness, size and distribution of solid contaminant particles, types of contamination (soft, hard etc.) The influence of these parameters on bearing life is complex and many of the parameters are difficult to quantify. It is therefore not possible to allocate precise values to η_c that would have general validity. However, some guideline values are given in **table 4**.

If the bearing is used in an application with a satisfactory record in the field and past life calculations were based on the use of the old adjustment factor a_{23} then a corresponding (implicit value) η_c factor can be derived to give an a_{SKF} equivalent to the a_{23} adjustment as explained in the section "A special case – the adjustment factor a_{23}" on **page 68**.

Note that this approach will probably indicate only an approximate value of the effective factor η_c for the contamination level of the application. A second method to obtain a value for the factor η_c that is representative for an application is by quantifying the contamination level of the lubricant as input for the evaluation of the value for the factor η_c.

Table 4

Guideline values for factor η_c for different levels of contamination

Condition	**Factor η_c** [1] for bearings with diameter d_m < 100 mm	$d_m \geq$ 100 mm
Extreme cleanliness Particle size of the order of the lubricant film thickness Laboratory conditions	1	1
High cleanliness Oil filtered through extremely fine filter Conditions typical of bearings greased for life and sealed	0,8 …0,6	0,9 … 0,8
Normal cleanliness Oil filtered through fine filter Conditions typical of bearings greased for life and shielded	0,6 … 0,5	0,8 … 0,6
Slight contamination Slight contamination in lubricant	0,5 … 0,3	0,6 … 0,4
Typical contamination Conditions typical of bearings without integral seals, coarse filtering, wear particles and ingress from surroundings	0,3 … 0,1	0,4 … 0,2
Severe contamination Bearing environment heavily contaminated and bearing arrangement with inadequate sealing.	0,1 … 0	0,1 … 0
Very severe contamination (under extreme contamination values of η_c can be outside the scale resulting in a more severe reduction of life than predicted by the equation for L_{nm})	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. In case of very heavy contamination (η_c = 0), failure will be caused by wear, the useful life of the bearing can be shorter than the rated life

ISO contamination classification and filter rating

The standard method for classifying the contamination level in a lubrication system is described in ISO 4406:1999. In this classification system the result of the solid particle counting is converted into a code using a scale number (➔ **table** 5 and **diagram** 7).

One method for checking the contamination level of the bearing oil is the microscope counting method. With this counting method two scale numbers, relating to the number of particles ≥ 5 µm and ≥ 15 µm, are used. Another method refers to automatic particle counters, where three scale numbers are used relating to the number of particles ≥ 4 µm, ≥ 6 µm and ≥ 14 µm. The classification of the contamination level comprises three scale numbers.

Typical examples of contamination level classifications for lubricating oil are –/15/12 (A) or 22/18/13 (B) as shown in **diagram** 7 on **page 65**.

Example A means that the oil contains between 160 and 320 particles ≥ 5 µm and between 20 and 40 particles ≥ 15 µm per millilitre oil. Though it would be ideal if lubricating oils were continuously filtered, the viability of a filtration system would depend on the optimization between increased costs and increased service performance of the bearing.

A filter rating is an indication of filter efficiency. The efficiency of filters is defined as the filter rating or reduction factor β, which is related to a given particle size. The higher the β value, the more efficient the filter is for the specified particle size. Therefore both the β value and the specified particle size have to be considered. The filter rating β is expressed as the relationship between the number of specified particles before and after filtering. This can be calculated as follows

$$\beta_x = \frac{n_1}{n_2}$$

where

β_x = filter rating related to a specified particle size x

x = particle size, µm

n_1 = number of particles per volume unit (100 ml) larger than x µm upstream the filter

n_2 = number of particles per volume unit (100 ml) larger than x µm downstream the filter

Note:

The filter rating β only relates to one particle size in µm, which is shown as the index e.g. β_3, β_6, β_{12}, etc. For example, a complete rating "β_6 = 75" means that only 1 of 75 particles of 6 µm or larger will pass through the filter. Therefore both the β value and the specified particle size have to be considered.

Table 5

ISO classification – allocation of scale number

Number of particles per millilitre oil over	incl.	Scale number
2 500 000		> 28
1 300 000	2 500 000	28
640 000	1 300 000	27
320 000	640 000	26
160 000	320 000	25
80 000	160 000	24
40 000	80 000	23
20 000	40 000	22
10 000	20 000	21
5 000	10 000	20
2 500	5 000	19
1 300	2 500	18
640	1 300	17
320	640	16
160	320	15
80	160	14
40	80	13
20	40	12
10	20	11
5	10	10
2,5	5	9
1,3	2,5	8
0,64	1,3	7
0,32	0,64	6
0,16	0,32	5
0,08	0,16	4
0,04	0,08	3
0,02	0,04	2
0,01	0,02	1
0,00	0,01	0

Determination of η_c when the contamination level is known

For oil lubrication, once the oil contamination level is known, either from a microscope counting or from an automatic particle counter analysis described in ISO 4406:1999, or indirectly as a result of the filtration ratio that is applied in an oil circulation system, this information can be used to determine the factor η_c for the contamination. Note that the factor η_c cannot be derived solely from the measure of oil contamination. It depends strongly on the lubrication condition, i.e. κ and the size of the bearing. A simplified method according to DIN ISO 281 Addendum 4:2003 is presented here to obtain the η_c factor for a given application. From the oil contamination code (or filtration ratio of the application), the contamination factor η_c is obtained, using the bearing mean diameter $d_m = 0{,}5\,(d + D)$, mm, and the viscosity ratio κ of that bearing (➔ **diagrams** 8 and 9, **page 66**).

Diagrams 8 and 9 give typical values for the factor η_c for circulating oil lubrication with different degrees of oil filtration and oil contamination codes. Similar contamination factors can be applied in applications where an oil bath shows virtually no increase in the contamination particles present in the system. On the other hand, if the number of particles in an oil bath continues to increase over time, due to excessive wear or the introduction of contaminants, this must be reflected in the choice of the factor η_c used for the oil bath system as indicated in DIN ISO 281 Addendum 4:2003.

For grease lubrication η_c can also be determined in a similar way, although the contamination may be difficult to measure and is therefore defined in a simple, qualitative manner.

Diagrams 10 and 11, **page 67**, give typical values for the factor η_c for grease lubrication for operating conditions of extreme cleanliness and normal cleanliness.

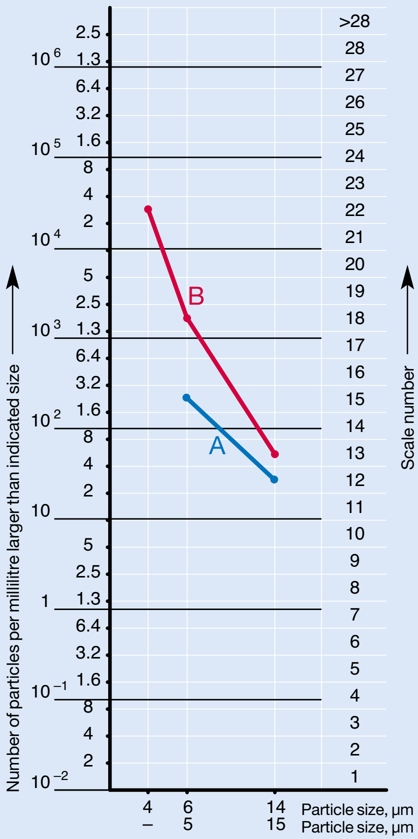
For other degrees of contamination for circulating oil, oil bath and grease lubrication, please refer to DIN ISO 281 Addendum 4:2003 or consult the SKF application engineering service.

An indication of the strong effect of contamination on fatigue life can be obtained from the following example. Several 6305 deep groove ball bearings with and without seals were tested in a highly contaminated environment (a gearbox with a considerable number of wear particles). No failures of the sealed bearings occurred and the tests were discontinued for practical reasons after the sealed bearings had run for periods which were at least 30 times longer than the experimental lives of the unsealed bearings. The unsealed bearing lives equalled 0,1 of the calculated L_{10} life, which corresponds to a factor $\eta_c = 0$ as indicated in **table** 4, **page 62**.

Diagrams 1 to 4, starting on **page 54**, indicate the importance of cleanliness in lubrication by the rapid reduction in the values for the factor a_{SKF} with a diminishing value of the factor η_c. When bearings with integral seals are used, contamination of the bearing can be kept to a minimum, but the life of the lubricant and the seals must also be taken into consideration.

Diagram 7

ISO classification and examples for particle counting



A = microscope particle counting (–/15/12)
B = automatic particle counter (22/18/13)

Diagram 8

Contamination factor η_c for
- **circulating oil lubrication**
- **solid contamination level –/15/12 to ISO 4406:1999**
- **filter rating $\beta_{12} = 200$**

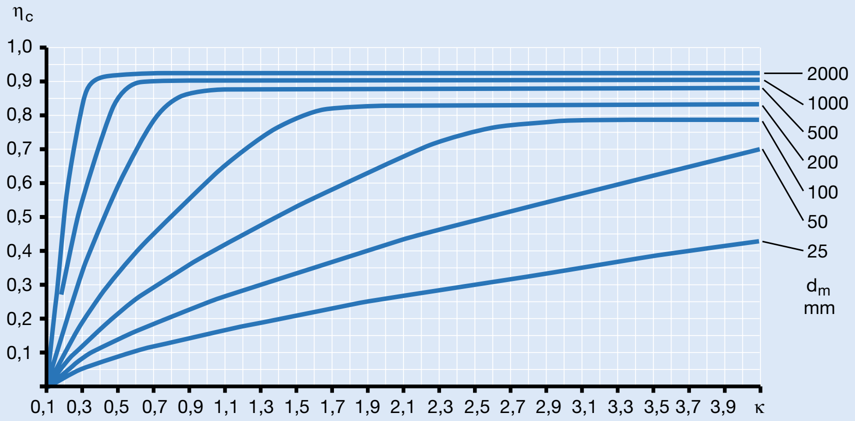


Diagram 9

Contamination factor η_c for
- **circulating oil lubrication**
- **solid contamination level –17/14 to ISO 4406:1999**
- **filter rating factor $\beta_{25} = 75$**

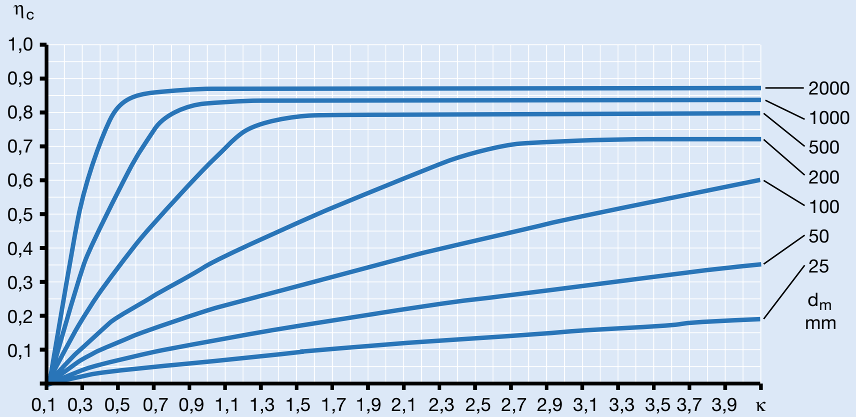


Diagram 10

Contamination factor η_c for grease lubrication, extreme cleanliness

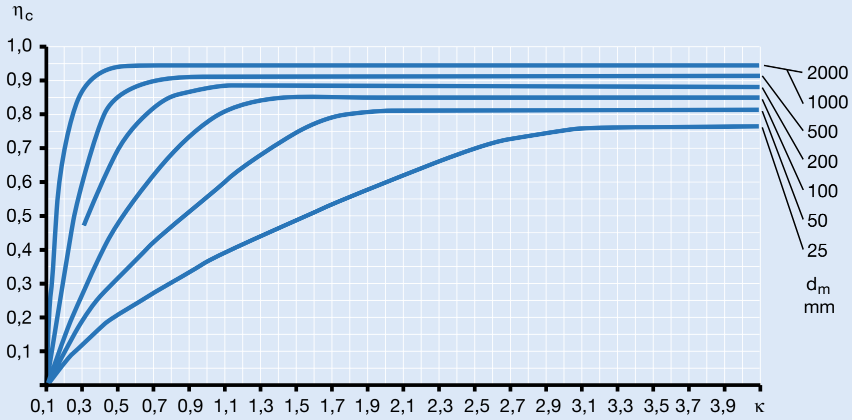
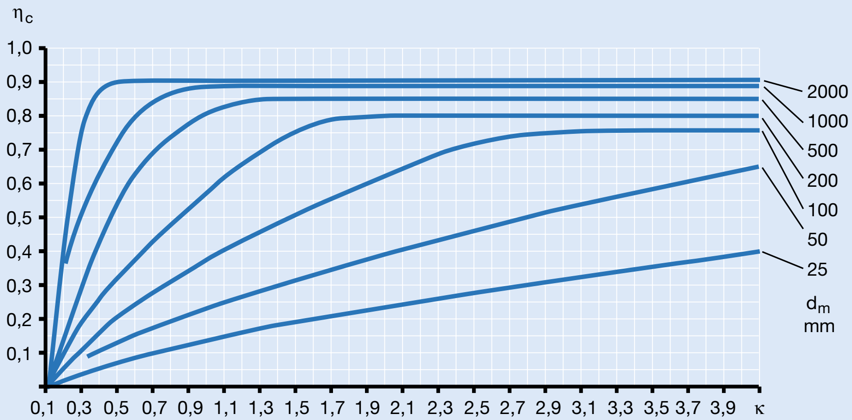


Diagram 11

Contamination factor η_c for grease lubrication, normal cleanliness



A special case – the adjustment factor a_{23}
In previous SKF catalogues the basic rating life was adjusted using the factor a_{23} for material and lubrication. This factor was introduced by SKF in 1975.

In ISO 281:1990/Amd 2:2000 reference is made to this type of life adjustment as a special case of the more general life modification factor a_{SKF}. The a_{23} adjustment implies a specific value of the "contamination-load ratio" $[\eta_c\,(P_u/P)]_{23}$ used in the diagrams for the SKF life modification factor a_{SKF}. Because the factor a_{23} is only viscosity ratio κ dependent, a κ scale is superimposed on the κ curves of **diagrams 1** to **4**, starting on **page 54**, for the factor a_{SKF} at the point where $\eta_c\,(P_u/P) = [\eta_c\,(P_u/P)]_{23}$. The factor η_c for contamination level thus becomes

$$\eta_c = [\eta_c\,(P_u/P)]_{23}/(P_u/P)$$

The location of the point where $\eta_c\,(P_u/P) = [\eta_c\,(P_u/P)]_{23}$ is marked by a dotted line and the values are listed in **table 6** for SKF standard as well as for SKF Explorer bearings. For instance, for standard radial ball bearings the corresponding η_c is

$$\eta_c = \frac{0{,}05}{P_u/P}$$

At that location of the "contamination-load ratio" $[\eta_c\,(P_u/P)]_{23} = 0{,}05$ in **diagram 1**, **page 54**, $a_{SKF} = a_{23}$ and a_{23} can be read directly from the a_{SKF} axis using the κ scale of the dotted line. The life can than be calculated with the simplified equation

$$L_{nm} = a_1\, a_{23}\, L_{10}$$

where
L_{nm} = SKF rating life (at 100 – n % reliability), millions of revolutions
L_{10} = basic rating life (at 90 % reliability), millions of revolutions
a_1 = life factor for reliability (➔ **table 1**, **page 53**)
a_{23} = adjustment factor for material and lubrication, when $\eta_c\,(P_u/P) = [\eta_c\,(P_u/P)]_{23}$ (➔ **diagrams 1** to **4**, starting on **page 54**)

Table 6

Contamination-load ratio $[\eta_c\,(P_u/P)]_{23}$

Bearing type	**Ratio $[\eta_c\,(P_u/P)]_{23}$** for SKF standard bearings	SKF Explorer bearings
Radial bearings		
Ball bearings	0,05	0,04
Roller bearings	0,32	0,23
Thrust bearings		
Ball bearings	0,16	–
Roller bearings	0,79	0,56

Using the adjustment factor a_{23} implies in practice a stress condition characterized by a value of $\eta_c (P_u/P) = [\eta_c (P_u/P)]_{23}$. If the actual $\eta_c (P_u/P)$ of the bearing is lower or higher than the $[\eta_c (P_u/P)]_{23}$ value, there will be an over or under estimation of the life performance. In other words applications with heavy loads and increased contamination or light loads and improved cleanliness are not well represented by the adjustment factor a_{23}.

For standard bearings operating at a load ratio C/P of about 5 the contamination level for $a_{SKF} = a_{23}$ will require an η_c factor of about 0,4 to 0,5. If the actual cleanliness of the application is lower than the normal level the use of the a_{23} adjustment leads to an overestimation of the life performance of the bearing. Therefore SKF recommends using only the a_{SKF} method to improve reliability in the selection of the bearing size.

The correspondence between the adjustment factors a_{23} and a_{SKF} is useful if it is required to convert applications that were traditionally designed using the adjustment factor a_{23} to the use of the more general a_{SKF} adjustment factor. Indeed many applications that have a satisfactory record of operation, initially calculated using the adjustment factor a_{23}, can be easily converted to an equivalent factor a_{SKF}.

In practice this implies the adoption of a contamination factor η_c of the application based on the “contamination-load ratios” $[\eta_c (P_u/P)]_{23}$ listed in **table 6**. The factor η_c derived in this way represents a simple approximation or the actual factor η_c. This first estimation of the factor η_c can be further improved using oil cleanliness ratings as described in the section “Determination of η_c when the contamination level is known”, starting on **page 64**. See also calculation example 2 on **page 78**.

Life calculation with variable operating conditions

In applications where bearing load varies over time both in magnitude and direction with changes of speed, temperature, lubrication conditions and level of contamination, the bearing life cannot be calculated directly without the need of the intermediate calculation step of an equivalent load related to the variable load conditions. Given the complexity of the system, this intermediate parameter would not be easy to determine and would not simplify the calculation.

Therefore, in the case of fluctuating operating conditions it is necessary to reduce the load spectrum or duty cycle of the application to a limited number of simpler load cases (➔ **diagram 12**). In case of continuously variable load, each different load level can be accumulated and the load spectrum reduced to a histogram of constant load blocks, each characterizing a given percentage or time-fraction of the operation of the application. Note that heavy and medium loads consume bearing life at a faster rate than lighter loads. Therefore it is important to have shock and peak loads well represented in the load diagram even if the occurrence of these loads is relatively rare and limited to a few revolutions.

Within each duty interval or "bin", the bearing load and operating conditions can be averaged to some constant value. Furthermore the number of operating hours or revolutions expected from each duty interval shows the life fraction required by that particular load condition. Thus for instance denoting with N_1 the number of revolutions required under the load condition P_1, and with N the total life cycle of the application, then life fraction $U_1 = N_1/N$ will be used by the load condition P_1, which has a calculated life of L_{10m1}. Under variable operating conditions bearing life can be predicted using the equation

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

where

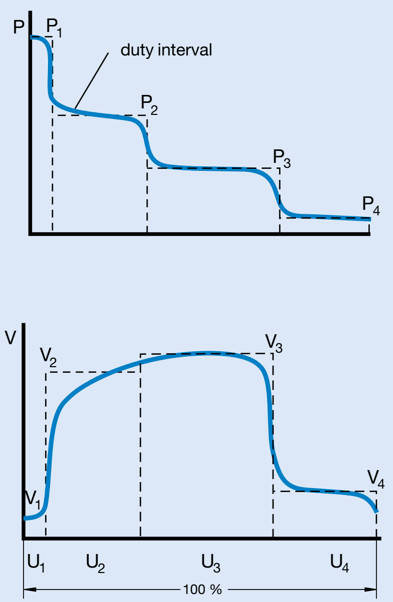
L_{10m} = rating life, millions of revolutions

$L_{10m1}, L_{10m2}, ..$ = fraction rating lives under constant conditions 1, 2, …, millions of revolutions

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

The use of this calculation method is very much dependent on the availability of representative load diagrams for the application. Note that such load history can also be derived from typical operating conditions or standard duty cycles required from that type of application.

Diagram 12



Influence of operating temperature

The dimensions of a bearing in operation change as a result of structural transformations within the material. These transformations are influenced by temperature, time and stress.

To avoid inadmissible dimensional changes in operation due to the structural transformation, bearing materials are subjected to a special heat treatment (stabilization) process (➔ **table** 7).

Depending on the bearing type, standard bearings made from through hardened and induction-hardened steels have a recommended maximum operating temperature, between 120 and 200 °C. These maximum operating temperatures are directly related to the heat treatment process. Where applicable, additional information is found in the introductory text of the product section.

If the normal operating temperatures of the application are higher than the recommended maximum temperature, a bearing with a higher stabilization class is preferred.

For applications where bearings operate continuously at elevated temperatures the dynamic load carrying capacity of the bearing may need to be adjusted.

For further information please consult the SKF application engineering service.

The satisfactory operation of bearings at elevated temperatures also depends on whether the chosen lubricant will retain its lubricating properties and whether the materials used for the seals, cages etc. are suitable (➔ sections “Lubrication”, starting on **page 229**, and “Materials for rolling bearings”, starting on **page 138**).

In general, for bearings operating at high temperatures requiring higher stability class than S1, please contact the SKF application engineering service.

Requisite rating life

When determining the bearing size it is general practice to verify the applied dynamic loads with the specification life of the application. This usually depends on the type of machine and the requirements regarding duration of service and operational reliability. In the absence of previous experience, the guideline values given in **tables** 8 and 9, **page 72**, can be used.

Table **7**

Dimensional stability

Stabilization class	Stabilization up to
SN	120 °C
S0	150 °C
S1	200 °C
S2	250 °C
S3	300 °C
S4	350 °C

Table 8

Guideline values of specification life for different types of machine

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	3 000 ... 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

Table 9

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

Type of vehicle	Specification life Millions of km
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

Dynamic bearing loads

Calculation of dynamic bearing loads
The loads acting on a bearing can be calculated according to the laws of mechanics if the external forces (e.g. forces from power transmission, work forces or inertia forces) are known or can be calculated. When calculating the load components for a single bearing, the shaft is considered as a beam resting on rigid, moment-free supports for the sake of simplification. 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 a bearing arrangement is to be calculated using readily available aids such as a pocket calculator. 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. In these programs, the bearings, shaft and housing are considered as resilient components of a system.

External forces that arise, for example, from the inherent weight of the shaft and the components that it carries, or from the weight of a vehicle, and the other inertia forces are either known or can be calculated. However, when determining the work forces (rolling forces, cutting forces in machine tools etc.), shock forces and additional dynamic forces, e.g. as a result of unbalance, it is often necessary to rely on estimates based on experience with similar machines or bearing arrangements.

Gear trains

With gear trains, 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 in the gear itself or by the input drive or power take-off. Additional dynamic forces in gears result from form errors of the teeth and from unbalanced rotating components. Because of the requirements for quiet running, gears are made to high standards of accuracy and these forces are generally so small that they can be neglected when making bearing calculations.

Additional forces arising from the type and mode of operation of the machines coupled to the gear can only be determined when the operating conditions are known. Their influence on the rating lives of the bearings is considered using an "operation" factor that takes into account shock loads and the efficiency of the gear. Values of this factor for different operating conditions can usually be found in information published by the gear manufacturer.

Belt drives

For belt drives it is necessary to take into account the effective belt pull (circumferential force), which is dependent on the transmitted torque, when calculating bearing loads. The belt pull must be multiplied by a factor, which is dependent on the type of belt, its preload, belt tension and any additional dynamic forces. Belt manufacturers usually publish values. However, should information not be available, the following values can be used for

- 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 when the distance between shafts is short, for heavy or shock-type duty, or where belt tension is high.

Equivalent dynamic bearing load

If the calculated bearing load F obtained when using the above information is found to fulfil the requirements for the basic dynamic load rating C, i.e. the load is constant in magnitude and direction and acts radially on a radial bearing or axially and centrically on a thrust bearing, then P = F and the load may be inserted directly in the life equations.

In all other cases it is first necessary to calculate the equivalent dynamic bearing load. This is defined as that hypothetical load, constant in magnitude and direction, acting radially on radial bearings or axially and centrically on a thrust bearing which, if applied, would have the same influence on bearing life as the actual loads to which the bearing is subjected (➔ **fig 2**).

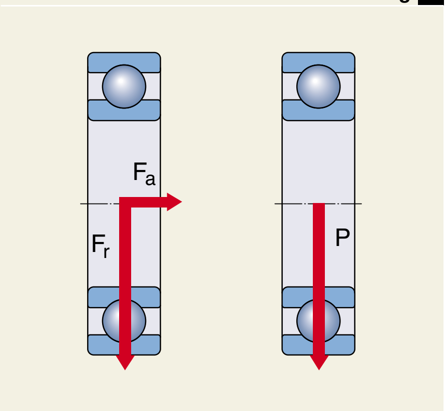
Radial bearings are often subjected to simultaneously acting radial and axial loads. If the resultant load is constant in magnitude and direction, the equivalent dynamic bearing load P can be obtained from the general equation

$$P = X\,F_r + Y\,F_a$$

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

Fig 2



An additional 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 are generally significant.

The same general equation also applies to spherical roller thrust bearings, which can accommodate both axial and radial loads. For thrust bearings which can accommodate only purely axial loads, e.g. thrust ball bearings and cylindrical roller thrust bearings, the equation can be simplified, provided the load acts centrically, to

$$P = F_a$$

All information and data required for calculating the equivalent dynamic bearing load will be found in the introductory text to each product section and in the product tables.

Fluctuating bearing load

In many cases the magnitude of the load fluctuates. The formula for life calculation with variable operating conditions should be applied (➔ **page 70**).

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 e.g. 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 13**), the mean load can be obtained from

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

Diagram 13

Load averaging

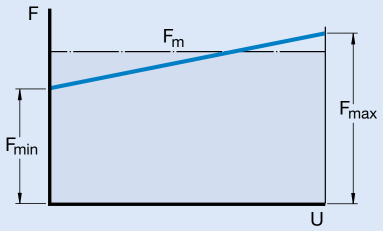


Diagram 14

Rotating load

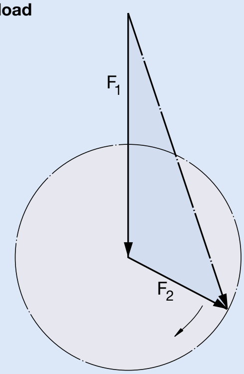
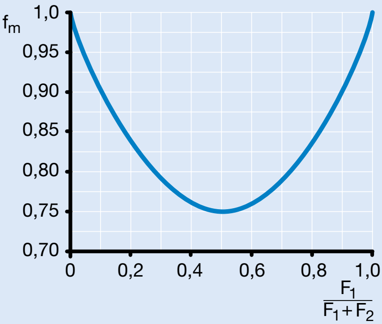


Diagram 15



Rotating load

If, as illustrated in **diagram** 14, the load on the bearing consists of a load F_1 which is constant in magnitude and direction (e.g. the weight of a rotor) and a rotating constant load F_2 (e.g. an unbalance load), the mean load can be obtained from

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

Values for the factor f_m can be obtained from **diagram** 15.

Requisite minimum load

The correlation between load and service life is less evident at very light loads. Other failure mechanisms than fatigue are determining.

In order to provide satisfactory operation, roller bearings must always be subjected to a given minimum load. A general "rule of thumb" indicates that minimum loads corresponding to 0,02 C should be imposed on roller bearings and minimum loads corresponding to 0,01 C on ball bearings. The importance of applying a minimum load increases where accelerations in the bearing are high, and where speeds are in the region of 50 % or more of the limiting speeds quoted in the product tables (➔ section "Speeds and vibration", starting on **page 107**). If minimum load requirements cannot be met, NoWear bearings could be considered (➔ **page 939**).

Recommendations for calculating the requisite minimum loads for the different bearing types are given in the text preceding each table section.

Selecting bearing size using the static load carrying capacity

Bearing size should be selected on the basis of static load ratings C_0 instead of on bearing life when one of the following conditions exist:

- the bearing is stationary and is subjected to continuous or intermittent (shock) loads;
- the bearing makes slow oscillating or alignment movements under load;
- the bearing rotates under load at very slow speed ($n < 10$ r/min) and is only required to have a short life (the life equation in this case, for a given equivalent load P would give such a low requisite basic dynamic load rating C, that the bearing selected on a life basis would be seriously overloaded in service);
- the bearing rotates and, in addition to the normal operating loads, has to sustain heavy shock loads

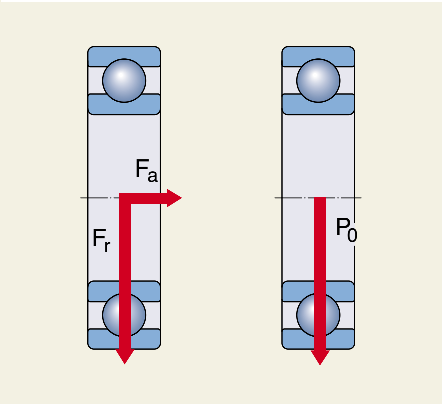
In all these cases, the permissible load for a bearing is determined not by material fatigue but by the amount of permanent deformation to the raceway caused by the load. Loads acting on a stationary bearing, or one which is slowly oscillating, as well as shock loads on a rotating bearing can produce flattened areas on the rolling elements and indentations in the raceways. The indentations may be irregularly spaced around the raceway, or may be evenly spaced at positions corresponding to the spacing of the rolling elements. If the load acts for several revolutions the deformation will be evenly distributed over the whole raceway. Permanent deformations in the bearing can lead to vibration in the bearing, noisy operation and increased friction. It is also possible that the internal clearance will increase or the character of the fits may be changed.

The extent to which these changes are detrimental to bearing performance depends on the demands placed on the bearing in a particular application. It is therefore necessary to make sure that permanent deformations do not occur, or occur to a very limited extent only, by selecting a bearing with sufficiently high static load carrying capacity, if one of the following demands has to be satisfied:

- high reliability,
- quiet running (e.g. for electric motors),
- vibration-free operation (e.g. for machine tools),
- constant bearing friction torque (e.g. for measuring apparatus and test equipment),
- low starting friction under load (e.g. for cranes).

When determining bearing size based on the static load carrying capacity a given safety factor s_0 which represents the relationship between the basic static load rating C_0 and the equivalent static bearing load P_0 is used to calculate the requisite basic static load rating.

Fig 3



Equivalent static bearing load

Static loads comprising radial and axial components must be converted into an equivalent static bearing load. This is defined as the load (radial for radial bearings and axial for thrust bearings) which, if applied, would cause the same maximum rolling element load in the bearing as the actual load. It is obtained from the general equation

$$P_0 = X_0\,F_r + Y_0\,F_a$$

where
P_0 = equivalent static bearing load, kN
F_r = actual radial bearing load (see below), kN
F_a = actual axial bearing load (see below), kN
X_0 = radial load factor for the bearing
Y_0 = axial load factor for the bearing

Note:
When calculating P_0, the maximum load that can occur should be used and its radial and axial components (➔ **fig** 3) inserted in the equation above. If a static load acts in different directions on a bearing, the magnitude of these components will change. In these cases, the components of the load giving the largest value of the equivalent static bearing load P_0 should be used. Information and data necessary for the calculation of the equivalent static bearing load will be found in the introductory text to each product section and in the tables.

Required basic static load rating

The required basic static load rating C_0 can be determined from

$$C_0 = s_0\,P_0$$

where
C_0 = basic static load rating, kN
P_0 = equivalent static bearing load, kN
s_0 = static safety factor

Guideline values based on experience are given in **table** 10 for the static safety factor s_0 for ball and roller bearings for various applications requiring smooth running. At elevated temperatures the static load carrying capacity is reduced. Further information will be supplied on request.

Checking the static load carrying capacity

For dynamically loaded bearings it is advisable, where the equivalent static bearing load P_0 is known, to check that the static load carrying capacity is adequate using

$$s_0 = C_0/P_0$$

If the s_0 value obtained is less than the recommended guideline value (➔ **table** 10), a bearing with a higher basic static load rating should be selected.

Table 10

Guideline values for the static safety factor s_0

Type of operation	**Rotating bearing** Requirements regarding quiet running: unimportant		normal		high		**Non-rotating bearing**	
	Ball bearings	Roller bearings	Ball bearings	Roller bearings	Ball bearings	Roller bearings	Ball bearings	Roller bearings
Smooth, vibration-free	0,5	1	1	1,5	2	3	0,4	0,8
Normal	0,5	1	1	1,5	2	3,5	0,5	1
Pronounced shock loads[1]	≥ 1,5	≥ 2,5	≥ 1,5	≥ 3	≥ 2	≥ 4	≥ 1	≥ 2

For spherical roller thrust bearings it is advisable to use $s_0 \geq 4$

[1] Where the magnitude of the load is not known, values of s_0 at least as large as those quoted above should be used. If the magnitude of the shock loads is exactly known, smaller values of s_0 can be applied

Calculation examples

Example 1
An SKF Explorer 6309 deep groove ball bearing is to operate at 3 000 r/min under a constant radial load F_r = 10 kN. Oil lubrication is to be used, the oil having an actual kinematic viscosity ν = 20 mm²/s at normal operating temperature. The desired reliability is 90 % and it is assumed that the operating conditions are very clean. What will be the basic and SKF rating lives?

a) The basic rating life for 90 % reliability is

$$L_{10} = \left(\frac{C}{P}\right)^3$$

- From the product table for bearing 6309, C = 55,3 kN. Since the load is purely radial, $P = F_r$ = 10 kN (➔ "Equivalent dynamic bearing load" on **page 74**).

$$L_{10} = (55{,}3/10)^3$$

$$= 169 \text{ millions of revolutions}$$

- or in operating hours, using

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

$$L_{10h} = 1\,000\,000/(60 \times 3\,000) \times 169$$

$$= 940 \text{ operating hours}$$

b) The SKF rating life for 90 % reliability is

$$L_{10m} = a_1\,a_{SKF}\,L_{10}$$

- As a reliability of 90 % is required, the L_{10m} life is to be calculated and a_1 = 1 (➔ **table 1**, **page 53**).
- From the product table for bearing 6309, d_m = 0,5 (d + D) = 0,5 (45 + 100) = 72,5 mm
- From **diagram 5**, **page 60**, the rated oil viscosity at operating temperature for a speed of 3 000 r/min, ν_1 = 8,15 mm²/s. Therefore $\kappa = \nu/\nu_1$ = 20/8,15 = 2,45
- Again from the product table P_u = 1,34 kN and P_u/P = 1,34/10 = 0,134. As the conditions are very clean, η_c = 0,8 and $\eta_c\,P_u/P$ = 0,107. With κ = 2,45 and using the SKF Explorer scale of **diagram 1**, **page 54**, the value of a_{SKF} = 8 is obtained. Then according to the SKF rating life equation

$$L_{10m} = 1 \times 8 \times 169$$

$$= 1\,352 \text{ millions of revolutions}$$

or in operating hours using

$$L_{10mh} = \frac{10^6}{60\,n}\,L_{10m}$$

$$L_{10mh} = 1\,000\,000/(60 \times 3\,000) \times 1\,352$$

$$= 7\,512 \text{ operating hours}$$

Example 2
The SKF Explorer 6309 deep groove ball bearing in example 1 belongs to an existing application that was calculated some years ago using the adjustment factor a_{23}. This application fully satisfied the requirements. It is requested to recalculate the life of this bearing in terms of the adjustment factor a_{23} and also of the factor a_{SKF} (based on the field experience of this application), i.e. $a_{SKF} = a_{23}$. Finally it is requested to obtain the equivalent factor η_c for the contamination level in the application under the condition $a_{SKF} = a_{23}$.

- With κ = 2,45, using the κ scale superimposed on the κ curves for the SKF life modification factor a_{SKF} of **diagram 1** on **page 54**, factor $a_{23} \approx 1{,}8$ which can be read on the a_{SKF} axis. Taking into account that this application fully satisfied the requirement, it can be safely assumed that $a_{SKF} = a_{23}$, thus

$$L_{10mh} = a_{23}\,L_{10h} = a_{SKF}\,L_{10h}$$

and

$$L_{10mh} = 1{,}8 \times 940 = 1\,690 \text{ operating hours}$$

- The factor η_c corresponding to this life adjustment is according to **table** 6 on **page 68** and for an SKF Explorer 6309 bearing with $P_u/P = 0,134$

  $\eta_c = [\eta_c\ (P_u/P)]_{23}\ /(P_u/P) = 0,04/0,134 = 0,3$

Example 3
An existing application has to be reviewed. A 6309-2RS1 deep groove ball bearing with integral seals and grease filling, is working under the same conditions as described in example 2 ($\kappa = 2,45$). The contamination conditions of this application have to be checked to determine if it is possible to reduce the costs for a minimum requisite life of 3 000 hours of operation.

- Considering grease lubrication and integral seals the level of contamination can be characterized as high cleanliness and from **table** 4 on **page 62**, $\eta_c = 0,8$. With $P_u/P = 0,134$, $\eta_c\ (P_u/P) = 0,107$ and $\kappa = 2,45$ from the SKF Explorer scale in **diagram** 1 on **page 54**, $a_{SKF} = 8$.

  $L_{10mh} = 8 \times 940 = 7\,520$ operating hours

- For a lower cost version – if possible – of the same bearing arrangement an SKF Explorer 6309-2Z bearing with shields is chosen. The contamination level can be characterized as normal cleanliness, then from **table** 4 on **page 62**, $\eta_c = 0,5$. With $P_u/P = 0,134$, $\eta_c\ (P_u/P) = 0,067$ and $\kappa = 2,45$ from the SKF Explorer scale in **diagram** 1 on **page 54**, $a_{SKF} \approx 3,5$.

  $L_{10mh} = 3,5 \times 940 = 3\,290$ operating hours

Conclusion: If possible, this application would be able to take advantage of a more cost effective solution by replacing the sealed bearing with a shielded one.

Note that the use of the rating life based on the a_{23} adjustment factor would not allow this design evaluation. Furthermore it would not be possible to reach the requisite life (➔ example 2, calculated life with the a_{23} adjustment factor would only give 1 690 operating hours).

Example 4
The 6309 deep groove ball bearing used in example 1 belongs to an existing application that was calculated some years ago using the adjustment factor a_{23}. From the field, there have been complaints of bearing failures. It is required to evaluate the design of this bearing application to determine suitable steps to increase its reliability.

- First the life is determined based on the a_{23} factor. With $\kappa = 2,45$, using the κ scale superimposed on the κ curves for the SKF life modification factor a_{SKF} in **diagram** 1 on **page 54**, $a_{23} = 1,8$ which can be read on the a_{SKF} axis.

  $L_{10mh} = a_{23} \times L_{10h} = 1,8 \times 940$

  $= 1\,690$ operating hours

- The factor η_c corresponding to this life adjustment factor a_{23} is according to **table** 4 on **page 62** and for $P_u/P = 0,134$

  $\eta_c = [\eta_c\ (P_u/P)]_{23}\ /(P_u/P) = 0,04/0,134 = 0,3$

- A microscope counting of an oil sample taken from the application indicated a contamination classification of –/17/14 according to ISO 4406:1999. The contamination consisted mainly of wear particles originated in the system. This can be characterized as "typical contamination", thus from **table** 4 on **page 62** and also from **diagram** 9 on **page 66**, $\eta_c = 0,2$. With $P_u/P = 0,134$, $\eta_c\ (P_u/P) = 0,0268$ and $\kappa = 2,45$ from **diagram** 1 on **page 54**, $a_{SKF} \approx 1,2$.

  $L_{10mh} = 1,2 \times 940 = 1\,130$ operating hours

- By using the SKF Explorer 6309-2RS1 bearing with integral contact seals, the level of contamination can be reduced to the level of "high cleanliness". Then from **table** 4 on **page 62**, $\eta_c = 0,8$. With $P_u/P = 0,134$, $\eta_c\ (P_u/P) = 0,107$ and $\kappa = 2,45$ from the SKF Explorer scale in **diagram** 1 on **page 54**, $a_{SKF} = 8$.

  $L_{10mh} = 8 \times 940 = 7\,520$ operating hours

Conclusion: This application has a level of contamination that is more severe than the factor $\eta_c = 0,3$ for the contamination level implicit when using the factor a_{23} while the real operating conditions, which are typical for contaminated industrial transmissions, call for a factor $\eta_c = 0,2$ when using the factor a_{SKF}.

This may explain the cause of the failures that were experienced with this application. The use of an SKF Explorer 6309-2RS1 bearing with integral contact seals will increase the reliability considerably and solve this problem.

Example 5

The duty cycle of a sealed SKF Explorer spherical roller bearing 24026-2CS2/VT143 used in heavy transportation equipment of a steel plant has the operating conditions listed in the table below.

The static load of this application is determined reasonably accurately, taking into account the inertia of the load during the loading operation and the occurrence of shock loads for accidental load dropping.

It is required to verify the dynamic and static load conditions of this application assuming a required operating life of 60 000 hours and a minimum static safety factor of 1,5.

- From the product table and introductory text:

  Load ratings:
  $C = 540$ kN; $C_0 = 815$ kN; $P_u = 81,5$ kN

  Dimensions: $d = 130$ mm; $D = 200$ mm, thus $d_m = 0,5\,(130 + 200) = 165$ mm

  Grease filling: Extreme pressure mineral oil based grease using a lithium soap of NLGI consistency class 2, with permissible temperature range between –20 and +110 °C and a base oil viscosity at 40 and 100 °C of 200 and 16 mm^2/s, respectively.

Example 5/1

Operating conditions

Duty interval	Equivalent dynamic load	Time Interval	Speed	Temperature	Equivalent static load
–	kN	–	r/min	°C	kN
1	200	0,05	50	50	500
2	125	0,40	300	65	500
3	75	0,45	400	65	500
4	50	0,10	200	60	500

- The following calculations are made or values determined:

  1. ν_1 = rated viscosity, mm^2/s (➔ **diagram** 5 on **page 60**) – input: d_m and speed

  2. ν = actual operating viscosity, mm^2/s (➔ **diagram** 6 on **page 61**) – input: lubricant viscosity at 40 °C and operating temperature

  3. κ = viscosity ratio – calculated (ν/ν_1)

  4. η_c = factor for contamination level (➔ **table** 4 on **page 62**) – "High cleanliness", sealed bearing: 0,8

  5. L_{10h} = basic rating life according to the equation listed on **page 52** – input: C, P and n

  6. a_{SKF} = from **diagram** 2 on **page 55** – input: SKF Explorer bearing, η_c, P_u, P and κ

  7. L_{10mh} = SKF rating life according to the equation listed on **page 53** – input a_{SKF} and L_{10h}

  8. L_{10mh} = SKF rating life according to the equation listed on **page 70** – input L_{10mh1}, L_{10mh2} and U_1, U_2, …

The SKF rating life of 84 300 hours is longer than the required service life, thus the dynamic load conditions of the bearing are verified.

Finally the static safety factor of this application is examined.

$$s_0 = \frac{C_0}{P_0} = \frac{815}{500} = 1{,}63$$

$s_0 = 1{,}63 > s_{0\,req}$

The above shows that the static safety of this application is verified. As the static load is determined accurately, the relatively small margin between the calculated and recommended static safety is of no concern.

Example 5/2

Calculation values

Duty interval	Equivalent dynamic load	Required viscosity ν_1 ~	Operating viscosity ν ~	κ[1] ~	η_c ~	Basic rating life L_{10h}	a_{SKF}	SKF rating life L_{10mh}	Time fraction	Resulting SKF rating life L_{10mh}
–	kN	mm^2/s	mm^2/s	–	–	h	–	h	–	h
1	200	120	120	1	0,8	9 136	1,2	11 050	0,05	84 300
2	125	25	60	2,3	0,8	7 295	7,8	57 260	0,40	
3	75	20	60	3	0,8	30 030	43	1 318 000	0,45	
4	50	36	75	2	0,8	232 040	50	11 600 000	0,10	

[1] Grease with EP additives

SKF calculation tools

SKF possesses one of the most comprehensive and powerful sets of modelling and simulation packages in the bearing industry. They range from easy-to-use tools based on SKF General Catalogue formulae to the most sophisticated calculation and simulation systems, running on parallel computers.

The company's philosophy is to develop a range of programs to satisfy a number of customer requirements; from fairly simple design checks, through moderately complex investigations, to the most advanced simulations for bearing and machine design. Wherever possible these programs are available for in-the-field use on customers' or SKF engineers' laptops, desk top PCs or workstations. Moreover, particular care is taken to provide integration and interoperability of the different systems with each other.

Interactive Engineering Catalogue

The Interactive Engineering Catalogue (IEC) is an easy-to-use tool for bearing selection and calculation. Bearing searches are available based on designation or dimensions, and simple bearing arrangements can be evaluated as well. The equations used are in line with this SKF General Catalogue.

It also allows the generation of CAD bearing drawings that can be imported into customer application drawings developed with the major CAD commercial packages.

The Interactive Engineering Catalogue also contains in addition to the complete range of rolling bearings catalogues covering bearing units, bearing housings, plain bearings and seals.

The SKF Interactive Engineering Catalogue is published on CD-ROM or on the Internet at www.skf.com.

SKF Toolbox

The SKF Toolbox is a set of engineering calculation programs accessible through the SKF website, www.skf.com. It contains several calculation tools and uses theory from this General Catalogue and some basic mechanical engineering equations. For instance, it can calculate bearing clearance reduction due to interference fit and operating temperature, contamination factors or data for mounting of specific bearing arrangements.

Ginger

Ginger is the new mainstream bearing application program used by SKF engineers to find the best solution for customers' bearing arrangements. Ginger, currently in preparation, is the successor of Beacon and its technology allows the modelling in a 3D graphic environment of flexible systems incorporating customer components. Ginger combines the ability to model generic mechanical systems (using also shafts, gears, housings etc.) with a precise bearing model for an in-depth analysis of the system behaviour in a virtual environment. It also performs bearing rolling fatigue evaluation using the SKF rating life in particular. Ginger is the result of several years of specific research and development within SKF.

Orpheus

The numerical tool Orpheus allows studying and optimizing the dynamic behaviour of noise and vibration in critical bearing applications (e.g. electric motors, gearboxes). It can be used to solve the complete non-linear equations of motion of a set of bearings and their surrounding components, including gears, shafts and housings.

It can provide profound understanding of and advice on the dynamic behaviour of an application, including the bearings, accounting for form deviations (waviness) and mounting errors (misalignment). This enables SKF engineers to determine the most suitable bearing type and size as well as the corresponding mounting and preload conditions for a given application.

Beast

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

Other programs

In addition to the above-mentioned programs, SKF has developed dedicated computer programs that enable SKF scientists to provide customers with bearings having an optimized bearing surface finish to extend bearing life under severe operating conditions. These programs can calculate the lubricant film thickness in elasto-hydrodynamically lubricated contacts. In addition, the local film thickness resulting from the deformation of the three dimensional surface topography inside such contacts is calculated in detail and the consequent reduction of bearing fatigue life.

In order to complete the necessary capabilities for their tasks, SKF engineers use commercial packages to perform e.g. finite element or generic system dynamics analyses. These tools are integrated with the SKF proprietary systems allowing a faster and more robust connection with customer data and models.

SKF Engineering Consultancy Services

The basic information required to calculate and design a bearing arrangement can be found in this catalogue. But there are applications where it is desirable to be able to predict the expected bearing life as accurately as possible either because sufficient experience with similar bearing arrangements is lacking or because economy and/or operational reliability are of extreme importance. In such cases for example it is advisable to consult the "SKF Engineering Consultancy Services". They provide calculations and simulations utilizing high-tech computer programs, in combination with an almost one hundred year global experience in the field of rotating machine components.

They can provide support with the complete SKF application know-how. The SKF application specialists will

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

SKF Engineering Consultancy Services provides a new approach to services concerning machines and installations for OEM and end users. Some of these service benefits are:

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

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

- analytical modelling of complete bearing arrangements, consisting of shaft, housing, gears, couplings, etc.,

- static analysis, i.e. determination of elastic deformations and stresses in components of mechanical systems,
- dynamic analysis, i.e. determination of the vibration behaviour of systems under working conditions ("virtual testing"),
- visual and animated presentation of structural and component deflection,
- optimizing system costs, service life, vibration and noise level.

The High-Tech computer programs used within the SKF Engineering Consultancy Services as standard for calculation and simulations are briefly described in the section "SKF calculation tools".

For further information regarding the activities of the SKF Engineering Consultancy Services please contact the nearest SKF company.

SKF life testing

SKF endurance testing activities are concentrated at the SKF Engineering & Research Centre in the Netherlands. The test facilities there are unique in the bearing industry as regards sophistication and number of test rigs. The centre also supports work carried out at the research facilities of the major SKF manufacturing companies.

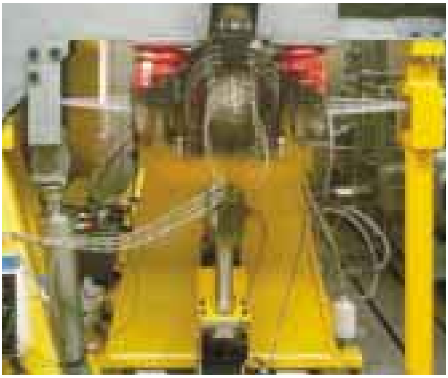
SKF undertakes life testing, mainly to be able to continuously improve its products. It is essential to understand and to formulate the fundamental physical laws governing bearing behaviour as a function of internal and external variables. Such variables may represent material properties, internal bearing geometry and conformity, cage design, misalignment, temperature and other operating conditions. However, many influencing factors are not of static but rather of dynamic nature. Examples are the topography of working contact surfaces, the material structure, the internal geometry and the lubricant properties, which continuously undergo change during the bearing operation.

SKF also undertakes life testing to

- assure the performance commitments made in product catalogues,
- audit the quality of the SKF standard bearing production,
- research the influences of lubricants and lubricating conditions on bearing life,
- support the development of theories for rolling contact fatigue,
- compare with competitive products.

The powerful and firmly controlled life testing procedure combined with post-test investigations with modern and highly sophisticated equipment makes it possible to investigate the factors and their interactions in a systematic way.

The SKF Explorer bearings are an example of the implementation of the optimized influencing factors on the basis of analytical simulation models and experimental verification at component and full bearing level.




M = φish φrs Mrr
+ Msl + Mseal
+ Mdrag

Friction

The friction in a rolling bearing is the determining factor where heat generation in the bearing is concerned and consequently for the operating temperature.

The amount of friction depends on the load and on several other factors, the most important of which are the bearing type and size, the operating speed, the properties of the lubricant and the quantity of lubricant.

The total resistance to rotation of a bearing is made up of the rolling and sliding friction in the rolling contacts, in the contact areas between rolling elements and cage, as well as in the guiding surfaces for the rolling elements or the cage, the friction in the lubricant and the sliding friction of contact seals when applicable.

Estimation of the frictional moment

Under certain conditions:

- bearing load $P \approx 0{,}1\,C$,
- good lubrication and
- normal operating conditions

the frictional moment can be calculated with sufficient accuracy using a constant coefficient of friction μ from the following equation

$$M = 0{,}5\,\mu\,P\,d$$

where

M = frictional moment, Nmm

μ = constant coefficient of friction for the bearing (➔ **table 1**)

P = equivalent dynamic bearing load, N

d = bearing bore diameter, mm

More accurate calculation of the frictional moment

One approach to calculate the frictional moment of a rolling bearing is to divide the frictional moment into a so-called load independent moment M_0 and a load dependent moment M_1 and add them together later, giving

$$M = M_0 + M_1$$

This has been the approach until now. However, more accurate methods are available if the division is based on the type of friction source rather than on its dependency on load. In fact, M_0 accounts for the additional external sources of friction, together with the "hydrodynamic" component of rolling friction, which also has a load dependent part.

Table 1

Constant coefficient of friction μ for unsealed bearings

Bearing type	**Coefficient of friction μ**
Deep groove ball bearings	0,0015
Angular contact ball bearings	
– single row	0,0020
– double row	0,0024
– four-point contact	0,0024
Self-aligning ball bearings	0,0010
Cylindrical roller bearings	
– with cage, when $F_a \approx 0$	0,0011
– full complement, when $F_a \approx 0$	0,0020
Taper roller bearings	0,0018
Spherical roller bearings	0,0018
CARB toroidal roller bearings	0,0016
Thrust ball bearings	0,0013
Cylindrical roller thrust bearings	0,0050
Spherical roller thrust bearings	0,0018

The new SKF model for calculation of the frictional moment

The new SKF model for calculating the frictional moment allows a more accurate calculation of the frictional moment generated in SKF rolling bearings according to the above-mentioned equation

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

The new SKF model is derived from more advanced computational models developed by SKF and is designed to provide approximate reference values, under the following application conditions:

- grease lubrication or normal methods of oil lubrication: oil bath, oil-spot and oil jet;
- for paired bearings, calculate the frictional moment for each bearing separately and then add them up. The radial load is divided equally over the two bearings; the axial load is shared according to the bearing arrangement;
- loads equal to or larger than the recommended minimum load;
- constant loads in magnitude and direction;
- normal operational clearance.

To accurately calculate the frictional moment in a rolling bearing, four different sources must be taken into account

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

where

M = total frictional moment, Nmm
M_{rr} = rolling frictional moment, Nmm
M_{sl} = sliding frictional moment, Nmm
M_{seal} = frictional moment of the seals, Nmm
M_{drag} = frictional moment of drag losses, churning, splashing etc, Nmm

This new approach identifies the sources of friction in every contact occurring in the bearing and combines them; in addition the seal contribution and additional external sources can be added as required to predict the overall frictional moment. Since the model looks into every single contact (raceways and flanges), changes of design and improvements of the surfaces can readily be taken into consideration, making the model more able to reflect improvements in SKF bearing designs and easier to update.

In the following sections the new SKF model for calculating frictional moments starts with the simplest form of the rolling, sliding and seal contributions. In the next section the effects of the oil level in the bearing, high-speed starvation, inlet shear heating and mixed lubrication will be described.

Note:
The formulae given here lead to rather complex calculations. It is therefore strongly advised to make use of the calculation facilities provided in the "SKF Interactive Engineering Catalogue" available on CD-ROM or online at www.skf.com.

Rolling frictional moment
The rolling frictional moment is calculated from the equation

$$M_{rr} = G_{rr} (\nu\, n)^{0,6}$$

where
M_{rr} = rolling frictional moment, Nmm
G_{rr} = a variable that depends on
- the bearing type,
- the bearing mean diameter, mm $d_m = 0,5\,(d + D)$
- the radial load F_r, N
- the axial load F_a, N

n = rotational speed, r/min
ν = kinematic viscosity of the lubricant at the operating temperature, mm^2/s (for grease lubrication the base oil viscosity)

The values for G_{rr} can be obtained using the equations given in **table 2** and the geometry constants R given in **table 3**, starting on **page 92**. Both loads, F_r and F_a, are always considered as positive.

Sliding frictional moment
The sliding frictional moment is calculated using

$$M_{sl} = G_{sl}\, \mu_{sl}$$

where
M_{sl} = sliding frictional moment, Nmm
G_{sl} = a variable that depends on
- the bearing type,
- the bearing mean diameter $d_m = 0,5\,(d + D)$, mm
- the radial load F_r, N
- the axial load F_a, N

μ_{sl} = sliding friction coefficient, which can be set to the value for full film conditions, i.e. $\kappa \geq 2$,
0,05 for lubrication with mineral oils
0,04 for lubrication with synthetic oils
0,1 for lubrication with transmission fluids
For cylindrical or taper roller bearings use the following values instead:
0,02 for cylindrical roller bearings
0,002 for taper roller bearings

The values for G_{sl} can be obtained using the equations given in **table 2** and the geometry constants S given in **table 3**, starting on **page 92**.

Frictional moment of seals
Where bearings are fitted with contact seals the frictional losses arising from the seal may exceed those generated in the bearing. The frictional moment of seals for bearings that are sealed at both sides can be estimated using the following empirical equation

$$M_{seal} = K_{S1}\, d_S^{\beta} + K_{S2}$$

where
M_{seal} = frictional moment of seals, Nmm
K_{S1} = constant depending on the bearing type
K_{S2} = constant depending on bearing and seal type
d_S = shoulder diameter listed in product tables (➔ **table 4**, **page 96**)
β = exponent depending on bearing and seal type

Values for constants K_{S1}, and K_{S2} and exponent β can be found in **table 4**, **page 96.**

M_{seal} is the frictional moment generated by two seals. In case there is one seal only, the friction generated is 0,5 M_{seal}.

For RSL seals for deep groove ball bearings with an outside diameter over 25 mm, use the calculated value of M_{seal} irrespective whether there is one or two seals.

Table 2a

Geometry and load dependent variables for rolling and sliding frictional moments – radial bearings

Bearing type	Rolling friction variables G_{rr}	Sliding friction variables G_{sl}
Deep groove ball bearings	when $F_a = 0$ $G_{rr} = R_1\, d_m^{1,96}\, F_r^{0,54}$ when $F_a > 0$ $G_{rr} = R_1\, d_m^{1,96} \left(F_r + \frac{R_2}{\sin \alpha_F} F_a\right)^{0,54}$ $\alpha_F = 24,6\, (F_a/C_0)^{0,24}$, degrees	when $F_a = 0$ $G_{sl} = S_1\, d_m^{-0,26}\, F_r^{5/3}$ when $F_a > 0$ $G_{sl} = S_1\, d_m^{-0,145} \left(F_r^5 + \frac{S_2\, d_m^{1,5}}{\sin \alpha_F} F_a^4\right)^{1/3}$
Angular contact ball bearings[1]	$G_{rr} = R_1\, d_m^{1,97} \left[F_r + F_g + R_2\, F_a\right]^{0,54}$ $F_g = R_3\, d_m^4\, n^2$	$G_{sl} = S_1\, d_m^{0,26} \left[(F_r + F_g)^{4/3} + S_2\, F_a^{4/3}\right]$ $F_g = S_3\, d_m^4\, n^2$
Four-point contact ball bearings	$G_{rr} = R_1\, d_m^{1,97} \left[F_r + F_g + R_2\, F_a\right]^{0,54}$ $F_g = R_3\, d_m^4\, n^2$	$G_{sl} = S_1\, d_m^{0,26} \left[(F_r + F_g)^{4/3} + S_2\, F_a^{4/3}\right]$ $F_g = S_3\, d_m^4\, n^2$
Self-aligning ball bearings	$G_{rr} = R_1\, d_m^2 \left[F_r + F_g + R_2\, F_a\right]^{0,54}$ $F_g = R_3\, d_m^{3,5}\, n^2$	$G_{sl} = S_1\, d_m^{-0,12} \left[(F_r + F_g)^{4/3} + S_2\, F_a^{4/3}\right]$ $F_g = S_3\, d_m^{3,5}\, n^2$
Cylindrical roller bearings	$G_{rr} = R_1\, d_m^{2,41}\, F_r^{0,31}$	$G_{sl} = S_1\, d_m^{0,9}\, F_a + S_2\, d_m\, F_r$
Taper roller bearings[1] For axial load factor Y for single row bearings ➔ product tables	$G_{rr} = R_1\, d_m^{2,38}\, (F_r + R_2\, Y\, F_a)^{0,31}$	$G_{sl} = S_1\, d_m^{0,82}\, (F_r + S_2\, Y\, F_a)$
Spherical roller bearings	$G_{rr.e} = R_1\, d_m^{1,85}\, (F_r + R_2\, F_a)^{0,54}$ $G_{rr.l} = R_3\, d_m^{2,3}\, (F_r + R_4\, F_a)^{0,31}$ when $G_{rr.e} < G_{rr.l}$ $G_{rr} = G_{rr.e}$ otherwise $G_{rr} = G_{rr.l}$	$G_{sl.e} = S_1\, d_m^{0,25} \left(F_r^4 + S_2\, F_a^4\right)^{1/3}$ $G_{sl.l} = S_3\, d_m^{0,94} \left(F_r^3 + S_4\, F_a^3\right)^{1/3}$ when $G_{sl.e} < G_{sl.l}$ $G_{sl} = G_{sl.e}$ otherwise $G_{sl} = G_{sl.l}$
CARB toroidal roller bearings	when $F_r < (R_2^{1,85}\, d_m^{0,78}/R_1^{1,85})^{2,35}$ $G_{rr.e} = R_1\, d_m^{1,97}\, F_r^{0,54}$ otherwise $G_{rr.l} = R_2\, d_m^{2,37}\, F_r^{0,31}$	when $F_r < (S_2\, d_m^{1,24}/S_1)^{1,5}$ $G_{sl.e} = S_1\, d_m^{-0,19}\, F_r^{5/3}$ otherwise $G_{sl.l} = S_2\, d_m^{1,05}\, F_r$

[1] The value to be used for F_a is the external axial load

Table 2b

Geometry and load dependent variables for rolling and sliding frictional moments – thrust bearings

Bearing type	Rolling friction variables G_{rr}	Sliding friction variables G_{sl}
Thrust ball bearings	$G_{rr} = R_1\, d_m^{1,83}\ F_a^{0,54}$	$G_{sl} = S_1\, d_m^{0,05}\ F_a^{4/3}$
Cylindrical roller thrust bearings	$G_{rr} = R_1\, d_m^{2,38}\ F_a^{0,31}$	$G_{sl} = S_1\, d_m^{0,62}\ F_a$
Spherical roller thrust bearings	$G_{rr.e} = R_1\, d_m^{1,96}\,(F_r + R_2\ F_a)^{0,54}$ $G_{rr.l} = R_3\, d_m^{2,39}\,(F_r + R_4\ F_a)^{0,31}$ when $G_{rr.e} < G_{rr.l}$ $G_{rr} = G_{rr.e}$ otherwise $G_{rr} = G_{rr.l}$	$G_{sl.e} = S_1\, d_m^{-0,35}\left(F_r^{5/3} + S_2\ F_a^{5/3}\right)$ $G_{sl.l} = S_3\, d_m^{0,89}\,(F_r + F_a)$ when $G_{sl.e} < G_{sl.l}$ $G_{sr} = G_{sl.e}$ otherwise $G_{sr} = G_{sl.l}$ $G_f = S_4\, d_m^{0,76}\,(F_r + S_5\ F_a)$ $G_{sl} = (G_{sr} + \phi_{bl}\ G_f)$

Table 3

Geometry constants for rolling and sliding frictional moments

Bearing type	Geometry constants for rolling frictional moments R_1	R_2	R_3	sliding frictional moments S_1	S_2	S_3
Deep groove ball bearings	See table 3a			See table 3a		
Angular contact ball bearings,						
– single row	$5,03 \times 10^{-7}$	1,97	$1,90 \times 10^{-12}$	$1,30 \times 10^{-2}$	0,68	$1,91 \times 10^{-12}$
– double row	$6,34 \times 10^{-7}$	1,41	$7,83 \times 10^{-13}$	$7,56 \times 10^{-3}$	1,21	$7,83 \times 10^{-13}$
– four-point contact	$4,78 \times 10^{-7}$	2,42	$1,40 \times 10^{-12}$	$1,20 \times 10^{-2}$	0,9	$1,40 \times 10^{-12}$
Self-aligning ball bearings	See table 3b			See table 3b		
Cylindrical roller bearings	See table 3c			See table 3c		
Taper roller bearings	See table 3d			See table 3d		
Spherical roller bearings	See table 3e			See table 3e		
CARB toroidal roller bearings	See table 3f			See table 3f		
Thrust ball bearings	$1,03 \times 10^{-6}$			$1,6 \times 10^{-2}$		
Cylindrical roller thrust bearings	$2,25 \times 10^{-6}$			0,154		
Spherical roller thrust bearings	See table 3g			See table 3g		

Table 3a

Geometry constants for rolling and sliding frictional moments of deep groove ball bearings

Bearing series	Geometry constants for rolling frictional moments R_1	R_2	sliding frictional moments S_1	S_2
2, 3	$4,4 \times 10^{-7}$	1,7	$2,00 \times 10^{-3}$	100
42, 43	$5,4 \times 10^{-7}$	0,96	$3,00 \times 10^{-3}$	40
60, 630	$4,1 \times 10^{-7}$	1,7	$3,73 \times 10^{-3}$	14,6
62, 622	$3,9 \times 10^{-7}$	1,7	$3,23 \times 10^{-3}$	36,5
63, 623	$3,7 \times 10^{-7}$	1,7	$2,84 \times 10^{-3}$	92,8
64	$3,6 \times 10^{-7}$	1,7	$2,43 \times 10^{-3}$	198
160, 161	$4,3 \times 10^{-7}$	1,7	$4,63 \times 10^{-3}$	4,25
617, 618, 628, 637, 638	$4,7 \times 10^{-7}$	1,7	$6,50 \times 10^{-3}$	0,78
619, 639	$4,3 \times 10^{-7}$	1,7	$4,75 \times 10^{-3}$	3,6

Table 3b

Geometry constants for rolling and sliding frictional moments of self-aligning ball bearings

Bearing series	Geometry constants for rolling frictional moments R_1	R_2	R_3	sliding frictional moments S_1	S_2	S_3
12	$3,25 \times 10^{-7}$	6,51	$2,43 \times 10^{-12}$	$4,36 \times 10^{-3}$	9,33	$2,43 \times 10^{-12}$
13	$3,11 \times 10^{-7}$	5,76	$3,52 \times 10^{-12}$	$5,76 \times 10^{-3}$	8,03	$3,52 \times 10^{-12}$
22	$3,13 \times 10^{-7}$	5,54	$3,12 \times 10^{-12}$	$5,84 \times 10^{-3}$	6,60	$3,12 \times 10^{-12}$
23	$3,11 \times 10^{-7}$	3,87	$5,41 \times 10^{-12}$	0,01	4,35	$5,41 \times 10^{-12}$
112	$3,25 \times 10^{-7}$	6,16	$2,48 \times 10^{-12}$	$4,33 \times 10^{-3}$	8,44	$2,48 \times 10^{-12}$
130	$2,39 \times 10^{-7}$	5,81	$1,10 \times 10^{-12}$	$7,25 \times 10^{-3}$	7,98	$1,10 \times 10^{-12}$
139	$2,44 \times 10^{-7}$	7,96	$5,63 \times 10^{-13}$	$4,51 \times 10^{-3}$	12,11	$5,63 \times 10^{-13}$

Table 3c

Geometry constants for rolling and sliding frictional moments of cylindrical roller bearings

Bearing series	Geometry constants for rolling frictional moments R_1	sliding frictional moments S_1	S_2
Bearing with cage of the N, NU, NJ or NUP design			
2, 3	$1,09 \times 10^{-6}$	0,16	0,0015
4	$1,00 \times 10^{-6}$	0,16	0,0015
10	$1,12 \times 10^{-6}$	0,17	0,0015
12	$1,23 \times 10^{-6}$	0,16	0,0015
20	$1,23 \times 10^{-6}$	0,16	0,0015
22	$1,40 \times 10^{-6}$	0,16	0,0015
23	$1,48 \times 10^{-6}$	0,16	0,0015
Full complement bearings of the NCF, NJG, NNC, NNCF, NNC and NNF design			
18, 28, 29, 30, 48, 49, 50	$2,13 \times 10^{-6}$	0,16	0,0015

Table 3d

Geometry constants for rolling and sliding frictional moments of taper roller bearings

Bearing series	Geometry constants for rolling frictional moments R_1	R_2	sliding frictional moments S_1	S_2
302	$1{,}76 \times 10^{-6}$	10,9	0,017	2
303	$1{,}69 \times 10^{-6}$	10,9	0,017	2
313	$1{,}84 \times 10^{-6}$	10,9	0,048	2
320 X	$2{,}38 \times 10^{-6}$	10,9	0,014	2
322	$2{,}27 \times 10^{-6}$	10,9	0,018	2
322 B	$2{,}38 \times 10^{-6}$	10,9	0,026	2
323	$2{,}38 \times 10^{-6}$	10,9	0,019	2
323 B	$2{,}79 \times 10^{-6}$	10,9	0,030	2
329	$2{,}31 \times 10^{-6}$	10,9	0,009	2
330	$2{,}71 \times 10^{-6}$	11,3	0,010	2
331	$2{,}71 \times 10^{-6}$	10,9	0,015	2
332	$2{,}71 \times 10^{-6}$	10,9	0,018	2
LL	$1{,}72 \times 10^{-6}$	10,9	0,0057	2
L	$2{,}19 \times 10^{-6}$	10,9	0,0093	2
LM	$2{,}25 \times 10^{-6}$	10,9	0,011	2
M	$2{,}48 \times 10^{-6}$	10,9	0,015	2
HM	$2{,}60 \times 10^{-6}$	10,9	0,020	2
H	$2{,}66 \times 10^{-6}$	10,9	0,025	2
HH	$2{,}51 \times 10^{-6}$	10,9	0,027	2
All other	$2{,}31 \times 10^{-6}$	10,9	0,019	2

Table 3e

Geometry constants for rolling and sliding frictional moments of spherical roller bearings

Bearing series	Geometry constants for rolling frictional moments R_1	R_2	R_3	R_4	sliding frictional moments S_1	S_2	S_3	S_4
213 E, 222 E	$1{,}6 \times 10^{-6}$	5,84	$2{,}81 \times 10^{-6}$	5,8	$3{,}62 \times 10^{-3}$	508	$8{,}8 \times 10^{-3}$	117
222	$2{,}0 \times 10^{-6}$	5,54	$2{,}92 \times 10^{-6}$	5,5	$5{,}10 \times 10^{-3}$	414	$9{,}7 \times 10^{-3}$	100
223	$1{,}7 \times 10^{-6}$	4,1	$3{,}13 \times 10^{-6}$	4,05	$6{,}92 \times 10^{-3}$	124	$1{,}7 \times 10^{-2}$	41
223 E	$1{,}6 \times 10^{-6}$	4,1	$3{,}14 \times 10^{-6}$	4,05	$6{,}23 \times 10^{-3}$	124	$1{,}7 \times 10^{-2}$	41
230	$2{,}4 \times 10^{-6}$	6,44	$3{,}76 \times 10^{-6}$	6,4	$4{,}13 \times 10^{-3}$	755	$1{,}1 \times 10^{-2}$	160
231	$2{,}4 \times 10^{-6}$	4,7	$4{,}04 \times 10^{-6}$	4,72	$6{,}70 \times 10^{-3}$	231	$1{,}7 \times 10^{-2}$	65
232	$2{,}3 \times 10^{-6}$	4,1	$4{,}00 \times 10^{-6}$	4,05	$8{,}66 \times 10^{-3}$	126	$2{,}1 \times 10^{-2}$	41
238	$3{,}1 \times 10^{-6}$	12,1	$3{,}82 \times 10^{-6}$	12	$1{,}74 \times 10^{-3}$	9 495	$5{,}9 \times 10^{-3}$	1 057
239	$2{,}7 \times 10^{-6}$	8,53	$3{,}87 \times 10^{-6}$	8,47	$2{,}77 \times 10^{-3}$	2 330	$8{,}5 \times 10^{-3}$	371
240	$2{,}9 \times 10^{-6}$	4,87	$4{,}78 \times 10^{-6}$	4,84	$6{,}95 \times 10^{-3}$	240	$2{,}1 \times 10^{-2}$	68
241	$2{,}6 \times 10^{-6}$	3,8	$4{,}79 \times 10^{-6}$	3,7	$1{,}00 \times 10^{-2}$	86,7	$2{,}9 \times 10^{-2}$	31
248	$3{,}8 \times 10^{-6}$	9,4	$5{,}09 \times 10^{-6}$	9,3	$2{,}80 \times 10^{-3}$	3 415	$1{,}2 \times 10^{-2}$	486
249	$3{,}0 \times 10^{-6}$	6,67	$5{,}09 \times 10^{-6}$	6,62	$3{,}90 \times 10^{-3}$	887	$1{,}7 \times 10^{-2}$	180

Table 3f

Geometry constants for rolling and sliding frictional moments of CARB toroidal roller bearings

Bearing series	Geometry constants for rolling frictional moments R_1	R_2	sliding frictional moments S_1	S_2
C 22	$1,17 \times 10^{-6}$	$2,08 \times 10^{-6}$	$1,32 \times 10^{-3}$	$0,8 \times 10^{-2}$
C 23	$1,20 \times 10^{-6}$	$2,28 \times 10^{-6}$	$1,24 \times 10^{-3}$	$0,9 \times 10^{-2}$
C 30	$1,40 \times 10^{-6}$	$2,59 \times 10^{-6}$	$1,58 \times 10^{-3}$	$1,0 \times 10^{-2}$
C 31	$1,37 \times 10^{-6}$	$2,77 \times 10^{-6}$	$1,30 \times 10^{-3}$	$1,1 \times 10^{-2}$
C 32	$1,33 \times 10^{-6}$	$2,63 \times 10^{-6}$	$1,31 \times 10^{-3}$	$1,1 \times 10^{-2}$
C 39	$1,45 \times 10^{-6}$	$2,55 \times 10^{-6}$	$1,84 \times 10^{-3}$	$1,0 \times 10^{-2}$
C 40	$1,53 \times 10^{-6}$	$3,15 \times 10^{-6}$	$1,50 \times 10^{-3}$	$1,3 \times 10^{-2}$
C 41	$1,49 \times 10^{-6}$	$3,11 \times 10^{-6}$	$1,32 \times 10^{-3}$	$1,3 \times 10^{-2}$
C 49	$1,49 \times 10^{-6}$	$3,24 \times 10^{-6}$	$1,39 \times 10^{-3}$	$1,5 \times 10^{-2}$
C 59	$1,77 \times 10^{-6}$	$3,81 \times 10^{-6}$	$1,80 \times 10^{-3}$	$1,8 \times 10^{-2}$
C 60	$1,83 \times 10^{-6}$	$5,22 \times 10^{-6}$	$1,17 \times 10^{-3}$	$2,8 \times 10^{-2}$
C 69	$1,85 \times 10^{-6}$	$4,53 \times 10^{-6}$	$1,61 \times 10^{-3}$	$2,3 \times 10^{-2}$

Table 3g

Geometry constants for rolling and sliding frictional moments of spherical roller thrust bearings

Bearing series	Geometry constants for rolling frictional moments R_1	R_2	R_3	R_4	sliding frictional moments S_1	S_2	S_3	S_4	S_5
292	$1,32 \times 10^{-6}$	1,57	$1,97 \times 10^{-6}$	3,21	$4,53 \times 10^{-3}$	0,26	0,02	0,1	0,6
292 E	$1,32 \times 10^{-6}$	1,65	$2,09 \times 10^{-6}$	2,92	$5,98 \times 10^{-3}$	0,23	0,03	0,17	0,56
293	$1,39 \times 10^{-6}$	1,66	$1,96 \times 10^{-6}$	3,23	$5,52 \times 10^{-3}$	0,25	0,02	0,1	0,6
293 E	$1,16 \times 10^{-6}$	1,64	$2,00 \times 10^{-6}$	3,04	$4,26 \times 10^{-3}$	0,23	0,025	0,15	0,58
294 E	$1,25 \times 10^{-6}$	1,67	$2,15 \times 10^{-6}$	2,86	$6,42 \times 10^{-3}$	0,21	0,04	0,2	0,54

Table 4

Seal frictional moment: Calculation exponent and constants

Seal type Bearing type	**Bearing outside diameter** D over	incl.	**Exponent and constants** β	K_{S1}	K_{S2}	**Seal counter-face diameter** d_s[1)]
RSL seals						
Deep groove ball bearings		25	0	0	0	d_2
	25	52	2,25	0,018	0	d_2
RZ seals						
Deep groove ball bearings		175	0	0	0	d_1
RSH seals						
Deep groove ball bearings		52	2,25	0,028	2	d_2
RS1 seals						
Deep groove ball bearings		62	2,25	0,023	2	d_1, d_2
	62	80	2,25	0,018	20	d_1, d_2
	80	100	2,25	0,018	15	d_1, d_2
	100		2,25	0,018	0	d_1, d_2
Angular contact ball bearings	30	120	2	0,014	10	d_1
Self-aligning ball bearings	30	125	2	0,014	10	d_2
LS seals						
Cylindrical roller bearings	42	360	2	0,032	50	E
CS, CS2 and CS5 seals						
Spherical roller bearings	62	300	2	0,057	50	d_2
CARB toroidal roller bearings	42	340	2	0,057	50	d_2

[1)] Designation of the dimension listed in the product table

Additional effects on frictional moments in bearings

In order to follow more closely the real behaviour of the bearing, and if an even more accurate calculation is needed, the new SKF model is able to consider additional effects which can be added into the equation. Those additional effects include:

- inlet shear heating reduction
- replenishment – starvation speed effects for oil-spot, oil jet, grease and low level oil bath lubrication
- drag loss effects in oil bath lubrication
- mixed lubrication for low speeds and/or low viscosities

Including these additional sources, the final equation for the total frictional moment of a bearing is

$$M = \phi_{ish}\, \phi_{rs}\, M_{rr} + M_{sl} + M_{seal} + M_{drag}$$

where

M = total frictional moment of bearing, Nmm
M_{rr} = $G_{rr}\, (\nu\, n)^{0,6}$
M_{sl} = $G_{sl}\, \mu_{sl}$
M_{seal} = $K_{S1}\, d_S^{\beta} + K_{S2}$
M_{drag} = frictional moment of drag losses, churning, splashing etc, Nmm
ϕ_{ish} = inlet shear heating reduction factor
ϕ_{rs} = kinematic replenishment/starvation reduction factor

The reduction factors ϕ_{ish} and ϕ_{rs} are introduced in the new SKF friction model to account for the effects of inlet shear heating reduction and high-speed replenishment/starvation of rolling friction, respectively. The sliding friction coefficient μ_{sl} increases for low speeds and/or viscosity to account for the mixed lubrication regime.

Inlet shear heating reduction factor

When sufficient lubricant is available in the bearing, not all of it can go through the contacts; only a tiny amount of lubricant is used to build-up the film thickness. Due to this effect, some of the oil close to the contact inlet will be rejected and will produce reverse flow (➔ **fig** 1). This reverse flow shears the lubricant, generating heat that lowers the oil viscosity and reduces the film thickness and rolling friction component.

For the effect described above, the inlet shear heating reduction factor can be obtained approximately from

$$\phi_{ish} = \frac{1}{1 + 1{,}84 \times 10^{-9} \, (n \, d_m)^{1{,}28} \, \nu^{0{,}64}}$$

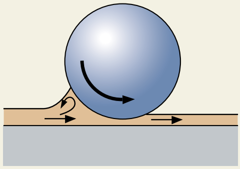
where

ϕ_{ish} = inlet shear heating reduction factor
n = rotational speed, r/min
d_m = the bearing mean diameter, mm
ν = kinematic viscosity of the lubricant at the operating temperature, mm^2/s (for grease lubrication the base oil viscosity)

Values for the inlet shear heating factor ϕ_{ish} can be obtained from **diagram** 1 as a function of the combined parameter $(n \, d_m)^{1{,}28} \, \nu^{0{,}64}$.

Fig 1

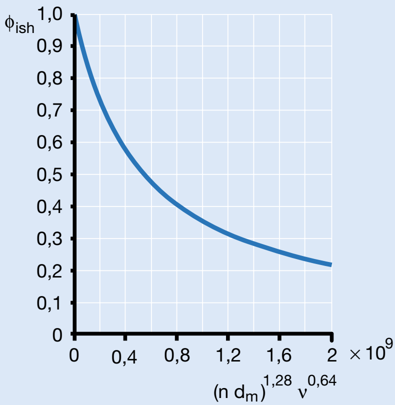
Reverse flow at the inlet of the contact



Lubricant reverse flow

Diagram 1

Inlet shear heating factor ϕ_{ish}



Kinematic replenishment/starvation reduction factor

For oil-spot, oil jet, low level oil bath lubrication conditions (e.g. oil level lower than the lowest rolling element centre) and grease lubrication, subsequent overrolling of the raceways might push away the excess of lubricant. Due to the bearing speed or high viscosity, the lubricant at the edges of the contacts might not have sufficient time to replenish the raceways, this effect is called "kinematic starvation" and causes a drop in the film thickness and in the rolling friction.

For the type of lubrication conditions described above, the kinematic replenishment/starvation reduction factor can be obtained approximately from

$$\phi_{rs} = \frac{1}{e^{K_{rs}\,\nu\, n\,(d+D)\sqrt{\frac{K_Z}{2\,(D-d)}}}}$$

where

ϕ_{rs} = kinematic replenishment/starvation reduction factor
e = base of natural logarithm = 2,718
K_{rs} = the replenishment/starvation constant
3×10^{-8} for low level oil bath and oil jet lubrication
6×10^{-8} for grease and oil-spot lubrication
K_Z = bearing type related geometry constant (➔ **table** 5)
ν = kinematic viscosity at the operating temperature, mm²/s
n = rotational speed, r/min
d = bearing bore diameter, mm
D = bearing outside diameter, mm

Drag losses in oil bath lubrication

As drag losses are the most important additional sources of friction, the additional source term therefore is reduced to the drag losses component M_{drag}.

In oil bath lubrication, the bearing is partially, or in special situations, completely submerged. Under these conditions the size and geometry of the oil reservoir together with the oil level used can have a substantial impact on the bearing frictional moment. For a very large oil bath, disregarding any reservoir size interaction and any influence of other mechanical elements working close to the bearing, e.g. external oil agitation, gears or cams, the drag losses in a bearing as a function of the oil level in the reservoir can be approximated from the variable V_M plotted in **diagram** 2 as a function of the oil level H (➔ **fig** 2) and the bearing mean diameter $d_m = 0,5\,(d + D)$. **Diagram** 2 can be applied for bearing speeds up to the reference speed of the bearing. At higher speeds and high oil levels other effects might have an important influence in the results.

The variable V_M in **diagram** 2 is related to the frictional moment of drag losses for ball bearings by

Table 5

Geometry constants K_Z and K_L for drag loss calculation

Bearing type	Geometry constants K_Z	K_L
Deep groove ball bearings		
– single and double row	3,1	–
Angular contact ball bearings		–
– single row	4,4	–
– double row	3,1	–
– four-point contact	3,1	–
Self-aligning ball bearings	4,8	–
Cylindrical roller bearings		
– single and double row	5,1	0,65
– full complement, single and double row	6,2	0,7
Taper roller bearings	6	0,7
Spherical roller bearings	5,5	0,8
CARB toroidal roller bearings		
– with cage	5,3	0,8
– full complement	6	0,75
Thrust ball bearings	3,8	–
Cylindrical roller thrust bearings	4,4	0,43
Spherical roller thrust bearings	5,6	0,58[1]

[1] Only for single mounted bearings

$$M_{drag} = V_M\, K_{ball}\, d_m^5\, n^2$$

and for roller bearings by

$$M_{drag} = 10\, V_M\, K_{roll}\, B\, d_m^4\, n^2$$

where

M_{drag} = frictional moment of drag losses, Nmm
V_M = variable as a function of the oil level according to **diagram 2**
K_{ball} = ball bearing related constant, see below
K_{roll} = roller bearing related constant, see below
d_m = bearing mean diameter, mm
B = bearing inner ring width, mm
n = rotational speed, r/min

Values for the variable V_M can be obtained in **diagram 2**, **page 99**, from the red curve for ball bearings and from the blue curve for roller bearings.

The ball bearing related constant is defined as

$$K_{ball} = \frac{i_{rw}\; K_Z\, (d + D)}{D - d} \times 10^{-12}$$

and the roller bearing related constant as

$$K_{roll} = \frac{K_L\; K_Z\, (d + D)}{D - d} \times 10^{-12}$$

where

K_{ball} = ball bearing related constant
K_{roll} = roller bearing related constant
i_{rw} = the number of ball rows
K_Z = bearing type related geometry constant (➔ **table 5**, **page 98**)
K_L = roller bearing type related geometry constant (➔ **table 5**, **page 98**)
d = bearing bore diameter, mm
D = bearing outside diameter, mm

Fig 2

Oil level in an oil bath

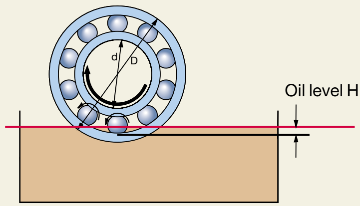
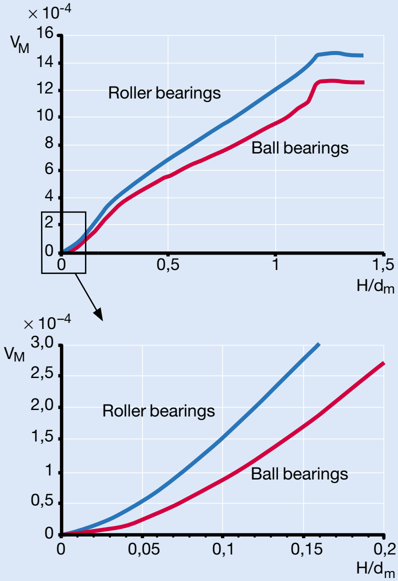


Diagram 2

Drag loss variable V_M



Note:
To calculate drag losses for oil jet lubrication, one can use the oil bath model, with the oil level to half the roller diameter and multiply the obtained value for M_{drag} by a factor of two.

To calculate drag losses for vertical shaft arrangements an approximate value can be obtained by using the model for fully submerged bearings and multiply the obtained value for M_{drag} by a factor equal to that width (height) that is submerged relative to the total bearing width (height).

Mixed lubrication for low speeds and viscosities
For operating conditions of small κ values (≤ 2) the application lies in the mixed lubrication regime; occasional metal-to-metal contact may occur, which increases friction. A sketch of a typical bearing frictional moment as a function of rotational speed and viscosity is depicted in **diagram** 3. During the start-up period with increasing speed or viscosity the frictional moment decreases, since a lubricating film is built up and the bearing enters into the full elasto-hydrodynamic (EHL) regime. With higher speeds or viscosities friction increases due to the increase of film thickness until high-speed starvation and thermal effects reduce friction again.

The sliding friction coefficient can be calculated with the following equation

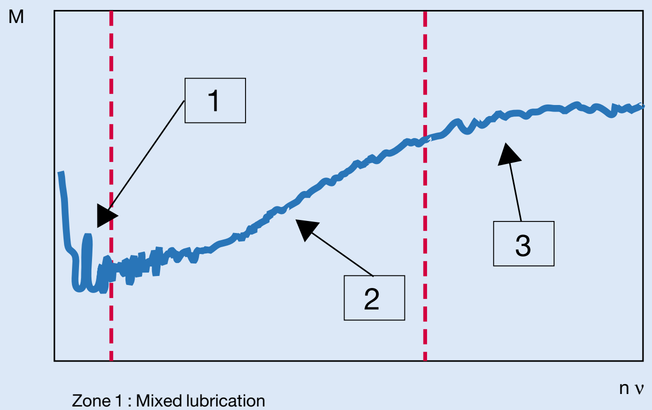
$$\mu_{sl} = \phi_{bl}\ \mu_{bl} + (1 - \phi_{bl})\ \mu_{EHL}$$

where

μ_{sl} = sliding friction coefficient
ϕ_{bl} = mixed lubrication weighting factor, see below
μ_{bl} = coefficient depending on the additive package in the lubricant, approximate value 0,15

Diagram 3

Bearing frictional moment as a function of the speed and viscosity



Zone 1 : Mixed lubrication
Zone 2 : EHL Elasto-hydrodynamic lubrication
Zone 3 : EHL + thermal and starvation effects

μ_{EHL} = friction coefficient in full film conditions:
0,05 for lubrication with mineral oils
0,04 for lubrication with synthetic oils
0,1 for lubrication with transmission fluids
For applications with cylindrical or taper roller bearings, use following values instead:
0,02 for cylindrical roller bearings
0,002 for taper roller bearings

The weighting factor for the sliding frictional moment can be estimated using the following equation

$$\phi_{bl} = \frac{1}{e^{2,6 \times 10^{-8} (n\,\nu)^{1,4} d_m}}$$

where

ϕ_{bl} = weighting factor for the sliding frictional moment
e = base of natural logarithm = 2,718
n = operational speed, r/min
ν = kinematic viscosity of the lubricant at the operating temperature, mm^2/s (for grease lubrication the base oil viscosity)
d_m = the bearing mean diameter, mm

An estimation of the weighting factor ϕ_{bl} for the sliding friction moment can be made using the curve shown in **diagram** 4.

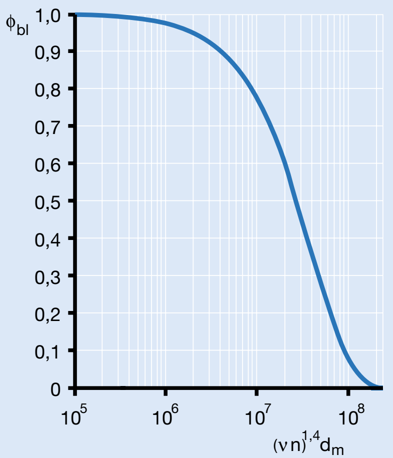
Effects of clearance and misalignment on friction

Changes in clearance and/or misalignment in bearings will modify the frictional moment. The above-described model considers normal clearance and an aligned bearing. However, high bearing operating temperatures or high speed might reduce internal bearing clearance, which can increase friction. Misalignment generally increases friction, however, for self-aligning ball bearings, spherical roller bearings, CARB toroidal roller bearings and spherical roller thrust bearings the corresponding increase of friction with misalignment is negligible.

For specific application conditions sensitive to changes of clearance and misalignment please contact the SKF application engineering service.

Diagram 4

Weighting factor for the sliding friction coefficient ϕ_{bl}



Effects of grease filling on friction

When grease lubrication is used, and the bearing has just been filled (or refilled) with the recommended amount of grease, the bearing can produce considerably higher frictional values during the first hours or days of operation (depending on the speed) than had been calculated originally. This is because the grease takes time to redistribute itself within the free space in the bearing; meanwhile it is churned and moved around. To estimate this effect, multiply the initial rolling frictional moment by a factor of 2 for light series and a factor of 4 for heavy series. However, after this "running-in" period, the frictional moment comes down to similar values as oil lubricated bearings; in many cases even lower values are possible. If the bearing is filled with an excessive amount of grease, higher values of friction in the bearing may result. Please refer to the section "Relubrication", starting on **page 237**, or contact the SKF application engineering service for more detailed information.

Frictional behaviour of hybrid bearings

Due to the higher values for the modulus of elasticity of ceramics, hybrid bearings will have smaller contact areas, which favour a reduction of the rolling and sliding friction components. In addition, the lower density of ceramics compared with steel reduces the centrifugal forces, and this also may reduce friction at high speeds.

In the above equations, the friction torque for hybrid angular contact ball bearings can be calculated by substituting the geometry constants R_3 and S_3 of the all-steel bearings by 0,41 R_3 and 0,41 S_3 respectively.

High speed designs with hybrid deep groove ball bearings include the practise to axially preload the bearing arrangement. The deep groove ball bearings will under such a condition act as angular contact ball bearings and thus see a similar reduction in friction level at high speeds. However, such a friction calculation needs to be done in cooperation with the SKF application engineering service.

Starting torque

The starting torque of a rolling bearing is defined as the frictional moment that must be overcome in order for the bearing to start rotating from the stationary condition. Under normal ambient temperature, +20 to +30 °C, starting at zero speed and $\mu_{sl} = \mu_{bl}$, the starting torque can be calculated using only the sliding frictional moment and the frictional moment of seals, if present. Therefore

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

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

However, the starting torque can be considerably higher for roller bearings with a large contact angle, up to four times as high as for taper roller bearings of series 313, 322 B, 323 B and T7FC, and up to eight times as high for spherical roller thrust bearings.

Power loss and bearing temperature

The power loss in the bearing as a result of bearing friction can be obtained using the equation

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

where
N_R = power loss, W
M = total frictional moment of the bearing, Nmm
n = operational speed, r/min

If the cooling factor (the heat to be removed from the bearing per degree of temperature difference between bearing and ambient) is known, a rough estimate of the temperature increase in the bearing can be obtained using

$$\Delta T = N_R/W_s$$

where
ΔT = temperature increase, °C
N_R = power loss, W
W_s = cooling factor, W/°C

Calculation example

A spherical roller bearing 22208 E is to operate at a speed of 3 500 r/min under the following operating conditions:

Actual radial bearing load $F_r = 2\,990$ N

Actual axial bearing load $F_a = 100$ N

Inner ring rotation

Operating temperature + 40 °C

Oil bath lubrication

Oil level H = 2,5 mm above the edge of the outer ring raceway under static conditions. Mineral oil having a kinematic viscosity $\nu = 68$ mm^2/s at 40 °C

Requirement:
What will be the total frictional moment?

1. Calculation of the geometry and load dependent variables

According to **table 2a** on **page 91** with bearing mean diameter

$d_m = 0{,}5\,(d + D) = 0{,}5\,(40 + 80) = 60$ mm

- Rolling friction variables

$$G_{rr.e} = R_1\, d_m^{1,85}\, (F_r + R_2\, F_a)^{0,54}$$

$$= 1{,}6 \times 10^{-6} \times 60^{1,85} \times$$

$$(2\,990 + 5{,}84 \times 100)^{0,54} = 0{,}26$$

$$G_{rr.l} = R_3\, d_m^{2,3}\, (F_r + R_4\, F_a)^{0,31}$$

$$= 2{,}81 \times 10^{-6} \times 60^{2,3}$$

$$(2\,990 + 5{,}8 \times 100)^{0,31} = 0{,}436$$

since $G_{rr.e} < G_{rr.l}$, then

$G_{rr} = 0{,}26$

- Sliding friction variables

$$G_{sl.e} = S_1\, d_m^{0,25}\, (F_r^{\,4} + S_2\, F_a^{\,4})^{1/3}$$

$$= 3{,}62 \times 10^{-3} \times 60^{0,25} \times$$

$$(2\,990^4 + 508 \times 100^4)^{1/3} = 434$$

$$G_{sl.l} = S_3\, d_m^{0,94}\, (F_r^{\,3} + S_4\, F_a^{\,3})^{1/3}$$

$$= 8{,}8 \times 10^{-3} \times 60^{0,94} \times$$

$$(2\,990^3 + 117 \times 100^3)^{1/3} = 1\,236{,}6$$

since $G_{sl.e} < G_{sl.l}$, then

$G_{sl} = 434$

2. Calculation of the rolling frictional moment

$$M_{rr} = G_{rr}\, (\nu\, n)^{0,6}$$

$$= 0{,}26 \times (68 \times 3\,500)^{0,6}$$

$$= 437{,}4 \text{ Nmm}$$

3. Calculation of the sliding frictional moment

Assuming full film conditions, $\kappa > 2$

$$M_{sl} = \mu_{sl}\, G_{sl}$$

$$= 0{,}05 \times 434 = 21{,}7 \text{ Nmm}$$

4. Calculation of the inlet shear heating reduction factor

$$\phi_{ish} = \frac{1}{1 + 1{,}84 \times 10^{-9} \times (n \times d_m)^{1,28}\, \nu^{0,64}}$$

$$= \frac{1}{1 + 1{,}84 \times 10^{-9} \times (3\,500 \times 60)^{1,28}\, \nu^{0,64}}$$

$$\approx 0{,}85$$

5. Calculation of kinematic replenishment/starvation reduction factor for oil bath lubrication

$$\phi_{rs} = \frac{1}{e^{K_{rs}\,\nu\, n\,(d+D)\sqrt{\frac{K_Z}{2\,(D-d)}}}}$$

$$= \frac{1}{e^{3 \times 10^{-8} \times 68 \times 3\,500 \times (40+80)\sqrt{\frac{5{,}5}{2 \times (80-40)}}}}$$

$$\approx 0{,}8$$

6. Calculation of the drag losses in oil bath lubrication

With a drag loss variable as function of

$H/d_m = 2{,}5/60 = 0{,}041$

From the **diagram** **2** on **page 99,** it can be seen that drag losses are small, since $H/d_m < 0{,}1$. However, they can still be taken into account. For roller bearings the drag loss variable V_M becomes approximately $0{,}03 \times 10^{-3}$.

Then the roller bearing related constant can be obtained from

$$K_{roll} = \frac{K_L\, K_Z\,(d+D)}{D-d} \times 10^{-12}$$

$$= \frac{0{,}8 \times 5{,}5 \times (40+80)}{80-40} \times 10^{-12}$$

$$= 13{,}2 \times 10^{-12}$$

The drag losses can then be obtained, as an approximation, from

$$M_{drag} = 10\, V_M\, K_{roll}\, B\, d_m^4\, n^2$$

$$= 10 \times 0{,}03 \times 10^{-3}$$

$$\times 13{,}2 \times 10^{-12} \times 23 \times 60^4 \times 3\,500^2$$

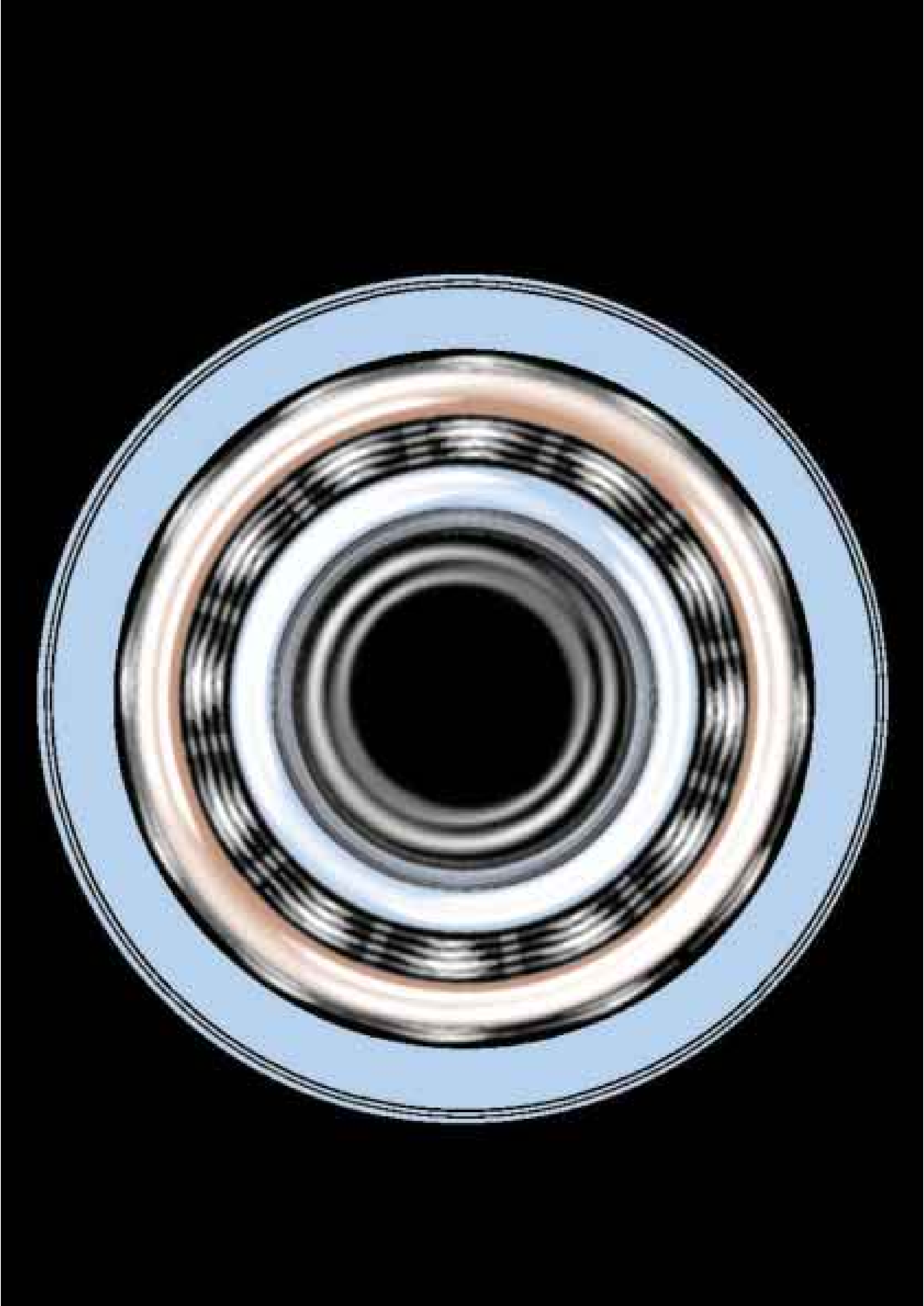
$$= 14{,}5 \text{ Nmm}$$

7. Calculation of the total frictional moment of 22208 E according to the new SKF model

$$M = \phi_{ish}\, \phi_{rs}\, M_{rr} + M_{sl} + M_{drag}$$

$$= 0{,}85 \times 0{,}8 \times 437{,}4 + 21{,}7 + 14{,}5$$

$$= 333{,}6 \text{ Nmm}$$



Speeds and vibration

There is a limit to the speed at which rolling bearings can be operated. Generally it is the operating temperature for the lubricant being used or the material of the bearing components that sets the limit.

The speed at which limiting operating temperature is reached depends on the frictional heat generated in the bearing (including any externally applied heat) and the amount of heat that can be transported away from the bearing.

Bearing type and size, internal design, load, lubrication and cooling conditions as well as cage design, accuracy and internal clearance all play a part in determining speed capability.

In the product tables generally two speeds are listed: (thermal) reference speed and (kinematical) limiting speed, the value of which depending on what criteria are considered.

Reference speeds

The (thermal) reference speed listed in the product tables represent a reference value that is to be used to determine the permissible operational speed of the bearing subjected to a certain load and running with a certain lubricant viscosity.

The values of the reference speed listed are in according with ISO 15312 (where thrust ball bearings are excluded). This ISO standard has been established for oil lubrication, but is also valid for grease lubrication.

The reference speed for a given bearing represents the speed, under specified operating conditions, at which there is equilibrium between the heat that is generated by the bearing and the heat that is dissipated from the bearing to the shaft, housing and lubricant.

The reference conditions according to ISO 15312 for obtaining this heat balance are:

- a temperature increase of 50 °C above an ambient temperature of 20 °C, i.e. a bearing temperature of 70 °C, measured on the bearing stationary outer ring or housing washer;
- radial bearing: a constant radial load, being 5 % of the basic static load rating C_0;
- thrust bearing: a constant axial load, being 2 % of the basic static load rating C_0;
- open bearings with Normal clearance

for oil lubricated bearings:
- lubricant: mineral oil without EP additives having a kinematic viscosity at 70 °C of:
  $\nu = 12\ mm^2/s$ (ISO VG 32) for radial bearings
  $\nu = 24\ mm^2/s$ (ISO VG 68) for thrust roller bearings
- method of lubrication; oil bath with the oil reaching up to the middle of the rolling element in the lowest position.

for grease lubricated bearings:
- lubricant: regular lithium soap grease with mineral base oil having a viscosity of 100 to 200 mm^2/s at 40 °C (e.g. ISO VG 150);
- grease quantity: approximately 30 % of the free space in the bearing.

A temperature peak may occur during initial start-up of a grease lubricated bearing. Therefore the bearing may have to be in operation for up to 10 to 20 hours before it reaches normal operating temperature.

Under these specific conditions the reference speed for oil and grease lubrication will be equal.

It may be necessary to reduce the ratings in applications where the outer ring rotates.

For certain bearings, where the speed limit is not determined by heat from the rolling element/raceway contacts, only limiting speeds are shown in the bearing tables. These include, for example, bearings with contact seals.

Influence of load and oil viscosity on reference speed/permissible speed
When load and viscosity values higher than the reference values are applied, the frictional resistance will increase so that a bearing cannot operate at the suggested reference speed, unless higher temperatures can be permitted. Lower viscosity values may result in higher operational speeds.

The influence of load and kinematic viscosity on the reference speed can be obtained from the diagrams:

Diagram 1: Radial ball bearings, **page 110**
Diagram 2: Radial roller bearings, **page 111**
Diagram 3: Thrust ball bearings, **page 112**
Diagram 4: Thrust roller bearings, **page 113**

Values of the adjustment factors for oil lubrication

- f_P: for the influence of the equivalent dynamic bearing load P and
- f_v: for the influence of viscosity

can be obtained from **diagrams** 1 to 4 as a function of P/C_0 and the bearing mean diameter d_m

where
P = equivalent dynamic bearing load, kN
C_0 = basic static load rating, kN
d_m = bearing mean diameter
= 0,5 (d + D), mm

The viscosity values in the diagrams are expressed with ISO designations, for example, ISO VG 32, where 32 is the oil viscosity at 40 °C.

If the reference temperature of 70 °C is to remain unchanged, the permissible speed is obtained from

$$n_{perm} = f_P\, f_v\, n_r$$

where
n_{perm} = permissible speed of bearing, r/min
n_r = reference speed, r/min
f_P = adjustment factor for bearing load P
f_v = adjustment factor for oil viscosity

The diagrams are also valid for grease lubrication. However, the reference speed for grease lubrication is based on base oil viscosity VG 150, but can also be used for the viscosity range ISO VG 100 – ISO VG 200. For other viscosities, the value of f_v needs to be calculated as f_v for the base oil viscosity at 40 °C of the selected grease, divided by f_v for an ISO VG 150 oil.

Example 1
A 6210 deep groove ball bearing is subjected to a load $P = 0,24\ C_0$ and has an oil bath lubrication with oil viscosity 68 mm^2/s at 40 °C. Which reference speed can be expected?

For bearing 6210: $d_m = 0,5\ (50 + 90) =$ 70 mm. From **diagram** 1, **page 110**, with $d_m = 70$ mm and $P/C_0 = 0,24$, $f_P = 0,63$ and with $P/C_0 = 0,24$ and ISO VG 68, $f_v = 0,85$.

The permissible bearing speed for which an operating temperature of 70 °C can be expected, n_{perm}, will then be

$$n_{perm} = 0,63 \times 0,85 \times 15\,000 = 8\,030 \text{ r/min}$$

Example 2
A 22222 E spherical roller bearing is subjected to a load $P = 0,15\ C_0$ and is grease lubricated having a base oil viscosity 220 mm^2/s at 40 °C. Which reference speed can be expected?

For bearing 22222 E: $d_m = 0,5\ (110 + 200) =$ 155 mm. From **diagram** 2, **page 111**, with $d_m = 155$ mm and $P/C_0 = 0,15$, $f_P = 0,53$ and with $P/C_0 = 0,15$ and ISO VG 220, $f_v = 0,83$; with $P/C_0 = 0,15$ and ISO VG 150, $f_v = 0,87$.

The permissible bearing speed for which an operating temperature of 70 °C can be expected, n_{perm}, will then be

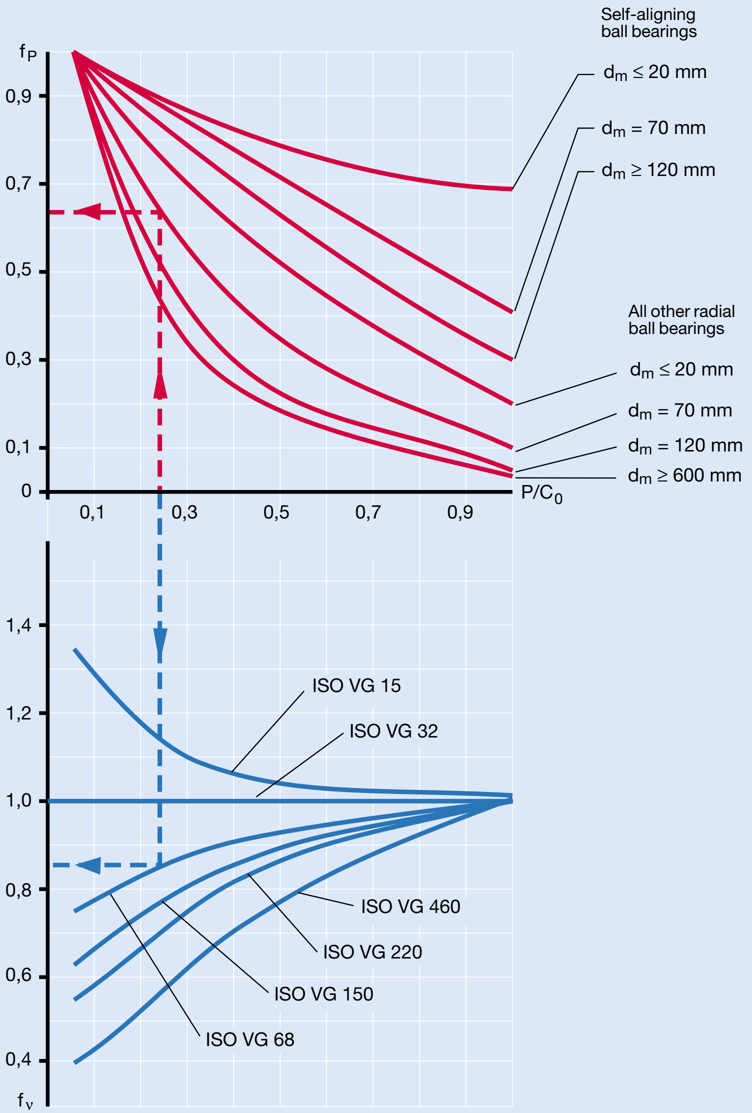
$$n_{perm} = 0,53 \times 0,83/0,87 \times 3\,000 = 1\,520 \text{ r/min}$$

Speeds above the reference speed
It is possible to operate bearings at speeds above the reference speed if the friction within the bearing can be reduced via a lubrication system that applies accurately measured small quantities of lubricant, or by removing heat either by a circulating oil lubrication system, by cooling ribs on the housing, or by directing cooling air streams (➔ section "Methods of oil lubrication", starting on **page 248**).

Any increase in speed above the reference speed without taking any of these precautions could cause bearing temperatures to rise excessively. An increase in bearing temperature means that lubricant viscosity is lowered and film formation is made more difficult, leading to even higher friction and further temperature increases. If, at the same time, the operational clearance in the bearing is reduced because of increased inner ring temperature, the final consequence

Diagram 1

Adjustment factors f_P and f_v for radial ball bearings



Adjustment factors f_P and f_ν for radial roller bearings

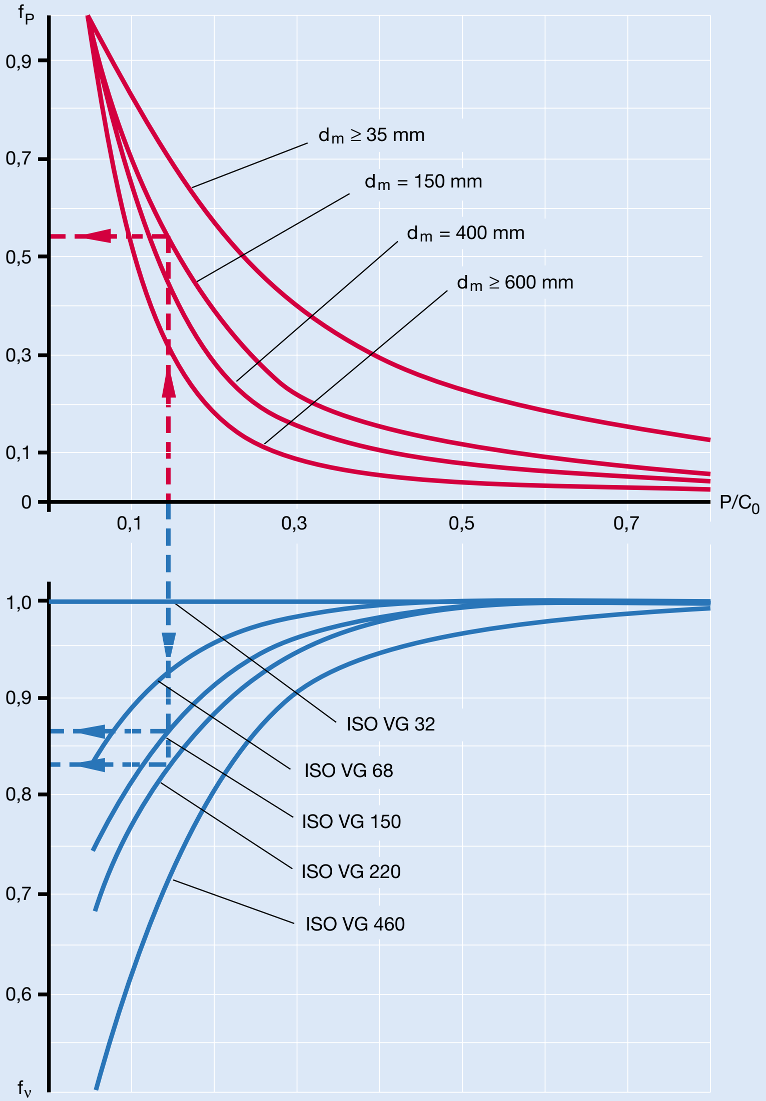


Diagram 3

Adjustment factors f_P and f_v for thrust ball bearings

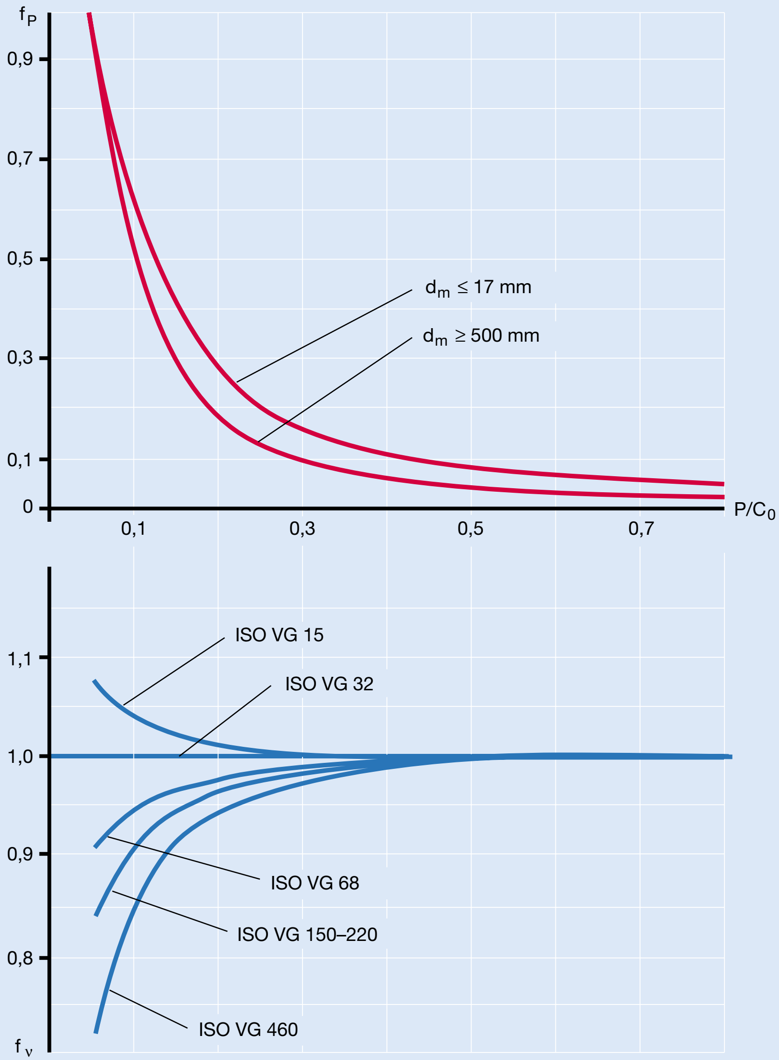
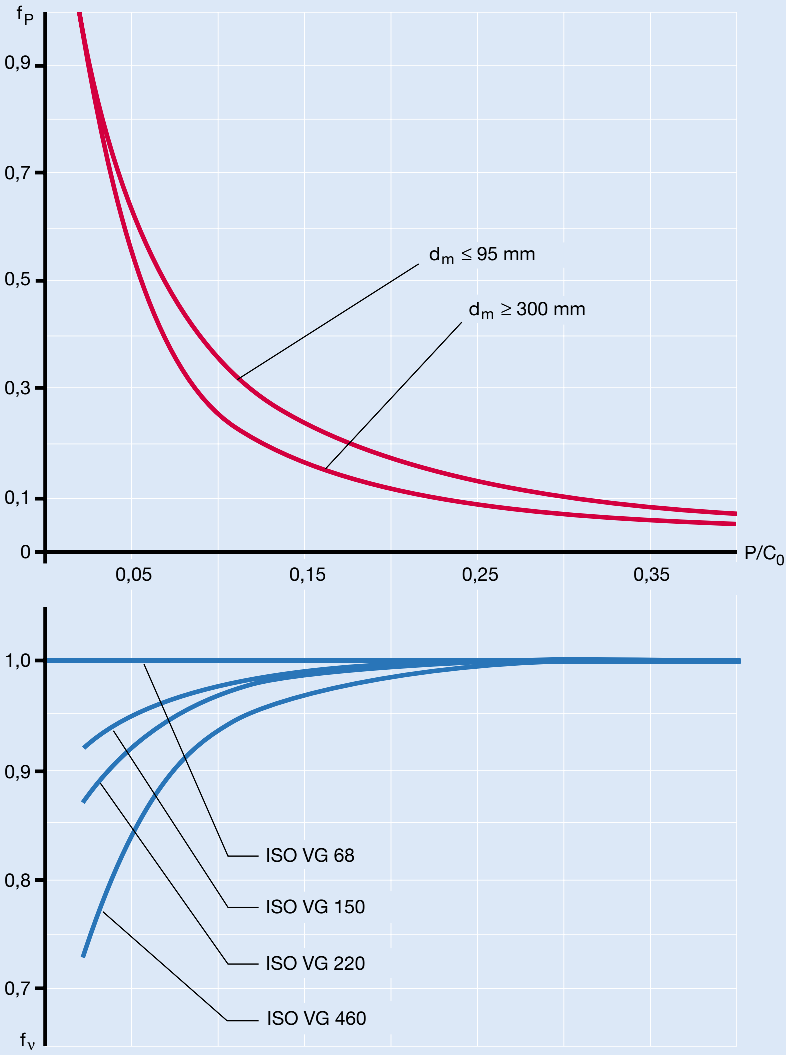


Diagram 4

Adjustment factors f_P and f_v for thrust roller bearings



would be bearing seizure. Any increase in speed above the reference speed generally means that the temperature difference between inner and outer rings is greater than normal. Usually, therefore, a bearing with a C3 internal clearance, which is greater than Normal, is required, and it may be necessary to look more closely at the temperature distribution in the bearing.

Limiting speeds

The limiting speed is determined by criteria that include the form stability or strength of the cage, lubrication of cage guiding surfaces, centrifugal and gyratory forces acting on the rolling elements, precision and other speed-limiting factors, such as seals and lubricant for sealed bearings.

Experience gained from laboratory tests and practical applications indicates that there are maximum speeds that should not be exceeded for technical reasons or because of the very high costs involved to keep the operating temperature at an acceptable level.

The limiting speeds shown in the bearing tables are valid for the bearing design and standard cage execution shown.

To run bearings at higher speeds than those shown in the tables some of the speed-limiting factors need to be improved, such as the running accuracy, cage material and design, lubrication and heat dissipation. It is therefore advisable to contact the SKF application engineering service.

For grease lubrication additional aspects should be considered such as lubrication of the cage guiding surfaces and the shear strength of the lubricant, which are determined by the base oil and thickener (➔ section “Grease lubrication”, starting on **page 231**).

Some open ball bearings have very low friction and reference speeds listed might be higher than the limiting speeds. Therefore, the permissible speed needs to be calculated and be compared to the limiting speed. The lower of the two values should be retained.

It should be remembered that if bearings are to function satisfactorily, at high speeds, they must be subjected to a given minimum load. Details will be found in the introductory texts of the product tables under the heading “Minimum load”.

Special cases

In certain applications the speed limits are superseded in importance by other considerations.

Low speeds
At very low speeds it is impossible for an elasto-hydrodynamic lubricant film to be built up in the contacts between the rolling elements and raceways. In these applications, lubricants containing EP additives should generally be used (➔ section “Grease lubrication”, starting on **page 231**).

Oscillating movements
With this type of movement the direction of rotation changes before the bearing has completed a single revolution. As the rotational speed is zero at the point where the direction of rotation is reversed, a full hydrodynamic film of lubricant cannot be maintained. In these cases it is important to use a lubricant containing an effective EP additive in order to obtain a boundary lubrication film that is able to support loads.

It is not possible to give a limit or a rating for the speed of such oscillating movements as the upper limit is not dictated by a heat balance but by the inertia forces that come into play. With each reversal of direction, there is a danger that inertia will cause the rolling elements to slide for a short distance and smear the raceways. The permissible accelerations and decelerations depend on the mass of the rolling elements and cage, the type and quantity of lubricant, the operational clearance and the bearing load. For connecting rod bearing arrangements, for example, preloaded bearings incorporating relatively small rolling elements with a small mass are used. General guidelines cannot be given and it is necessary to analyse the movements more precisely in individual cases. It is advisable to contact the SKF application engineering service.

Vibration generation in a bearing

In general a rolling bearing does not generate noise by itself. What is perceived as "bearing noise" is in fact the audible effect of the vibrations generated directly or indirectly by the bearing on the surrounding structure. This is the reason why most of the time noise problems can be considered as vibration problems involving the complete bearing application.

Excitation due to varying numbers of loaded rolling elements
When a radial load is applied to a bearing, the number of rolling elements carrying the load varies slightly during operation, i.e. 2-3-2-3…. This generates a displacement in the direction of the load. The resulting vibration cannot be avoided, but can be reduced by applying an axial preload to load all the rolling elements (not possible with cylindrical roller bearings).

Waviness of components
In cases where there is a tight fit between the bearing ring and the housing or the shaft, the bearing ring may take the shape of the adjacent component. If form deviations are present, these may cause vibrations during operation. It is therefore important to machine the shaft and housing seating to the required tolerances (➔ section "Tolerances for cylindrical form" on **page 194**).

Local damage
Due to mishandling or incorrect mounting, small sections on the raceways and rolling elements can be damaged. During operation, overrolling a damaged bearing component generates a specific vibration frequency. Frequency analysis of the vibrations can identify which bearing component suffered damage. This principle is used in SKF Condition Monitoring equipment to detect bearing damage.

To calculate SKF bearing frequencies please refer to the section "Calculations" in the SKF Interactive Engineering Catalogue on CD-ROM or online at www.skf.com.

Contaminants
If operating in contaminated conditions, dirt particles may enter the bearing and be overrolled by the rolling elements. The generated vibration level is dependent on the amount, the size and the composition of the overrolled contaminant particles. No typical frequency pattern is generated. However, an audible and disturbing noise may be created.

Influence of the bearing on the vibration behaviour of the application

In many applications bearing stiffness is of the same order as the stiffness of the surrounding structure. This opens the possibility to reduce vibrations of the application by properly choosing the bearing (including preload and clearance) and its arrangement in the application. There are three ways to reduce vibration:

- remove the critical excitation vibration from the application,
- dampen the critical excitation vibration between excitant component and resonant components,
- change the stiffness of the structure to change the critical frequency.


P63
R9
/QR
ZW
TN9
HT51B
2LS
D7
LM
D8
CC
RS
QE5

Bearing data – general

Dimensions

Manufacturers and users of rolling bearings are, for reasons of price, quality and ease of replacement, only interested in a limited number of bearing sizes. The International Organization for Standardization (ISO) has therefore laid down General Plans for the boundary dimensions of

- metric radial rolling bearings in standard ISO 15:1998, except taper roller bearings,
- metric radial taper roller bearings in standard ISO 355:1977 and
- metric thrust rolling bearings in standard ISO 104: 2002.

ISO General Plans

The ISO General Plans for boundary dimensions of radial bearings contains a progressive series of standardized outside diameters for every standard bore diameter arranged in Diameter Series 7, 8, 9, 0, 1, 2, 3 and 4 (in order of increasing outside diameter). Within each Diameter Series different Width Series have also been established (Width Series 8, 0, 1, 2, 3, 4, 5, 6 and 7 in order of increasing width). The Width Series for radial bearings corresponds to the Height Series for thrust bearings (Height Series 7, 9, 1 and 2 in order of increasing height).

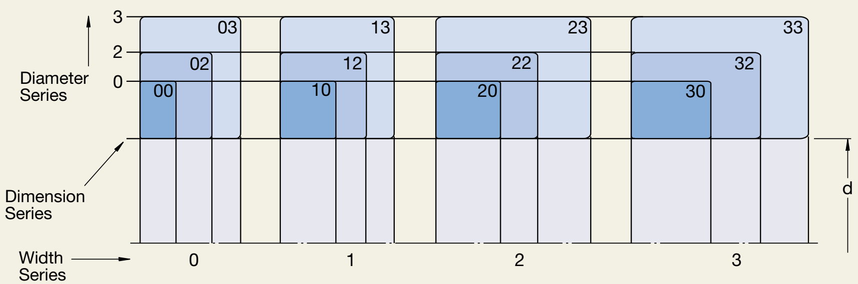
By combining a Width or Height Series with a Diameter Series, a Dimension Series, designated by two figures, is arrived at. The first figure identifies the Width or Height Series and the second the Diameter Series (➔ **fig 1**).

In the ISO General Plan for single row metric taper roller bearings, the boundary dimensions are grouped for certain ranges of the contact angle α, known as the Angle Series (Angle Series 2, 3, 4, 5, 6 and 7 in order of increasing angle). Based on the relationship between the outside and bore diameters, and between the total bearing width and the cross-sectional height, Diameter and Width Series have also been established. Here, a Dimension Series is obtained by combining the Angle Series with a Diameter and a Width Series (➔ **fig 2**). These Dimension Series consist of one figure for the Angle Series and two letters, where the first letter identifies the Diameter Series and the second the Width Series.

With a very few exceptions, dictated by rolling bearing development, the bearings in this catalogue comply with the ISO General Plans or with other ISO standards for the dimensions of some bearing types for which the ISO Dimension Series are inappropriate. Interchangeability is therefore guaranteed. Additional information is given under the heading “Dimensions” in the introductory texts to the individual product sections.

Experience has shown that the requirements of the vast majority of bearing applications can be met using bearings with these standardized dimensions.

Fig 1



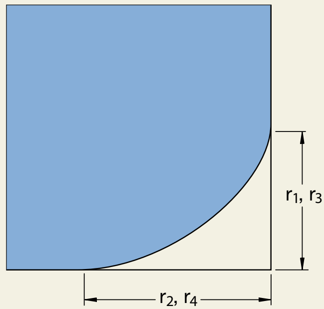
General Plans for inch-size bearings
A large group of bearings in inch sizes are inch-size taper roller bearings. The dimensions of these bearings conform to AFBMA Standard 19-1974 (ANSI B3.19-1975). ANSI/ABMA Standard 19.2-1994 has subsequently replaced this standard, but this later standard no longer includes dimensions.

In addition to the inch-size taper roller bearings, some inch-size ball bearings and cylindrical roller bearings that follow the earlier British Standard BS292-1:1982, are also available, but not shown in this catalogue. This standard has subsequently been withdrawn as a consequence of metrication and it is not recommended that these bearings be used for new designs.

Chamfer dimensions
Minimum values for the chamfer dimensions (➔ **fig 3**) in the radial direction (r_1, r_3) and the axial direction (r_2, r_4) are given in the product tables. These values are in accordance with the General Plans listed in the Standards

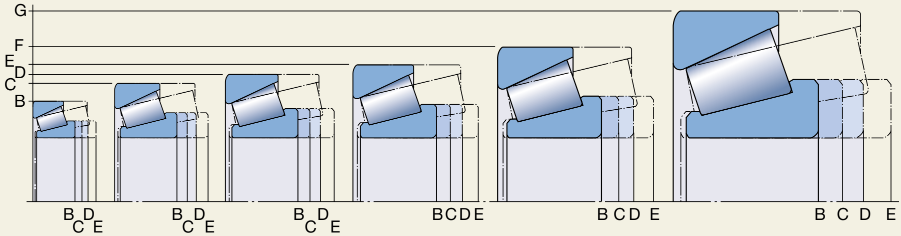
- ISO 15:1998, ISO 12043:1995 and ISO 12044:1995 for radial rolling bearings,
- ISO 355:1977 for radial taper roller bearings,
- ISO 104: 2002 for thrust rolling bearings.

Fig 3



The appropriate maximum chamfer limits, that are important when dimensioning fillet radii are in accordance with the Standard ISO 582:1995 and will be found in the section “Tolerances”, starting on **page 120**.

Fig 2



Tolerances

The dimensional and running accuracy of rolling bearings has been standardized internationally. In addition to the Normal tolerances the ISO standards also cover closer tolerances, e.g.

- tolerance class 6 which corresponds to SKF tolerance class P6
- tolerance class 5 which corresponds to SKF tolerance class P5.

For special applications, such as machine tool spindles, SKF also manufactures bearings with higher accuracy, e.g. to tolerance classes P4, P4A, PA9A, SP and UP. For additional information please refer to the SKF catalogue “High-precision bearings”.

Tolerance information about each bearing type is contained in the introductory texts to the various product table sections under the heading “Tolerances”. Bearings with higher accuracy than Normal are identified by a designation suffix for the tolerance class (➔ section “Supplementary designations”, starting on **page 151**).

Tolerance symbols
The tolerance symbols used in the tolerance **tables** 3 to 12 are listed together with their definitions in **table** 1 on **pages 122** and **123**.

Diameter Series identification
As the tolerances for the bore and outside diameter variation V_{dp} and V_{Dp} quoted in the tables for metric rolling bearings (except taper roller bearings) are not universally valid for all Diameter Series, and it is not always possible to immediately identify the ISO Diameter Series to which a bearing belongs from its designation, this information is provided in **table** 2 on **page 124**.

Tolerance tables
The actual tolerances are given in the tables referenced in the following.

Table 3: Normal tolerances for radial bearings, except taper roller bearings
Table 4: Class P6 tolerances for radial bearings, except taper roller bearings
Table 5: Class P5 tolerances for radial bearings, except taper roller bearings
Table 6: Normal and class CL7C tolerances for metric taper roller bearings
Table 7: Class CLN tolerances for metric taper roller bearings
Table 8: Class P5 tolerances for metric taper roller bearings
Table 9: Tolerances for inch-size taper roller bearings
Table 10: Tolerances for thrust bearings
Table 11: Normal, P6 and P5 class tolerances for tapered bore, taper 1:12
Table 12: Normal tolerances for tapered bore, taper 1:30

Where standardized, the values comply with ISO 492:2002, ISO 199:1997 and ANSI/ABMA Std 19.2:1994.

Limits for chamfer dimensions
To prevent the improper dimensioning of chamfers on associated components for rolling bearings and to facilitate the calculation of retaining ring location arrangements, the maximum chamfer limits for the relevant minimum chamfer dimensions (➔ **fig** 4) given in the product tables will be found in

Table 13: Chamfer dimension limits for metric radial and thrust bearings, except taper roller bearings
Table 14: Chamfer dimension limits for metric radial taper roller bearings
Table 15: Chamfer dimension limits for inch-size taper roller bearings

starting on **page 135**. These limits for metric bearings conform to ISO 582:1995. The chamfer dimension limits for inch-size taper roller bearings, which differ considerably from those for metric bearings, conform to ANSI/ABMA 19.2-1994.

The symbols used in the **tables** 13 to 15 are listed together with their definitions in **table** 1 on **pages 122** and **123**.

Fig 4

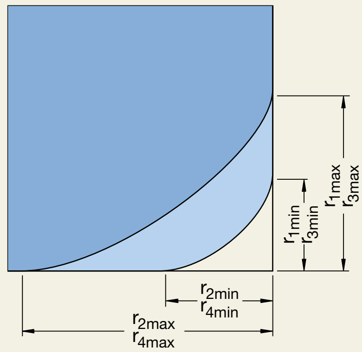


Table 1

Tolerance symbols

Tolerance symbol	Definition
	Bore diameter
d	Nominal bore diameter
d_s	Single bore diameter
d_{mp}	1. Mean bore diameter; arithmetical mean of the largest and smallest single bore diameters in one plane 2. Mean diameter at the small end of a tapered bore; arithmetical mean of the largest and smallest single diameters
Δ_{ds}	Deviation of a single bore diameter from the nominal ($\Delta_{ds} = d_s - d$)
Δ_{dmp}	Deviation of the mean bore diameter from the nominal ($\Delta_{dmp} = d_{mp} - d$)
V_{dp}	Bore diameter variation; difference between the largest and smallest single bore diameters in one plane
V_{dmp}	Mean bore diameter variation; i.e. the difference between the largest and smallest single plane mean bore diameters in one plane
d_1	Nominal diameter at theoretical large end of a tapered bore
d_{1mp}	Mean diameter at theoretical large end of tapered bore; arithmetical mean of the largest and smallest single bore diameters e
Δ_{d1mp}	Deviation of the mean bore diameter at the theoretical large end of a tapered bore from the nominal ($\Delta_{d1mp} = d_{1mp} - d_1$)
	Outside diameter
D	Nominal outside diameter
D_s	Single outside diameter
D_{mp}	Mean outside diameter; arithmetical mean of the largest and smallest single outside diameters in one plane
Δ_{Ds}	Deviation of a single outside diameter from the nominal ($\Delta_{Ds} = D_s - D$)
Δ_{Dmp}	Deviation of the mean outside diameter from the nominal ($\Delta_{Dmp} = D_{mp} - D$)
V_{Dp}	Outside diameter variation; difference between the largest and smallest single outside diameters in one plane
V_{Dmp}	Mean outside diameter variation; difference between the largest and smallest mean outside diameters of one ring or washer
	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

Tolerance symbols

Tolerance symbol	Definition
	Width
B, C	Nominal width of inner ring and outer ring, respectively
$\mathbf{B_s, C_s}$	Single width of inner ring and outer ring, respectively
$\mathbf{B_{1s}, C_{1s}}$	Single width of inner ring and outer ring, respectively, of a bearing specifically manufactured for paired mounting
Δ_{Bs}, Δ_{Cs}	Deviation of single inner ring width or single outer ring width from the nominal ($\Delta_{Bs} = B_s - B$; $\Delta_{Cs} = C_s - C$; $\Delta_{B1s} = B_{1s} - B_1$; $\Delta_{C1s} = C_{1s} - C_1$)
$\mathbf{V_{Bs}, V_{Cs}}$	Ring width variation; difference between the largest and smallest single widths of inner ring and of outer ring, respectively
$\mathbf{T_s}$	1. Single width (abutment width) of taper roller bearing; distance between inner ring (cone) back face and outer ring (cup) back face 2. Single height (H) of single direction thrust bearing (except spherical roller thrust bearing, see T_{4s})
$\mathbf{T_{1s}}$	1. Single width of cone assembled with master cup of taper roller bearing 2. Single height (H_1) of double direction thrust ball bearing with seating washer
$\mathbf{T_{2s}}$	1. Single width of cup assembled with master cone of taper roller bearing 2. Single height (H) of double direction thrust bearing
$\mathbf{T_{3s}}$	Single height (H_1) of double direction thrust ball bearing with seating washer
$\mathbf{T_{4s}}$	Single height (H) of spherical roller thrust bearing
$\Delta_{Ts}, \Delta_{T1s}$	1. Deviation of single width of taper roller bearing from the nominal ($\Delta_{Ts} = T_s - T$ etc.) 2. Deviation of single height of thrust bearing from the nominal ($\Delta_{Ts} = T_s - T$ etc.)
	Running accuracy
$\mathbf{K_{ia}, K_{ea}}$	Radial runout of assembled bearing inner ring and assembled bearing outer ring, respectively
$\mathbf{S_d}$	Side face runout with reference to bore (of inner ring)
$\mathbf{S_D}$	Outside inclination variation; variation in inclination of outside cylindrical surface to outer ring side face
$\mathbf{S_{ia}, S_{ea}}$	Side face runout of assembled bearing inner ring and assembled bearing outer ring, respectively
$\mathbf{S_i, S_e}$	Thickness variation, measured from middle of raceway to back (seating) face of shaft washer and of housing washer, respectively (axial runout)

Table 2

Diameter Series (radial bearings)

Bearing type	ISO Diameter Series 7, 8, 9	0, 1	2, 3, 4
Deep groove ball bearings[1]	607 617, 618, 619 627, 628 637, 638, 639	60 160, 161 630	2, 3 42, 43 62, 63, 64, 622, 623
Angular contact ball bearings			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
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 60	C 22, 23 C 32

[1] Bearings 604, 607, 608 and 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

[2] Bearing 108 belongs to diameter series 0, bearings 126, 127 and 129 to diameter series 2 and bearing 135 to series 3

Table 3

Normal tolerances for radial bearings, except taper roller bearings

Inner ring

d		Δ_{dmp} [1]		V_{dp} Diameter Series 7, 8, 9	0, 1	2, 3, 4	V_{dmp}	Δ_{Bs}		Δ_{B1s}		V_{Bs}	K_{ia}
over	incl.	high	low	max	max	max	max	high	low	high	low	max	max
mm		µm		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	0	–250	15	10
10	**18**	0	–8	10	8	6	6	0	–120	0	–250	20	10
18	**30**	0	–10	13	10	8	8	0	–120	0	–250	20	13
30	**50**	0	–12	15	12	9	9	0	–120	0	–250	20	15
50	**80**	0	–15	19	19	11	11	0	–150	0	–380	25	20
80	**120**	0	–20	25	25	15	15	0	–200	0	–380	25	25
120	**180**	0	–25	31	31	19	19	0	–250	0	–500	30	30
180	**250**	0	–30	38	38	23	23	0	–300	0	–500	30	40
250	**315**	0	–35	44	44	26	26	0	–350	0	–500	35	50
315	**400**	0	–40	50	50	30	30	0	–400	0	–630	40	60
400	**500**	0	–45	56	56	34	34	0	–450	0	–630	50	65
500	**630**	0	–50	63	63	38	38	0	–500	0	–800	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

1) Tolerances for tapered bores, → **tables 11** and **12** on **pages 133** and **134**

Outer ring

D		Δ_{dmp}		V_{Dp} [1] Diameter Series 7, 8, 9	0, 1	2, 3, 4	Bearings with shields or seals [2]	V_{Dmp} [1]	Δ_{Cs}, Δ_{C1s}, V_{Cs}	K_{ea}
over	incl.	high	low	max	max	max	max	max		max
mm		µm		µm				µm		µm
6	**18**	0	–8	10	8	6	10	6	Values are identical to those for inner ring of same bearing	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) Applies before bearing is assembled and after removal of internal and/or external snap ring
2) Applies only to bearings of Diameter Series 2, 3 and 4

Table 4

Class P6 tolerances for radial bearings, except taper roller bearings

Inner ring

d		Δ_{dmp}[1]		V_{dp} Diameter Series			V_{dmp}	Δ_{Bs}		Δ_{B1s}		V_{Bs}	K_{ia}
				7, 8, 9	0, 1	2, 3, 4							
over	incl.	high	low	max	max	max	max	high	low	high	low	max	max
mm		µm		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	0	–250	15	6
10	**18**	0	–7	9	7	5	5	0	–120	0	–250	20	7
18	**30**	0	–8	10	8	6	6	0	–120	0	–250	20	8
30	**50**	0	–10	13	10	8	8	0	–120	0	–250	20	10
50	**80**	0	–12	15	15	9	9	0	–150	0	–380	25	10
80	**120**	0	–15	19	19	11	11	0	–200	0	–380	25	13
120	**180**	0	–18	23	23	14	14	0	–250	0	–500	30	18
180	**250**	0	–22	28	28	17	17	0	–300	0	–500	30	20
250	**315**	0	–25	31	31	19	19	0	–350	0	–500	35	25
315	**400**	0	–30	38	38	23	23	0	–400	0	–630	40	30
400	**500**	0	–35	44	44	26	26	0	–450	0	–630	45	35
500	**630**	0	–40	50	50	30	30	0	–500	0	–800	50	40
630	**800**	0	–50	–	–	–	–	0	–750	–	–	55	45
800	**1 000**	0	–65	–	–	–	–	0	–1 000	–	–	60	50
1 000	**1 250**	0	–80	–	–	–	–	0	–1 250	–	–	70	60
1 250	**1 600**	0	–100	–	–	–	–	0	–1 600	–	–	70	70
1 600	**2 000**	0	–130	–	–	–	–	0	–2 000	–	–	80	80

[1] Tolerances for tapered bores, ➔ **table 11** on **page 133**

Outer ring

D		Δ_{Dmp}		V_{Dp}[1] Diameter Series			Sealed bearings[2]	V_{Dmp}[1]	Δ_{Cs}, Δ_{C1s}, V_{Cs}	K_{ea}
				7, 8, 9	0, 1	2, 3, 4				
over	incl.	high	low	max	max	max	max	max		max
mm		µm		µm				µm		µm
6	**18**	0	–7	9	7	5	9	5	Values are identical to those for inner ring of same bearing	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	–80	–	–	–	–	–		85
1 250	**1 600**	0	–100	–	–	–	–	–		100
1 600	**2 000**	0	–130	–	–	–	–	–		100
2 000	**2 500**	0	–160	–	–	–	–	–		120

[1] Applies before bearing is assembled and after removal of internal and/or external snap ring
[2] Applies only to bearings of Diameter Series 0, 1, 2, 3 and 4

Table 5

Class P5 tolerances for radial bearings, except taper roller bearings

Inner ring

d		Δ_{dmp}		V_{dp} Diameter Series 7, 8, 9	0, 1, 2, 3, 4	V_{dmp}	Δ_{Bs}		Δ_{B1s}		V_{Bs}	K_{ia}	S_d	S_{ia} [1)]
over	incl.	high	low	max	max	max	high	low	high	low	max	max	max	max
mm		µm		µm		µm	µm		µm		µm	µm	µm	µm
–	**2,5**	0	–5	5	4	3	0	–40	0	–250	5	4	7	7
2,5	**10**	0	–5	5	4	3	0	–40	0	–250	5	4	7	7
10	**18**	0	–5	5	4	3	0	–80	0	–250	5	4	7	7
18	**30**	0	–6	6	5	3	0	–120	0	–250	5	4	8	8
30	**50**	0	–8	8	6	4	0	–120	0	–250	5	5	8	8
50	**80**	0	–9	9	7	5	0	–150	0	–250	6	5	8	8
80	**120**	0	–10	10	8	5	0	–200	0	–380	7	6	9	9
120	**180**	0	–13	13	10	7	0	–250	0	–380	8	8	10	10
180	**250**	0	–15	15	12	8	0	–300	0	–500	10	10	11	13
250	**315**	0	–18	18	14	9	0	–350	0	–500	13	13	13	15
315	**400**	0	–23	23	18	1	0	–400	0	–630	15	15	15	20
400	**500**	0	–27	28	21	1	0	–450	0	–630	18	17	18	23
500	**630**	0	–33	35	26	1	0	–500	0	–800	20	19	20	25
630	**800**	0	–40	–	–	–	0	–750	–	–	26	22	26	30
800	**1 000**	0	–50	–	–	–	0	–1 000	–	–	32	26	32	30
1 000	**1 250**	0	–65	–	–	–	0	–1 250	–	–	38	30	38	30
1 250	**1 600**	0	–80	–	–	–	0	–1 600	–	–	45	35	45	30
1 600	**2 000**	0	–100	–	–	–	0	–2 000	–	–	55	40	55	30

1) Applies only to deep groove and angular contact ball bearings

Outer ring

D		Δ_{Dmp}		V_{Dp} [1)] Diameter Series 7,8,9	0,1,2,3,4	V_{Dmp}	Δ_{Cs}, Δ_{C1s}	V_{Cs}	K_{ea}	S_D	S_{ea} [2)]
over	incl.	high	low	max	max	max		max	max	max	max
mm		µm		µm		µm		µm	µm	µm	µm
6	**18**	0	–5	5	4	3	Values are identical to those for inner ring of same bearing	5	5	8	8
18	**30**	0	–6	6	5	3		5	6	8	8
30	**50**	0	–7	7	5	4		5	7	8	8
50	**80**	0	–9	9	7	5		6	8	8	10
80	**120**	0	–10	10	8	5		8	10	9	11
120	**150**	0	–11	11	8	6		8	11	10	13
150	**180**	0	–13	13	10	7		8	13	10	14
180	**250**	0	–15	15	11	8		10	15	11	15
250	**315**	0	–18	18	14	9		11	18	13	18
315	**400**	0	–20	20	15	10		13	20	13	20
400	**500**	0	–23	23	17	12		15	23	15	23
500	**630**	0	–28	28	21	14		18	25	18	25
630	**800**	0	–35	35	26	18		20	30	20	30
800	**1 000**	0	–40	50	29	25		25	35	25	35
1 000	**1 250**	0	–50	–	–	–		30	40	30	45
1 250	**1 600**	0	–65	–	–	–		35	45	35	55
1 600	**2 000**	0	–85	–	–	–		38	55	40	55
2 000	**2 500**	0	–110	–	–	–		45	65	50	55

1) Does not apply to sealed or shielded bearings
2) Applies only to deep groove and angular contact ball bearings

Table 6

Normal and class CL7C tolerances for metric taper roller bearings

Inner ring, bearing width and ring widths

d		Δ_{dmp}		V_{dp}	V_{dmp}	Δ_{Bs}		K_{ia} Class Normal	CL7C	Δ_{Ts}		Δ_{T1s}		Δ_{T2s}	
over	incl.	high	low	max	max	high	low	max	max	high	low	high	low	high	low
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		Δ_{Dmp}		V_{Dp}	V_{Dmp}	Δ_{Cs}	K_{ea} Class Normal	CL7C
over	incl.	high	low	max	max		max	max
mm		µm		µm	µm		µm	
18	**30**	0	–12	12	9	Values are identical to those for inner ring of same bearing	18	9
30	**50**	0	–14	14	11		20	10
50	**80**	0	–16	16	12		25	13
80	**120**	0	–18	18	14		35	18
120	**150**	0	–20	20	15		40	20
150	**180**	0	–25	25	19		45	23
180	**250**	0	–30	30	23		50	–
250	**315**	0	–35	35	26		60	–
315	**400**	0	–40	40	30		70	–
400	**500**	0	–45	45	34		80	–
500	**630**	0	–50	50	38		100	–
630	**800**	0	–75	75	55		120	–

Table 7

Class CLN tolerances for metric taper roller bearings

Inner ring, bearing width and ring widths

d		Δ_{dmp}		V_{dp}	V_{dmp}	Δ_{Bs}		Δ_{Cs}		K_{ia}	Δ_{Ts}		Δ_{T1s}		Δ_{T2s}	
over	incl.	high	low	max	max	high	low	high	low	max	high	low	high	low	high	low
mm		µm		µm	µm	µm		µm		µm	µm		µm		µm	
10	**18**	0	–12	12	9	0	–50	0	–100	15	+100	0	+50	0	+50	0
18	**30**	0	–12	12	9	0	–50	0	–100	18	+100	0	+50	0	+50	0
30	**50**	0	–12	12	9	0	–50	0	–100	20	+100	0	+50	0	+50	0
50	**80**	0	–15	15	11	0	–50	0	–100	25	+100	0	+50	0	+50	0
80	**120**	0	–20	20	15	0	–50	0	–100	30	+100	0	+50	0	+50	0
120	**180**	0	–25	25	19	0	–50	0	–100	35	+150	0	+50	0	+100	0
180	**250**	0	–30	30	23	0	–50	0	–100	50	+150	0	+50	0	+100	0
250	**315**	0	–35	35	26	0	–50	0	–100	60	+200	0	+100	0	+100	0
315	**400**	0	–40	40	30	0	–50	0	–100	70	+200	0	+100	0	+100	0

Outer ring

D		Δ_{Dmp}		V_{Dp}	V_{Dmp}	K_{ea}
over	incl.	high	low	max	max	max
mm		µm		µm	µm	µm
18	**30**	0	–12	12	9	18
30	**50**	0	–14	14	11	20
50	**80**	0	–16	16	12	25
80	**120**	0	–18	18	14	35
120	**150**	0	–20	20	15	40
150	**180**	0	–25	25	19	45
180	**250**	0	–30	30	23	50
250	**315**	0	–35	35	26	60
315	**400**	0	–40	40	30	70
400	**500**	0	–45	45	34	80
500	**630**	0	–50	50	38	100

Table 8

Class P5 tolerances for metric taper roller bearings

Inner ring and bearing width

d		Δ_{dmp}		V_{dp}	V_{dmp}	Δ_{Bs}		K_{ia}	S_d	Δ_{Ts}	
over	incl.	high	low	max	max	high	low	max	max	high	low
mm		µm		µm	µm	µm		µm	µm	µm	
10	**18**	0	–7	5	5	0	–200	5	7	+200	–200
18	**30**	0	–8	6	5	0	–200	5	8	+200	–200
30	**50**	0	–10	8	5	0	–240	6	8	+200	–200
50	**80**	0	–12	9	6	0	–300	7	8	+200	–200
80	**120**	0	–15	11	8	0	–400	8	9	+200	–200
120	**180**	0	–18	14	9	0	–500	11	10	+350	–250
180	**250**	0	–22	17	11	0	–600	13	11	+350	–250
250	**315**	0	–25	19	13	0	–700	16	13	+350	–250
315	**400**	0	–30	23	15	0	–800	19	15	+400	–400

Outer ring

D		Δ_{Dmp}		V_{Dp}	V_{Dmp}	Δ_{Cs}	K_{ea}	S_D
over	incl.	high	low	max	max		max	max
mm		µm		µm	µm		µm	µm
18	**30**	0	–8	6	5	Values are identical to those for inner ring of same bearing	6	8
30	**50**	0	–9	7	5		7	8
50	**80**	0	–11	8	6		8	8
80	**120**	0	–13	10	7		10	9
120	**150**	0	–15	11	8		11	10
150	**180**	0	–18	14	9		13	10
180	**250**	0	–20	15	10		15	11
250	**315**	0	–25	19	13		18	13
315	**400**	0	–28	22	14		20	13
400	**500**	0	–33	25	17		23	15
500	**630**	0	–38	29	19		25	18

Tolerances for inch-size taper roller bearings

Inner ring

d		Δ_{ds} Tolerance classes Normal, CL2		CL3, CL0	
over	incl.	high	low	high	low
mm		µm			
–	**76,2**	+13	0	+13	0
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		Δ_{Ds} Tolerance classes Normal CL2		CL3, CL0		K_{ia}, K_{ea}, S_{ia}, S_{ea} Tolerance classes Normal	CL2	CL3	CL0
over	incl.	high	low	high	low	max	max	max	max
mm		µm				µm			
–	**304,8**	+25	0	+13	0	51	38	8	4
304,8	**609,6**	+51	0	+25	0	51	38	18	9
609,6	**914,4**	+76	0	+38	0	76	51	51	26
914,4	**1 219,2**	+102	0	+51	0	76	–	76	38
1 219,2	**–**	+127	0	+76	0	76	–	76	–

Abutment width of single row bearing

d		D		Δ_{Ts} Tolerance classes Normal		CL2		CL3,CL0	
over	incl.	over	incl.	high	low	high	low	high	low
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		Shaft washer Tolerance classes Normal, P6, P5			Tolerance classes Normal	P6	P5	Housing washer Tolerance classes Normal, P6, P5			
		Δ_{dmp}		V_{dp}	S_i [1]	S_i [1]	S_i [1]	Δ_{Dmp}		V_{Dp}	S_e
over	incl.	high	low	max	max	max	max	high	low	max	max
mm		µm		µm	µm	µm	µm	µm		µm	
–	**18**	0	–8	6	10	5	3	0	–11	8	Values are identical to those 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	–	40	25	13	0	–75	55	
800	**1 000**	0	–100	–	45	30	15	0	–100	75	
1 000	**1 250**	0	–125	–	50	35	18	0	–125	–	
1 250	**1 600**	0	–160	–	60	40	21	0	–160	–	
1 600	**2 000**	–	–	–	–	–	–	0	–200	–	
2 000	**2 500**	–	–	–	–	–	–	0	–250	–	

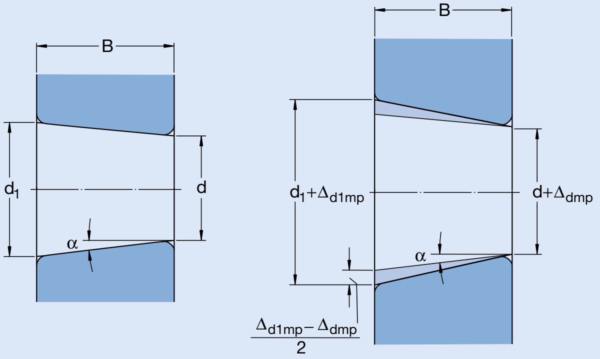
1) Does not apply to spherical roller thrust bearings

Bearing height

d		Tolerance classes Normal, P6, P5 Δ_{Ts}		Δ_{T1s}		Δ_{T2s}		Δ_{T3s}		Δ_{T4s} ISO		SKF	SKF Explorer
over	incl.	high	low	high	low	high	low	high	low	high	low	low	low
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	–400	–125	–100
80	**120**	+25	–300	+150	–300	+200	–500	+400	–500	0	–400	–150	–100
120	**180**	+25	–400	+150	–400	+200	–600	+400	–600	0	–500	–175	–125
180	**250**	+30	–400	+150	–400	+250	–600	+500	–600	0	–500	–200	–125
250	**315**	+40	–400	–	–	–	–	–	–	0	–700	–225	–150
315	**400**	+40	–500	–	–	–	–	–	–	0	–700	–300	–200
400	**500**	+50	–500	–	–	–	–	–	–	0	–900	–420	–
500	**630**	+60	–600	–	–	–	–	–	–	0	–1 200	–500	–
630	**800**	+70	–750	–	–	–	–	–	–	0	–1 400	–630	–
800	**1 000**	+80	–1 000	–	–	–	–	–	–	0	–1 800	–800	–
1 000	**1 250**	–	–	–	–	–	–	–	–	0	–2 400	–1 000	–
1 250	**1 600**	–	–	–	–	–	–	–	–	0	–	–1 200	–

Table 11

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



Half angle of taper 1:12

$\alpha = 2° \, 23' \, 9{,}4''$

Largest theoretical diameter d_1

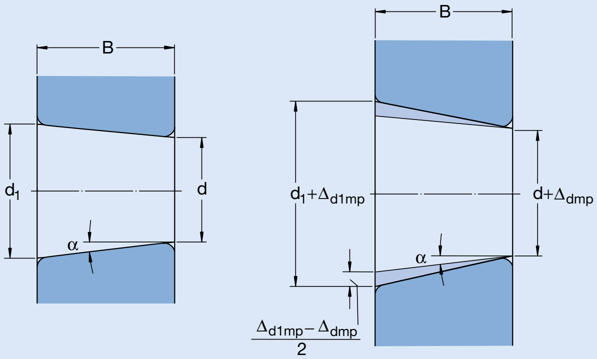
$d_1 = d + \frac{1}{12} \times B$

Bore diameter		Tolerance classes Normal, P6					Tolerance class P5				
d		Δ_{dmp}		V_{dp} [1]	$\Delta_{d1mp} - \Delta_{dmp}$		Δ_{dmp}		V_{dp} [1]	$\Delta_{d1mp} - \Delta_{dmp}$	
over	incl.	high	low	max	high	low	high	low	max	high	low
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	25	+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	0	+44	0	–	+44	0
630	**800**	+80	0	–	+80	0	+50	0	–	+50	0
800	**1000**	+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] Applies in any single radial plane of the bore

Table 12

Normal tolerances for tapered bores, taper 1:30



Half angle of taper 1:30

$\alpha = 0° \, 57' \, 17{,}4''$

Largest theoretical diameter d_1

$d_1 = d + \frac{1}{30} \times B$

Bore diameter d		Normal tolerances Δ_{dmp}		V_{dp} [1]	$\Delta_{d1mp} - \Delta_{dmp}$	
over	incl.	high	low	max	high	low
mm		µm		µm	µm	
80	**120**	+20	0	25	+40	0
120	**180**	+25	0	31	+50	0
180	**250**	+30	0	38	+55	0
250	**315**	+35	0	44	+60	0
315	**400**	+40	0	50	+65	0
400	**500**	+45	0	56	+75	0
500	**630**	+50	0	63	+85	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 single radial plane of the bore

Table 13

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

Minimum single chamfer dimension	Nominal bearing bore diameter		Maximum chamfer dimensions		
			Radial bearings		Thrust bearings
$r_{s\,min}$	d over	incl.	$r_{1,3}$ max	$r_{2,4}$ max	$r_{1,2,3,4}$ max
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 14

Chamfer dimension limits for metric radial taper roller bearings

Minimum single chamfer dimension	Nominal bearing bore/outside diameter		Maximum chamfer dimensions	
$r_{s\,min}$	d,D over	incl.	$r_{1,3}$ max	$r_{2,4}$ max
mm	mm		mm	
0,3	–	40	0,7	1,4
	40	–	0,9	1,6
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 **15**

Chamfer dimension limits for inch-size taper roller bearings

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

Bearing internal clearance

Bearing internal clearance (➔ **fig 5**) 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).

It is necessary to distinguish between the internal clearance of a bearing before mounting and the internal clearance in a mounted bearing which has reached its operating temperature (operational clearance). The initial internal clearance (before mounting) is greater than the operational clearance because different degrees of interference in the fits and differences in thermal expansion of the bearing rings and the associated components cause the rings to be expanded or compressed.

The radial internal clearance of a bearing is of considerable importance if satisfactory operation is to be obtained. As a general rule, ball bearings should always have an operational clearance that is virtually zero, or there may be a slight preload. Cylindrical, spherical and CARB toroidal roller bearings, on the other hand, should always have some residual clearance – however small – in operation. The same is true of taper roller bearings except in bearing arrangements where stiffness is desired, e.g. pinion bearing arrangements, where the bearings are mounted with a certain amount of preload (➔ section "Bearing preload", starting on **page 206**).

The bearing internal clearance referred to as Normal has been selected so that a suitable operational clearance will be obtained when bearings are mounted with the fits usually recommended and operating conditions are normal. Where operating and mounting conditions differ from the normal, e.g. where interference fits are used for both bearing rings, unusual temperatures prevail etc., bearings with greater or smaller internal clearance than Normal are required. In such cases, SKF recommends checking residual clearance in the bearing after it has been mounted.

Bearings having an internal clearance other than Normal are identified by the suffixes C1 to C5 (➔ **table 16**).

Tables giving the clearance values for the various bearing types will be found in the text preceding the relevant product section. For paired single row angular contact ball bearings and taper 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 clearance as the axial clearance is of greater importance in application design for these bearing types.

Fig 5

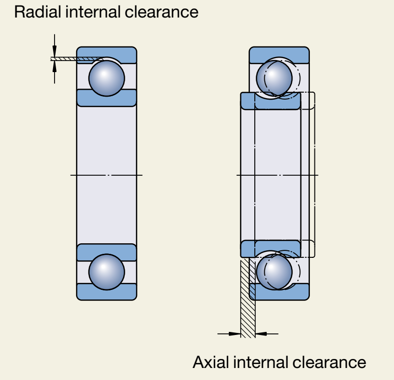


Table 16

Supplementary designation for internal clearance

Suffix	Internal clearance
C1	Less than C2
C2	Less than Normal
CN	Normal, only used in combination with letters indicating reduced or displaced clearance range.
C3	Greater than Normal
C4	Greater than C3
C5	Greater than C4

Materials for rolling bearings

The materials from which the bearing components are made determine to a large extent the performance and reliability of rolling bearings. For the bearing rings and rolling elements typical considerations include hardness for load carrying capacity, fatigue resistance under rolling contact conditions, under clean or contaminated lubrication conditions, and the dimensional stability of the bearing components. For the cage, considerations include friction, strain, inertia forces, and in some cases, the chemical action of certain lubricants, solvents, coolants and refrigerants. The relative importance of these considerations can be affected by other operational parameters such as corrosion, elevated temperatures, shock loads or combinations of these and other conditions.

Because SKF has the competence and facilities to provide a variety of materials, processes and coatings, SKF application engineers can assist in selecting those bearings that will provide superior performance for particular applications.

Contact seals integrated in rolling bearings can also have a considerable impact on the performance and reliability of the bearings. The materials they are made of have to offer excellent oxidation, thermal or chemical resistance.

In order to meet the needs of various applications, SKF uses different materials for bearing rings, rolling elements, cages and seals. Furthermore, in applications where sufficient lubrication cannot be achieved or where an electric current passing through the bearings has to be prevented, SKF bearings can be supplied with special coatings.

Materials for bearing rings and rolling elements

Through-hardening bearing steels

The most common through hardening bearing steel is a carbon chromium steel containing approximately 1 % carbon and 1,5 % chromium according to ISO 683-17:1999. Today, carbon-chromium steel is one of the oldest and most intensively investigated steels; due to the continuously increasing demands for extended bearing service life. The composition of this rolling bearing steel provides an optimum balance between manufacturing and application performance. This steel is normally given a martensitic or bainitic heat treatment during which it is hardened to the range of 58 to 65 HRC.

Within the last few years process developments have enabled more stringent cleanliness specifications to be realized, which has had a significant impact on the consistency and quality of SKF's bearing steel. The reduction of oxygen and harmful non-metallic inclusions has led to significantly improved properties of rolling bearing steels – the steels from which the SKF Explorer class bearings are made.

Induction-hardening bearing steels

Surface induction hardening offers the possibility to selectively harden a component's raceway, while leaving the remainder of the component unaffected by the hardening process. The steel grade and the manufacturing processes used prior to surface induction hardening dictate the properties in the unaffected area, which means that a combination of properties can be achieved in one component.

An example of this would be a flanged wheel hub bearing unit (HBU) where the properties of the unhardened flange are designed to resist structural fatigue, while the raceway is designed to resist rolling contact fatigue.

Case-hardening bearing steels

Chromium-nickel and manganese-chromium alloyed steels according to ISO 683-17:1999 with a carbon content of approximately 0,15 % are those case-hardening steels most commonly used for SKF rolling bearings.

In applications where there are high tensile interference fits and high shock loads, bearings with carburized rings and/or rolling elements are recommended.

Stainless bearing steels

The most common stainless steel used for SKF bearing rings and rolling elements is the high chromium content steel X65Cr14 according to ISO 683-17:1999.

It should be noted that for some applications, corrosion resistant coatings might be an excellent alternative to stainless steel. For additional information about alternative coatings, please consult the SKF application engineering service.

High-temperature bearing steels

Depending on the bearing type, standard bearings made from through hardened and surface-hardened steels have a recommended maximum operating temperature, which differs between 120 and 200 °C. The maximum operating temperature is directly related to the heat treatment process used in manufacturing components.

For operating temperatures up to 250 °C; a special heat treatment (stabilization) can be applied. In this case a reduction of the load carrying capacity of the bearing has to be taken into account.

For bearings operated at elevated temperatures, higher than 250 °C, for extended periods, highly alloyed steels like the 80MoCrV42-16 manufactured to ISO 683-17:1999 should be used because they retain their hardness and bearing performance characteristics even under extreme temperature conditions.

For additional information about high temperature bearing steels, please contact the SKF application engineering service.

Ceramics

The common ceramic used for SKF bearing rings and rolling elements is a bearing grade silicon nitride material. It consists of fine elongated grains of beta-silicon nitride in a glassy phase matrix. It provides a combination of favourable properties for rolling bearings, such as high hardness, low density, low thermal expansion, high electric resistivity, low dielectric constant and no response to magnetic fields (➔ **table 17**).

Table 17

Comparison of the material properties of bearing steel and silicon nitride

Material properties	**Bearing steel**	**Bearing grade silicon nitride**
Mechanical properties		
Density (g/cm^3)	7,9	3,2
Hardness	700 HV10	1 600 HV10
Modulus of elasticity (kN/mm^2)	210	310
Thermal expansion (10^{-6}/K)	12	3
Electrical properties (at 1 MHz)		
Electrical resistivity (Ωm)	$0,4 \times 10^{-6}$	10^{12}
	(Conductor)	(Insulator)
Dielectric strength (kV/mm)	–	15
Relative dielectric constant	–	8

Cage materials

Sheet steel cages
The majority of pressed sheet steel cages are made from continuously hot-rolled low carbon sheet steel according to (DIN) EN 10111:1998. These lightweight cages have relatively high strength and can be surface treated to further reduce friction and wear.

Pressed cages normally used in stainless steel bearings are made from stainless steel X5CrNi18-10 according to ISO 683-17:1999.

Machined steel cages

Machined steel cages are normally made of non-alloyed structural steel of type S355GT (St 52) according to EN 10 025:1990 + A:1993. To improve sliding and wear resistance properties some machined steel cages are surface treated.

Machined steel cages are used for large-size bearings or in applications where there is a danger that season cracking, caused by a chemical reaction, may occur if a brass cage were used. Steel cages can be used at operating temperatures up to 300 °C. They 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.

Sheet brass cages

Pressed sheet brass cages are used for some small and medium-sized bearings. The brass used for these cages conforms to EN 1652:1997. In applications such as compressors for refrigeration using ammonia, season cracking in sheet brass might occur, therefore machined brass or steel cages should be used instead.

Machined brass cages

Most brass cages are machined from a CW612N cast or wrought brass according to EN 1652:1997. They are unaffected by most common bearing lubricants, including synthetic oils and greases, and can be cleaned using normal organic solvents.

Brass cages should not be used at temperatures in excess of 250 °C.

Polymer cages

Polyamide 6,6
For the majority of injection moulded cages polyamide 6,6 is used. This material, with glass fibre reinforcement or without, is characterized by a favourable combination of strength and elasticity. The mechanical properties like strength and elasticity of polymeric materials are temperature dependent and subject to permanent changes under operating conditions, called ageing. The most important factors that play a role in this ageing behaviour are temperature, time and the medium (lubricant) to which the polymer is exposed. The relationship between these factors for glass fibre reinforced polyamide 6,6 is illustrated in **diagram 1**. It appears that the 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. In **table 18** the classification of lubricants into “aggressive” and “mild” is reflected by the “permissible operating temperature” for the use of cages made from glass fibre reinforced polyamide 6,6 in various lubricants. The permissible operating temperature in this table is defined as the temperature, which gives a cage ageing life of at least 10 000 operating hours.

Some media are even more “aggressive” than those listed in **table 18**. A typical example is ammonia applied as refrigerant in compressors. In those cases, cages made from glass fibre reinforced polyamide 6,6 should not be used at operating temperatures above + 70 °C or SKF needs to be consulted.

Towards the low operating temperature side, also a limit can be set since polyamide loses its elasticity which can result in cage failures. Cages made from glass fibre reinforced polyamide 6,6 should for this reason not be applied at a continuous operating temperature below –40 °C.

Where a high degree of toughness is a dominant factor, such as in railway axleboxes, a super-tough modification of polyamide 6,6 is applied. Please consult the SKF application engineering service for cage availability for specific bearing executions.

Polyamide 4,6
Glass fibre reinforced polyamide 4,6 is used for some small and medium size CARB toroidal roller bearings as standard. These cages have a 15 °C higher permissible operating temperature than those made from glass fibre reinforced polyamide 6,6.

Polyether ether ketone
The use of the glass fibre reinforced PEEK for cages has become common within SKF for demanding conditions regarding high speeds, chemical attack or high temperatures. The exceptional properties of PEEK are superior combination of strength and flexibility, high operating temperature range, high chemical and wear resistance and good processability. Due to these outstanding features, PEEK cages are available as standard for some ball and cylindrical roller bearings, like hybrid and/or high-precision bearings. The material does not show signs of ageing by temperature and oil additives up to +200 °C. However, the maximum temperature for high-speed use is limited to +150 °C as this is the softening temperature of the polymer.

Table 18

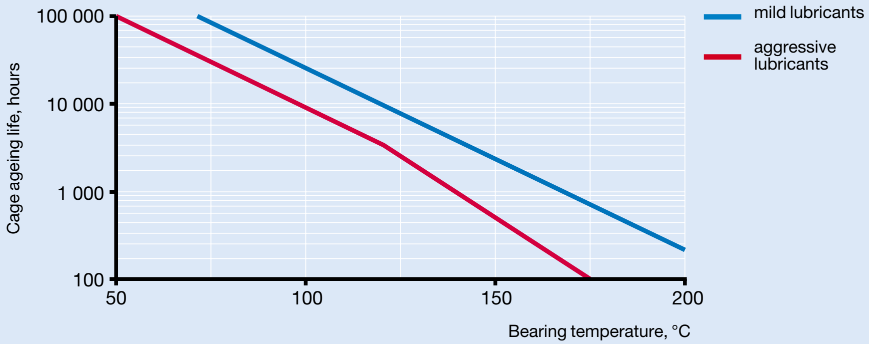
Permissible operating temperatures for cages of glass fibre reinforced polyamide 6,6 with various bearing lubricants

Lubricant	**Permissible operating temperature**[1]
Mineral oils	
Oils without EP additives, e.g. machine or hydraulic oils	120 °C
Oils with EP additives, e.g. industrial and automotive gearbox oils	110 °C
Oils with EP additives, e.g. automotive rear axle and differential gear oils (automotive), hypoid gear oils	100 °C
Synthetic oils	
Polyglycols, poly-alpha-olefins	120 °C
Diesters, silicones	110 °C
Phosphate esters	80 °C
Greases	
Lithium greases	120 °C
Polyurea, bentonite, calcium complex greases	120 °C
For sodium and calcium greases and other greases with a maximum operating temperature below 120 °C, the maximum temperature for the polyamide cage is the same as the maximum operating temperature for the grease.	

[1] Measured on the outside surface of the outer ring

Diagram 1

Cage ageing life for glass fibre reinforced polyamide 6,6



Phenolic resin cages
Lightweight, fabric reinforced phenolic resin cages can withstand high centrifugal as well as acceleration forces but are not able to accommodate high operating temperatures. In most cases, these cages are used as standard in high-precision angular contact ball bearings.

Other materials
In addition to the materials described above, SKF bearings for special applications may be fitted with cages made of other engineering polymer materials, light alloys or special cast iron. For information on cages made from alternative materials please consult the SKF application engineering service.

Seal materials
Seals integrated in SKF bearings are typically made from elastomer materials. The type of material can depend on the series and size of the bearing as well as the application requirements. SKF seals are generally produced from the materials listed below.

Acrylonitrile butadiene rubber

Acrylonitrile butadiene rubber (NBR) is the "universal" seal material. This copolymer, produced from acrylonitrile and butadiene, shows good resistance to the following media

- most mineral oils and greases with a mineral oil base,
- normal fuels: petrol, diesel and light heating oils,
- animal and vegetable oils and fats, and
- hot water

It also tolerates short-term dry running of the sealing lip. The permissible operating temperature range is –40 to +100 °C; for brief periods temperatures of up to +120 °C can be tolerated. At higher temperatures the material hardens.

Hydrogenated acrylonitrile butadiene rubber

Hydrogenated acrylonitrile butadiene rubber (HNBR) has appreciably better wear characteristics than nitrile rubber so that seals made of this material have a longer service life. Hydrogenated acrylonitrile butadiene rubber is also more resistant to heat, ageing and hardening in hot oil or ozone.

Mixtures of oil in air may have a negative impact on seal life. The upper operating temperature limit is +150 °C, which is appreciably higher than that of normal nitrile rubber.

Fluoro rubber

Fluoro rubbers (FPM) are characterized by their high thermal and chemical resistance. Their resistance to ageing and ozone is very good and their gas permeability is very slight. They have exceptionally good wear characteristics even under harsh environmental conditions and can withstand operating temperatures up to +200 °C. Seals made from this material can tolerate dry running of the lip for short periods.

Fluoro rubbers are also resistant to oils and hydraulic fluids, fuels and lubricants, mineral acids and aliphatic as well as aromatic hydrocarbons which would cause seals made from other materials to fail. In the presence of esters, ethers, ketones, certain amines and hot anhydrous hydrofluorides fluoro rubbers should not be used.

At temperatures above 300 °C, fluoro rubber gives off dangerous fumes. As handling seals made of fluoro rubber constitutes a potential safety risk, the safety precautions mentioned hereafter must always be considered.

Polyurethane

Polyurethane (AU) is a wear-resistant organic material, which has good elastic properties. It withstands operating temperatures in the range of –20 up to +80 °C. It has good resistance to mineral oil based greases, mineral oils with no or low quantity of EP additives, water and water-oil mixtures for example. It is not resistant to acids, alkalics or polar solvents.

WARNING!

Safety precautions for fluoro rubber

Fluoro rubber is very stable and harmless in normal operating conditions up to +200 °C. However, if exposed to extreme temperatures above 300 °C, e.g. fire or the flame of a cutting torch, fluoro rubber seals give off hazardous fumes. These fumes can be harmful if inhaled, as well as to the eyes. In addition, once the seals have been heated to such temperatures, they are dangerous to handle even after they have cooled and should not touch 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 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 material safety data sheet (MSDS).

If there is unintentional contact with the seals, wash hands with soap and plenty of water and 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 fluoro rubber seals or for any injury resulting from their use.

Coatings

Coating is a well-established method to upgrade materials and to provide bearings with additional features for specific application conditions. Two different coating methods developed by SKF are available and already successfully proven in many applications.

The surface coating, trademarked NoWear®, applies a low friction ceramic coating on the bearing inner surfaces to withstand long periods of operation under marginal lubrication for example. More details can be found in the section "NoWear bearings", starting on **page 939**.

The SKF INSOCOAT® coating, which can be applied to the exterior of the outer ring or inner ring of a bearing, offers resistance to the damage that can be caused by the passage of electric current through the bearing. More details can be found in the section "INSOCOAT bearings", starting on **page 905**.

Other coatings like zinc chromate for example, can offer an alternative to stainless steel in a corrosive environment, especially for ready-to-mount bearing units.

Cages

Cages have an appreciable influence on the suitability of rolling bearings. Their main purposes are

- keeping the rolling elements at an appropriate distance from each other and to prevent direct contact between neighbouring rolling elements, in order to keep friction and thus heat generation at a minimum;
- keeping the rolling elements evenly distributed around the complete circumference to provide even load distribution and quiet and uniform running;
- guiding the rolling elements in the unloaded zone, to improve the rolling conditions in the bearing and to prevent damaging sliding movements;
- retaining the rolling elements, where bearings are of a separable design and one bearing ring is removed during mounting or dismounting.

Cages are mechanically stressed by frictional, strain and inertia forces and they may also be subjected to the chemical action of certain lubricants, lubricant additives or products of their ageing, organic solvents or coolants. Therefore the design and material are of paramount importance for the performance of the cage as well as for the operational reliability of the bearing itself. This is the reason why SKF has developed various cage types and designs of different materials for the different bearing types.

In the introductory text to each product section information is provided regarding the standard cages fitted to the bearings and also possible alternatives. If a bearing with a non-standard cage is required it is always advisable to check availability before ordering.

In general the cages for SKF rolling bearings can be classified as pressed, solid or pin-type cages

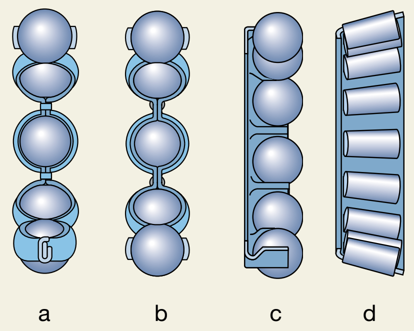
Pressed cages

Pressed cages for SKF bearings generally are made of sheet steel and with some exceptions of sheet brass (➔ **fig 6**). Depending on the bearing type pressed cages are designed as

- ribbon-type brass or steel cage (**a**)
- riveted steel cage (**b**)
- snap-type brass or steel cage (**c**)
- extremely strong window-type steel cage (**d**)

Pressed cages offer the advantage of lower weight as well as the advantage of more space inside the bearing, which facilitates entry of the lubricant into the bearing.

Fig 6



Solid cages

Solid cages for SKF bearings are made from brass, steel, light alloy, polymer or fabric reinforced phenolic resin (➔ **fig** 7). Depending on the bearing design they are designed as

- two-piece machined riveted cage (**a**),
- two-piece machined cage with integral rivets (**b**),
- one-piece machined window-type cage (**c**),
- double pronged machined cage (**d**),
- injection moulded polymer window-type cage (**e**),
- injection moulded polymer snap-type cage (**f**),
- one-piece machined cage of fabric reinforced phenolic resin.

Machined metal cages generally permit higher speeds and are necessary when movements additional to pure rotation are superimposed, particularly when conditions of high acceleration prevail. Suitable steps must be taken (e.g. oil lubrication) to provide sufficient supply of lubricant to the guiding surfaces of the cage and to the inside of the bearing. Machined cages are centred (➔ **fig** 8) either on the

- rolling elements (**a**),
- inner ring shoulder(s) (**b**) or
- outer ring shoulder(s) (**c**)

and are thus radially guided.

Fig 8

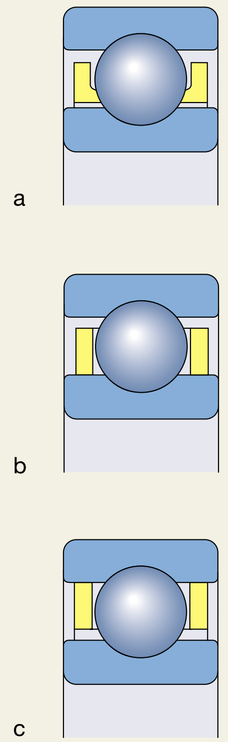
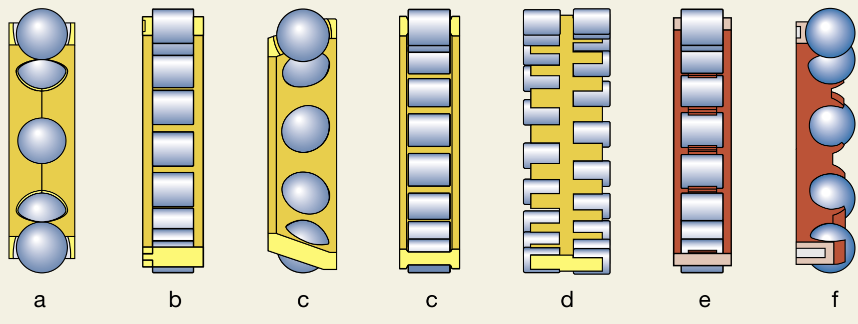


Fig 7

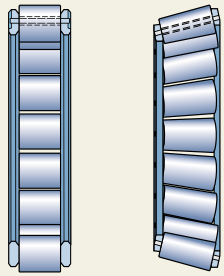


Solid polymer cages are characterized by a favourable combination of strength and elasticity. The good sliding properties of the polymer on lubricated steel surfaces and the smoothness of the cage surfaces in contact with the rolling elements produce just little friction so that heat generation and wear in the bearing are at a minimum. The low density of the material means that the inertia of the cage is small. The excellent running properties of polymer cages under lubricant starvation conditions permit continued operation of the bearing for a time without risk of seizure and secondary damage.

Pin-type cages
Steel pin-type cages need pierced rollers (➔ **fig 9**) and are only used together with large-sized roller bearings. These cages have relative low weight and allow a large number of rollers being incorporated.

Materials
Detailed information about materials used for cages can be found in the section "Materials for rolling bearings", starting on **page 138**.

Fig 9



Designations

Designations of rolling bearings consist of combinations of figures and/or letters, the significance of which is not immediately apparent. Therefore, the SKF designation system for rolling bearings will be described and the significance of the more common supplementary designations explained. To avoid confusion, the designations used for specific rolling bearing types, such as needle roller bearings, Y-bearings or high-precision bearings are not covered. More information about these will be found in the relevant catalogues. Also very specific bearing types, such as fixed-section bearings, slewing bearings or linear bearings are not covered either. These designations differ sometimes considerably from the system described here.

Bearing designations are divided into two main groups: designations for standard bearings and designations for special bearings. Standard bearings are bearings that normally have standardized dimensions, whereas special bearings have special dimensions dictated by customer demands. These customized bearings are also referred to as “drawing number” bearings and they will not be covered in detail in this section.

The complete designation may consist of a basic designation with or without one or more supplementary designations (➔ **fig 10**). The complete bearing designation, i.e. the basic designation with supplementary designations is always marked on the bearing package, whereas the designation marked on the bearing may sometimes be incomplete, e.g. for manufacturing reasons.

Basic designations identify the

- type,
- basic design and
- standard boundary dimensions

of a bearing. Supplementary designations identify

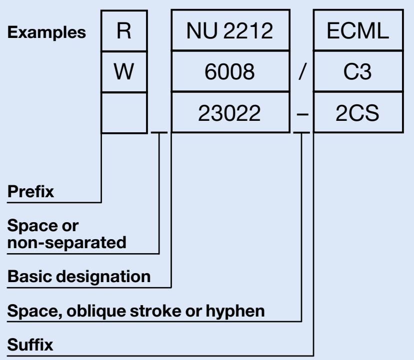
- bearing components and/or
- variants having a design and/or feature(s) that differ in some respect from the basic design.

Supplementary designations may precede the basic designation (prefixes) or follow it (suffixes). Where several supplementary designations are used to identify a given bearing, they are always written in a given order (➔ **fig 11**, **page 150**).

The list of supplementary designations presented in the following is not exhaustive but includes those most commonly used.

Fig 10

Bearing designation system



Basic designations

All SKF standard bearings have a characteristic basic designation, which generally consists of 3, 4 or 5 figures, or a combination of letters and figures. The design of the system used for almost all standard ball and roller bearing types is shown schematically in **diagram** 2. The figures and combinations of letters and figures have the following meaning:

- the first figure or the first letter or combination of letters identifies the bearing type; the actual bearing type can be seen from the presentation (➔ **diagram** 2).
- the following two figures identify the ISO Dimension Series; the first figure indicates the Width or Height Series (dimensions B, T or H respectively) and the second the Diameter Series (dimension D).
- the last two figures of the basic designation give the size code of the bearing; when multiplied by 5, the bore diameter in millimetres is obtained.

But there is no rule without some exceptions. The most important ones in the bearing designation system are listed below.

1. In a few cases the figure for the bearing type and/or the first figure of the Dimension Series identification is omitted. These figures are given in brackets in **diagram** 2.
2. For bearings having a bore diameter smaller than 10 mm or equal to or greater than 500 mm, the bore diameter is generally given in millimetres and is not coded. 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 of standard bearings according to ISO 15 that have bore diameters of 22, 28 or 32 mm, e.g. 62/22 (d = 22 mm).
3. Bearings with bore diameters of 10, 12, 15 and 17 mm have the following size code identifications:
   00 = 10 mm
   01 = 12 mm
   02 = 15 mm
   03 = 17 mm
4. For some smaller bearings having a bore diameter below 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 which deviate from the standard bore diameter of a bearing have always been given uncoded, in millimetres with up to three decimal places. This bore diameter identification is part of the basic designation and is separated from the basic designation by an oblique stroke, e.g. 6202/15.875 (d = 15,875 mm).

Series designations

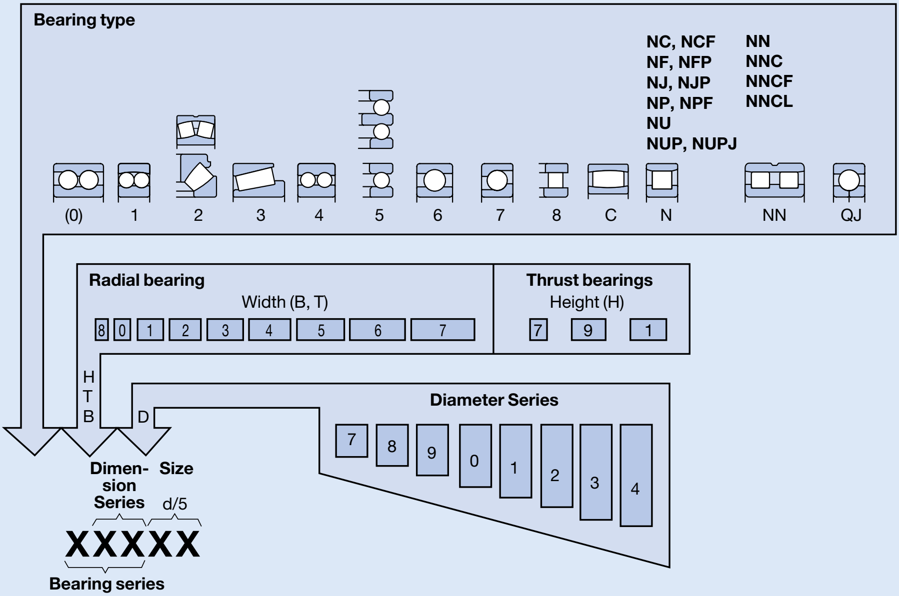
Each standard bearing belongs to a given bearing series, which is identified by the basic designation without the size identification. Series designations often include a suffix A, B, C, D or E or a combination of these letters e.g. CA. These are used to identify differences in internal design, e.g. contact angle.

The most common series designations are shown in **diagram** 2 above the bearing sketches. The figures in brackets are not included in the series designation.

Designation system for SKF standard metric ball and roller bearings

Bearing series

						6(0)4						
						623						
					544	6(0)3				(0)4		
		223			524	622				33		
		213			543	6(0)2				23		
		232			523	630			23	(0)3		
		222			542	6(1)0			32	22		
		241			522	16(0)0			22	12		
		231				639			41	(0)2		
		240	323		534	619			31	31	41	
		230	313		514	609			60	30	31	
		249	303		533	638			50	20	60	
	139	239	332		513	628	7(0)4	814	40	10	50	
	130	248	322		532	618	7(0)3	894	30	39	40	23
	(1)23	238	302		512	608	7(0)2	874	69	29	30	(0)3
	1(0)3		331		511	637	7(1)0	813	59	19	69	12
	(1)22	294	330		510	627	719	893	49	38	49	(0)2
(0)33	1(0)2	293	320	4(2)3	591	617	718	812	39	28	39	10
(0)32	1(1)0	292	329	4(2)2	590	607	708	811	29	18	48	19



Code	Bearing type	Code	Bearing type	Code	Bearing type
0	Double row angular contact ball bearings	**7**	Single row angular contact ball bearings	**QJ**	Four-point contact ball bearings
1	Self-aligning ball bearings	**8**	Cylindrical roller thrust bearings	**T**	Taper roller bearings according to ISO 355-1977
2	Spherical roller bearings, spherical roller thrust bearings	**C**	CARB toroidal roller bearings		
3	Taper roller bearings	**N**	Cylindrical roller bearings. A second and sometimes a third letter are used to identify the number of the rows or the configuration of the flanges, e.g. NJ, NU, NUP, NN, NNU, NNCF etc.		
4	Double row deep groove ball bearings				
5	Thrust ball bearings				
6	Single row deep groove ball bearings				

Fig 11

Designation system for suffixes

Designation example		Group 1	Group 2	Group 3	/	Group 4.1	4.2	4.3	4.4	4.5	4.6
6205-RS1NRTN9/P63LT20CVB123	6205		-RS1NR	TN9	/		P63			LT20C	VB123
23064 CCK/HA3C084S2W33	23064	CC	K		/	HA3	C084		S2	W33	

Basic designation

Space

Suffixes

Group 1: Internal design

Group 2: External design (seals, snap ring groove etc.)

Group 3: Cage design

Oblique stroke

Group 4: Variants

Group 4.1: Materials, heat treatment

Group 4.2: Accuracy, clearance, quiet running

Group 4.3: Bearing sets, paired bearings

Group 4.4: Stabilization

Group 4.5: Lubrication

Group 4.6: Other variants

Supplementary designations

Prefixes

Prefixes are used to identify components of a bearing and are usually then followed by the designation of the complete bearing, or to avoid confusion with other bearing designations. For example they are used in front of designations for taper roller bearings according to a system described in ANSI/ ABMA Standard 19 for (predominantly) inch-size bearings.

GS	Housing washer of a cylindrical roller thrust bearing
K	Cylindrical roller and cage thrust assembly
K-	Inner ring with roller and cage assembly (cone) or outer ring (cup) of inch-size taper roller bearing belonging to an ABMA standard series.
L	Separate inner or outer ring of a separable bearing
R	Inner or outer ring with roller (and cage) assembly of a separable bearing
W	Stainless steel deep groove ball bearing
WS	Shaft washer of a cylindrical roller thrust bearing
ZE	Bearing with SensorMount® feature

Suffixes

Suffixes are used to identify designs or variants which differ in some way from the original design, or which differ from the current standard design. The suffixes are divided into groups and when more than one special feature is to be identified; suffixes are given in the order shown in the scheme in **fig 11**, **page 150**.

The most commonly used suffixes are listed below. Note that not all variants are available.

A	Deviating or modified internal design with the same boundary dimensions. As a rule the significance of the letter is bound to the particular bearing or bearing series. Examples: 4210 A: Double row deep groove ball bearing without filling slots 3220 A: Double row angular contact ball bearing with a 30° contact angle
AC	Single row angular contact ball bearing with a 25° contact angle
ADA	Modified snap ring grooves in the outer ring; a two-piece inner ring held together by a retaining ring
B	Deviating or modified internal design with the same boundary dimensions. As a rule the significance of the letter is bound to the particular bearing series. Examples: 7224 B: Single row angular contact ball bearing with a 40° contact angle 32210 B: Steep contact angle taper roller bearing
Bxx(x)	B combined with a two or three-figure number identifies variants of the standard design that cannot be identified by generally applicable suffixes. Example: B20: Reduced width tolerance
C	Deviating or modified internal design with the same boundary dimensions. As a rule the significance of the letter is bound to the particular bearing series. Example: 21306 C: Spherical roller bearing with a flangeless inner ring, symmetrical rollers, loose guide ring and a window-type steel cage

CA 1. Spherical roller bearing of C design, but with retaining flanges on the inner ring and a machined cage
2. Single row angular contact ball bearing for universal matching. Two bearings arranged back-to-back or face-to-face will have a slight axial clearance before mounting

CAC Spherical roller bearing of the CA design but with enhanced roller guidance

CB 1. Single row angular contact ball bearing for universal matching. Two bearings arranged back-to-back or face-to-face will have a “Normal” axial clearance before mounting
2. Controlled axial clearance of double row angular contact ball bearings

CC 1. Spherical roller bearing of C design but with enhanced roller guidance
2. Single row angular contact ball bearing for universal matching. Two bearings arranged back-to-back or face-to-face will have a large axial clearance before mounting

CLN Taper roller bearing with tolerances corresponding to ISO tolerance class 6X

CL0 Inch-size taper roller bearing with tolerances to class 0 according to ANSI/ABMA Standard 19.2:1994

CL00 Inch-size taper roller bearing with tolerances to class 00 according to ANSI/ABMA Standard 19.2:1994

CL3 Inch-size taper roller bearing with tolerances to class 3 according to ANSI/ABMA Standard 19.2:1994

CL7C Taper roller bearing with special frictional behaviour and heightened running accuracy

CN Normal internal clearance, normally only used together with an additional letter that identifies a reduced or displaced clearance range.
Examples:
CNH Upper half of the Normal clearance range
CNM Two middle quarters of the Normal clearance range
CNL Lower half of the Normal clearance range
CNP Upper half of the Normal and lower half of C3 clearance
CNR Cylindrical roller bearings with Normal clearance to DIN 620-4:1982
The above letters H, M, L and P are also used together with the following clearance classes: C2, C3 and C4

CV Full complement cylindrical roller bearing with modified internal design

CS Contact seal of nitrile butadiene rubber (NBR) with sheet steel reinforcement on one side of the bearing

CS2 Contact seal of fluoro rubber (FPM) with sheet steel reinforcement on one side of the bearing

CS5 Contact seal of hydrogenated nitrile butadiene rubber (HNBR) with sheet steel reinforcement on one side of the bearing

2CS Contact seals of nitrile butadiene rubber (NBR) with sheet steel reinforcement on both sides of the bearing

2CS2 Contact seals of fluoro rubber (FPM) with sheet steel reinforcement on both sides of the bearing

2CS5 Contact seals of hydrogenated nitrile butadiene rubber (HNBR) with sheet steel reinforcement on both sides of the bearing

C1 Bearing internal clearance smaller than C2

C2 Bearing internal clearance smaller than Normal (CN)

C3 Bearing internal clearance greater than Normal (CN)

C4 Bearing internal clearance greater than C3

C5 Bearing internal clearance greater than C4

C02 Extra reduced tolerance for running accuracy of inner ring of assembled bearing

C04 Extra reduced tolerance for running accuracy of outer ring of assembled bearing

C08 C02 + C04

C083 C02 + C04 + C3

C10 Reduced tolerance for the bore and outside diameters

D Deviating or modified internal design with the same boundary dimensions; as a rule the significance of the letter is bound to the particular bearing series. Example:
3310 D: Double row angular contact ball bearing with a two-piece inner ring

DA Modified snap ring grooves in the outer ring; two-piece inner ring held together by a retaining ring

DB Two single row deep groove ball bearings (1), single row angular contact ball bearings (2) or single row taper roller bearings matched for mounting in a back-to-back arrangement. The letter(s) following the DB indicate the magnitude of the axial clearance or preload in the bearing pair before mounting.
- A Light preload (2)
- B Moderate preload (2)
- C Heavy preload (2)
- CA Small axial clearance (1, 2)
- CB Normal axial clearance (1, 2)
- CC Large axial clearance (1, 2)
- C Special axial clearance in μm
- GA Light preload (1)
- GB Moderate preload (1)
- G Special preload in daN

For paired taper roller bearings the design and arrangement of the intermediate rings between the inner and outer rings are identified by a two-figure number which is placed between DB and the above mentioned letters.

DF Two single row deep groove ball bearings, single row angular contact ball bearings or single row taper roller bearings matched for mounting in a face-to-face arrangement. The letter(s) following the DF are explained under DB

DT Two single row deep groove ball bearings, single row angular contact ball bearings or single row taper roller bearings matched for mounting in a tandem arrangement; for paired taper roller bearings the design and arrangement of the intermediate rings between the inner and/or outer rings are identified by a two-figure number which follows immediately after DT

E Deviating or modified internal design with the same boundary dimensions; as a rule the significance of the letter is bound to the particular bearing series; usually indicates reinforced rolling element complement. Example:
7212 BE: Single row angular contact ball bearing with a 40° contact angle and optimized internal design

EC Single row cylindrical roller bearing with an optimized internal design and with modified roller end/flange contact

ECA Spherical roller bearing of CA design but with reinforced rolling element complement

ECAC Spherical roller bearing of CAC design but with reinforced rolling element complement

F Machined steel or special cast iron cage, rolling element centred; different designs or materials are identified by a figure following the F, e.g. F1

FA Machined steel or special cast iron cage; outer ring centred

FB Machined steel or special cast iron cage; inner ring centred

G Single row angular contact ball bearing for universal matching. Two bearings arranged back-to-back or face-to-face will have a certain axial clearance before mounting

G.. Grease filling. A second letter indicates the operating temperature range of the grease and a third letter identifies the actual grease. The significance of the second letter is as follows:

- E Extreme pressure grease
- F Food compatible grease
- H, J High temperature grease, –20 to +130 °C
- L Low temperature grease, – 50 to +80 °C
- M Medium temperature grease, –30 to +110 °C
- W, X Low/high temperature grease, –40 to +140 °C

A figure following the three-letter grease code indicates that the filling degree deviates from the standard: Figures 1, 2 and 3 indicate smaller than standard, 4 up to 9 a larger fill.
Examples:

- GEA Extreme pressure grease, standard fill
- GLB2 Low temperature grease, 15 to 25 % fill

GA Single row angular contact ball bearing for universal matching. Two bearings arranged back-to-back or face-to-face will have a light preload before mounting

GB Single row angular contact ball bearing for universal matching. Two bearings arranged back-to-back or face-to-face will have a moderate preload before mounting

GC Single row angular contact ball bearing for universal matching. Two bearings arranged back-to-back or face-to-face will have a heavy preload before mounting

GJN Normal fill grade of polyurea base grease of consistency 2 to the NLGI Scale for a temperature range –30 to +150 °C

H Pressed snap-type steel cage, hardened

HA Bearing or bearing components of case-hardening steel. For closer identification HA is followed by one of the following figures

- 0 Complete bearing
- 1 Outer and inner rings
- 2 Outer ring
- 3 Inner ring
- 4 Outer ring, inner ring and rolling elements
- 5 Rolling elements
- 6 Outer ring and rolling elements
- 7 Inner ring and rolling elements

HB Bainite hardened bearing or bearing components. For closer identification HB is followed by one of the figures explained under HA

HC Bearing or bearing components of ceramic material. For closer identification HC is followed by one of the figures explained under HA

HE Bearing or bearing components of vacuum remelted steel. For closer identification HE is followed by one of the figures explained under HA

HM Martensite hardened bearing or bearing components. For closer identification HM is followed by one of the figures explained under HA

HN Special surface heat-treated bearing or bearing components. For closer identification HN is followed by one of the figures explained under HA

HT Grease fill for high operating temperatures (–20 to +130 °C). Greases, which differ from the selected standard grease for this temperature range, are identified by two-figure numbers following HT. Filling degrees other than standard are identified by a letter or letter/figure combination following HTxx:

- A Filling degree less than standard
- B Filling degree greater than standard
- C Filling degree greater than 70 %
- F1 Filling degree less than standard
- F7 Filling degree greater than standard
- F9 Filling degree greater than 70 %

Examples: HTB, HT22 or HT24B

HV Bearing or bearing components of hardenable stainless steel. For closer identification HV is followed by one of the figures explained under HA

J Pressed steel cage, rolling element centred, unhardened; different designs or materials are identified by a figure, e.g. J1

JR Cage comprising of two flat washers of unhardened sheet steel, riveted together

K Tapered bore, taper 1:12

K30 Tapered bore, taper 1:30

LHT Grease fill for low and high operating temperatures (–40 to +140 °C). A two-figure number following LHT identifies the actual grease. An additional letter or letter/figure combination as mentioned under “HT” identifies filling degrees other than standard. Examples: LHT23, LHT23C or LHT23F7

LS Land-riding contact seal with or without sheet steel reinforcement on one side of the bearing

2LS Land-riding contact seals with or without sheet steel reinforcement on both sides of the bearing

LT Grease fill for low operating temperatures (–50 to +80 °C). Greases, which differ from the selected standard grease for this temperature range are identified as explained under “HT”. Examples: LT, LT10 or LTF1

L4B Bearing rings and rolling elements with special surface coating

L5B Rolling elements with special surface coating

L5DA NoWear bearing with coated rolling elements

L7DA NoWear bearing with coated rolling elements and inner ring raceway(s)

M Machined brass cage, rolling element centred; different designs or materials are identified by a figure, e.g. M2

MA Machined brass cage, outer ring centred

MB Machined brass cage, inner ring centred

ML One-piece brass window-type cage, inner or outer ring centred

MP One-piece brass window-type cage, with punched or reamed pockets, inner or outer ring centred

MR One-piece brass window-type cage, rolling element centred

MT Grease fill for medium operating temperatures (–30 to +110 °C). A two-figure number following MT identifies the actual grease. An additional letter or letter/figure combination as mentioned under “HT” identifies filling degrees other than standard. Examples: MT33, MT37F9 or MT47

N Snap ring groove in the outer ring

NR Snap ring groove in the outer ring with appropriate snap ring

N1 One locating slot (notch) in one outer ring side face

N2 Two locating slots (notches) in one outer ring side face at 180° to each other

P Injection moulded cage of glass fibre reinforced polyamide 6,6, rolling element centred

PH Injection moulded cage of polyether ether ketone (PEEK), rolling element centred

PHA Injection moulded cage of polyether ether ketone (PEEK), outer ring centred

P4 Dimensional and running accuracy to ISO tolerance class 4

P5 Dimensional and running accuracy to ISO tolerance class 5

P6 Dimensional and running accuracy to ISO tolerance class 6

P62 P6 + C2

P63 P6 + C3

Q Optimized internal geometry and surface finish (taper roller bearing)

R 1. Flanged outer ring
2. Crowned runner surface (track runner bearing)

RS Contact seal of synthetic rubber with or without sheet steel reinforcement on one side of the bearing

RS1 Contact seal of acrylonitrile butadiene rubber (NBR) with sheet steel reinforcement on one side of the bearing

RS1Z Contact seal of acrylonitrile butadiene rubber (NBR) with sheet steel reinforcement on one side and one shield on the other side of the bearing

RS2 Contact seal of fluoro rubber (FPM) with sheet steel reinforcement on one side of the bearing

RSH Contact seal of acrylonitrile butadiene rubber (NBR) with sheet steel reinforcement on one side of the bearing

RSL Low-friction contact seal of acrylonitrile butadiene rubber (NBR) with sheet steel reinforcement on one side of the bearing

RZ Low-friction seal of acrylonitrile butadiene rubber (NBR) with sheet steel reinforcement on one side of the bearing

2RS Contact seals of synthetic rubber with sheet steel reinforcement on both sides of the bearing

2RS1 Contact seals of acrylonitrile butadiene rubber (NBR) with sheet steel reinforcement on both sides of the bearing

2RS2 Contact seals of fluoro rubber (FPM) with sheet steel reinforcement on both sides of the bearing

2RSH Contact seals of acrylonitrile butadiene rubber (NBR) with sheet steel reinforcement on both sides of the bearing

2RSL Low-friction contact seals of acrylonitrile butadiene rubber (NBR) with sheet steel reinforcement on both sides of the bearing

2RZ Low-friction seals of acrylonitrile butadiene rubber (NBR) with sheet steel reinforcement on both sides of the bearing

S0 Bearing rings or washers dimensionally stabilized for use at operating temperatures up to +150 °C

S1 Bearing rings or washers dimensionally stabilized for use at operating temperatures up to +200 °C

S2 Bearing rings or washers dimensionally stabilized for use at operating temperatures up to +250 °C

S3 Bearing rings or washers dimensionally stabilized for use at operating temperatures up to +300 °C

S4 Bearing rings or washers dimensionally stabilized for use at operating temperatures up to +350 °C

T Machined cage of fabric reinforced phenolic resin, rolling element centred

TB Window-type cage of fabric reinforced phenolic resin, inner ring centred

TH Snap-type cage of fabric reinforced phenolic resin, rolling element centred

TN Injection moulded cage of polyamide, rolling element centred

TNH Injection moulded cage of polyether ether ketone (PEEK), rolling element centred

TN9 Injection moulded cage of glass fibre reinforced polyamide 6,6, rolling element centred

U U combined with a one-figure number identifies a taper roller bearing, cone or cup, with reduced width tolerance. Examples:
U2: Width tolerance +0,05/0 mm
U4: Width tolerance +0,10/0 mm

V Full complement bearing (without cage)

V… V combined with a second letter, identifies a variant group, and followed by a three or four figure number denotes variants not covered by "standard" designation suffixes. Examples:
- VA Application oriented variants
- VB Boundary dimension deviations
- VE External or internal deviations
- VL Coatings
- VQ Quality and tolerances other than standard
- VS Clearance and preload
- VT Lubrication
- VU Miscellaneous applications

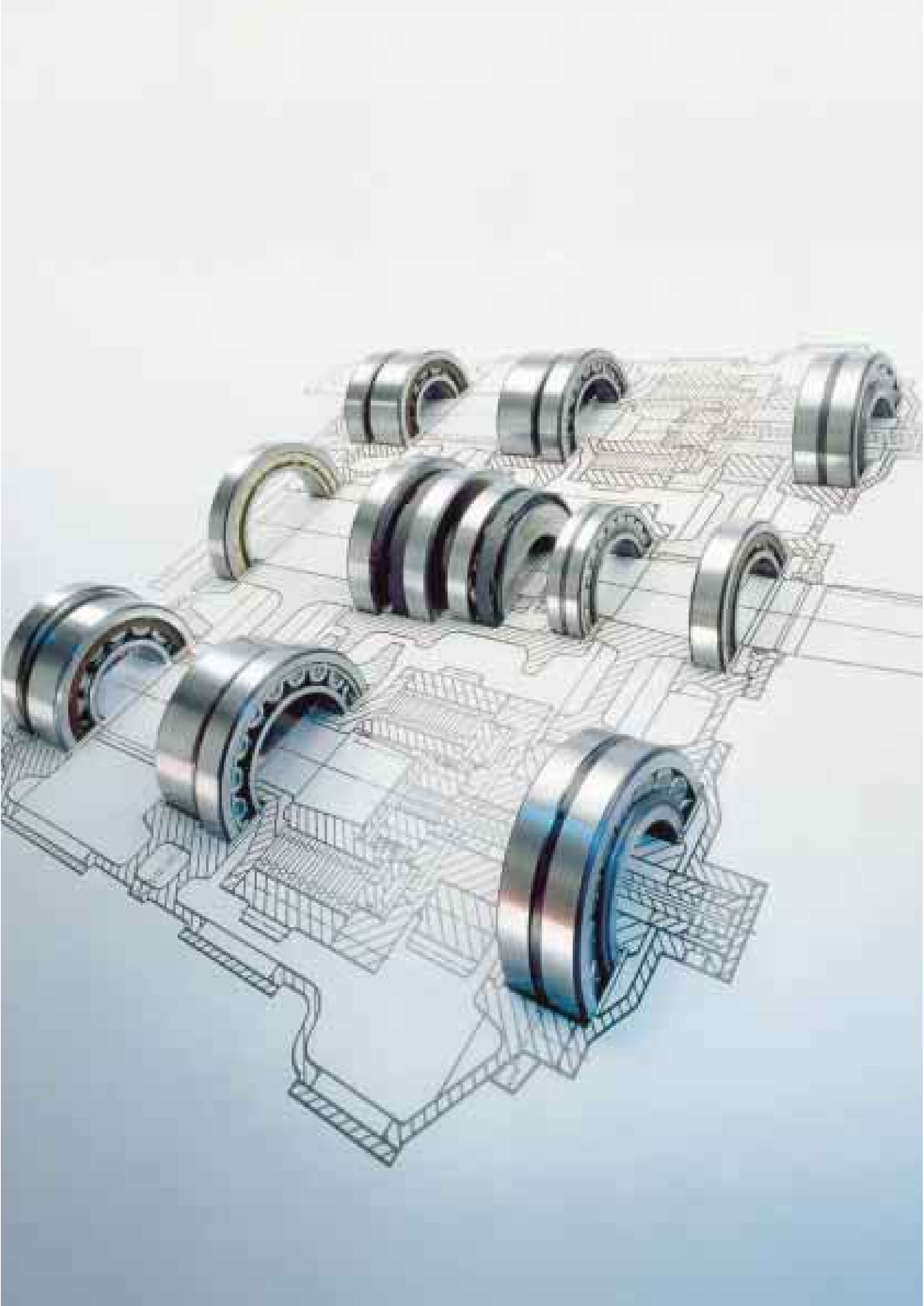
VA201 Bearing for high-temperature applications (e.g. kiln trucks)

VA208 Bearing for high-temperature applications

VA228 Bearing for high-temperature applications

VA301 Bearing for traction motors
VA305 VA301 + special inspection routines
VA3091 VA301 + VL0241
VA350 Bearing for railway axleboxes
VA405 Bearing for vibratory applications
VA406 Bearing for vibratory applications with special PTFE bore coating
VA820 Bearing for railway axleboxes according to EN 12080:1998
VC025 Bearing with specially debrisheat-treated components for applications in heavily contaminated environments
VE240 CARB bearing modified for greater axial displacement
VE447 Shaft washer with three equally spaced threaded holes in one side face to accommodate hoisting tackle
VE552 Outer ring with three equally spaced threaded holes in one side face to accommodate hoisting tackle
VE553 Outer ring with three equally spaced threaded holes in both side faces to accommodate hoisting tackle
VE632 Housing washer with three equally spaced threaded holes in one side face to accommodate hoisting tackle
VG114 Surface hardened pressed steel cage
VH Full complement cylindrical roller bearing with self-retaining roller set
VL0241 Aluminium oxide coated outside surface of outer ring for electrical resistance up to 1 000 V DC
VL2071 Aluminium oxide coated outside surface of inner ring for electrical resistance up to 1 000 V DC
VQ015 Inner ring with crowned raceway for increased permissible misalignment
VQ424 Running accuracy better than C08
VT143 Grease fill with an extreme pressure grease
W Without annular groove and lubrication holes in outer ring
WT Grease fill for low as well as high operating temperatures (–40 to +160 °C). Greases, which differ from the selected standard grease for this temperature range are identified as explained under "HT" Examples: WT or WTF1
W20 Three lubrication holes in the outer ring
W26 Six lubrication holes in the inner ring
W33 Annular groove and three lubrication holes in the outer ring
W513 Six lubrication holes in the inner ring and annular groove and three lubrication holes in the outer ring
W64 "Solid Oil" fill
W77 Plugged W33 lubrication holes
X
1. Boundary dimensions altered to conform to ISO standards
2. Cylindrical runner surface (track runner bearing)

Y Pressed brass cage, rolling element centred; different designs or materials are identified by a figure following the Y, e.g. Y1
Z Shield of pressed sheet steel on one side of the bearing
2Z Shields of pressed sheet steel on both sides of the bearing



Application of bearings

Bearing arrangements

The bearing arrangement of a rotating machine component, e.g. a shaft, generally requires two bearings to support and locate the component radially and axially relative to the stationary part of the machine, such as a housing. Depending on the application, load, requisite running accuracy and cost considerations the arrangement may consist of

- locating and non-locating bearing arrangements,
- adjusted bearing arrangements, or
- "floating" bearing arrangements.

Bearing arrangements consisting of a single bearing which can support radial, axial and moment loads, e.g. for an articulated joint, are not dealt with in this catalogue. If such arrangements are required it is advisable to contact the SKF application engineering service.

Locating and non-locating bearing arrangements
The locating bearing at one end of the shaft provides radial support and at the same time locates the shaft axially in both directions. It must, therefore, be fixed in position both on the shaft and in the housing. Suitable bearings are radial bearings which can accommodate combined loads, e.g. deep groove ball bearings, double row or paired single row angular contact ball bearings, self-aligning ball bearings, spherical roller bearings or matched taper roller bearings. Combinations of a radial bearing that can accommodate purely radial load, e.g. a cylindrical roller bearing having one ring without flanges, with a deep groove ball bearing, four-point contact ball bearing or a double direction thrust bearing can also be used as the locating bearing. The second bearing then provides axial location in both directions but must be mounted with radial freedom (i.e. have a clearance fit) in the housing.

The non-locating bearing at the other end of the shaft provides radial support only. It must also allow axial displacement so that the bearings do not mutually stress each other, e.g. when the shaft length changes as a result of thermal expansion. Axial displace-

Fig 1

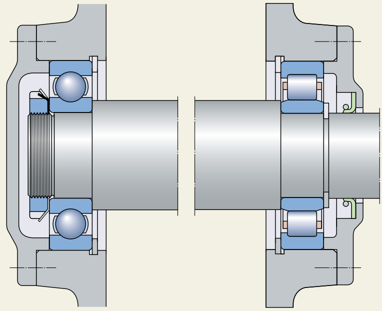


Fig 2

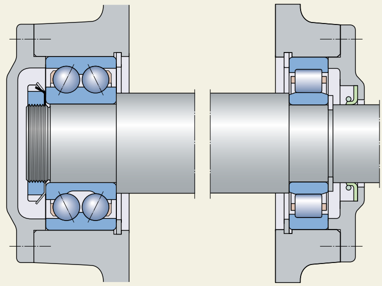


Fig 3

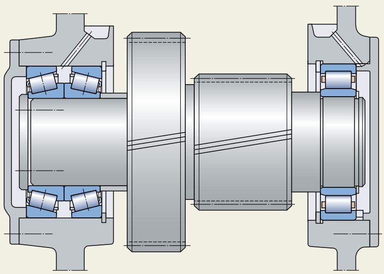


Fig 4

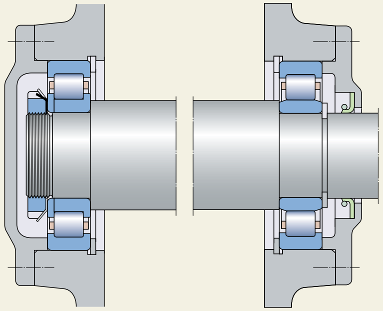


Fig 5

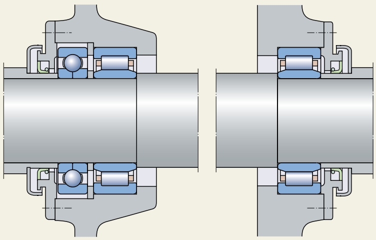
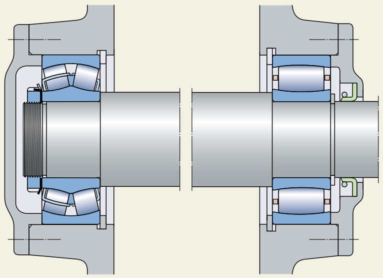


Fig 6



ment can take place within the bearing in the case of needle roller bearings, NU- and N-design cylindrical roller bearings and CARB toroidal roller bearings, or between one of the bearing rings and its seating, preferably between the outer ring and its seating in the housing bore.

From the large number of locating/non-locating bearing combinations the popular combinations are described in the following.

For stiff bearing arrangements where “frictionless” axial displacements should take place within the bearing the following combinations may be used:

- deep groove ball bearing/cylindrical roller bearing (➔ **fig** 1),
- double row angular contact ball bearing/cylindrical roller bearing (➔ **fig** 2),
- matched single row taper roller bearings/cylindrical roller bearing (➔ **fig** 3),
- NUP-design cylindrical roller bearing/NU-design cylindrical roller bearing (➔ **fig** 4), or
- NU-design cylindrical roller bearing and four-point contact ball bearing/NU-design cylindrical roller bearing (➔ **fig** 5).

For the above combinations, angular misalignment of the shaft must be kept to a minimum. If this is not possible it is advisable to use combinations of self-aligning bearings to allow for misalignment, viz.

- self-aligning ball bearing/CARB toroidal roller bearing or
- spherical roller bearing/CARB toroidal roller bearing (➔ **fig** 6).

The abillity of these arrangements to accommodate angular misalignments as well as axial displacements avoids generating internal axial forces in the bearing system.

For bearing arrangements with rotating inner ring load, where changes in the shaft length are to be accommodated between the bearing and its seating, axial displacement should take place between the outer ring of the bearing and the housing. The most usual combinations are

- deep groove ball bearing/deep groove ball bearing (➔ **fig** 7),
- self-aligning ball or spherical roller bearing/self-aligning ball or spherical roller bearing (➔ **fig** 8) and
- matched single row angular contact ball bearings/deep groove ball bearing (➔ **fig** 9).

Adjusted bearing arrangements
In adjusted bearing arrangements the shaft is axially located in one direction by the one bearing and in the opposite direction by the other bearing. This type of arrangement is referred to as "cross located" and is generally used for short shafts. Suitable bearings include all types of radial bearings that can accommodate axial loads in at least one direction, including

- angular contact ball bearings (➔ **fig** 10) and
- taper roller bearings (➔ **fig** 11).

In certain cases where single row angular contact ball bearings or taper roller bearings are used for cross-located arrangements, preload may be necessary (➔ **page 206**).

"Floating" bearing arrangements
Floating bearing arrangements are also cross located and are suitable where demands regarding axial location are moderate or where other components on the shaft serve to locate it axially.

Suitable bearings for this type of arrangement are:

- deep groove ball bearings (➔ **fig** 12),
- self-aligning ball bearings or
- spherical roller bearings.

In these types of arrangements it is important that one ring of each bearing should be able to move on or in its seating, preferably the outer ring in the housing. A floating bearing arrangement can also be obtained with two NJ-design cylindrical roller bearings, with offset inner rings (➔ **fig** 13). In this case the axial movement can take place within the bearing.

Fig 7

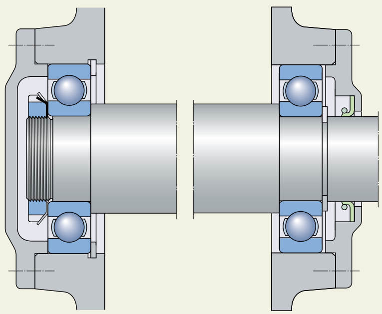


Fig 8

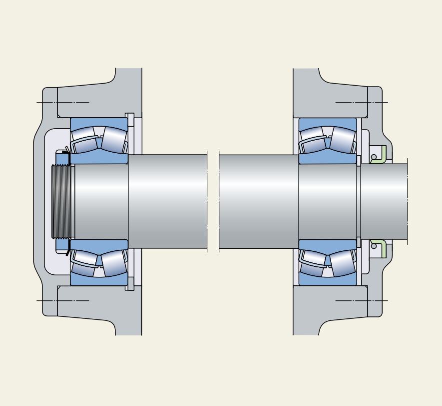


Fig 11

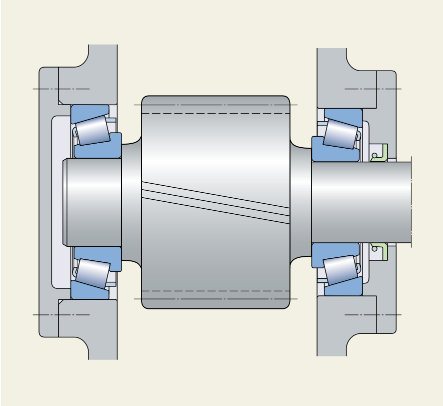


Fig 9

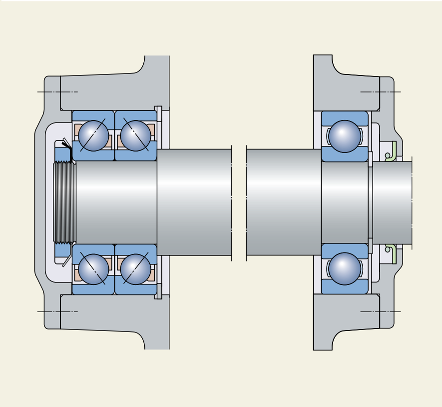


Fig 12

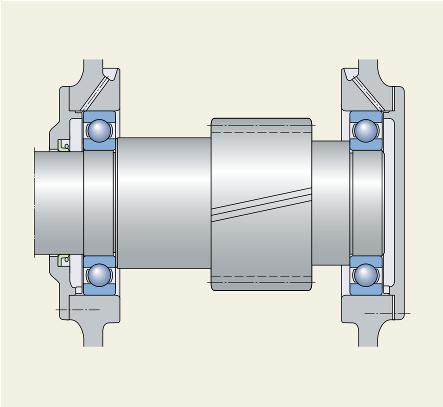


Fig 10

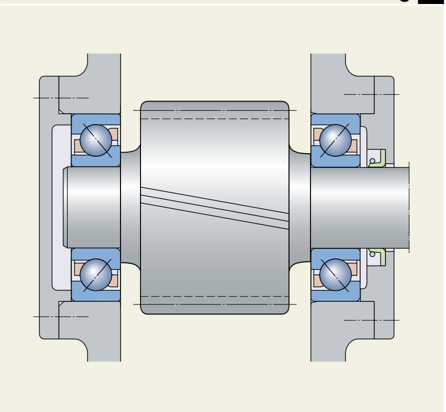
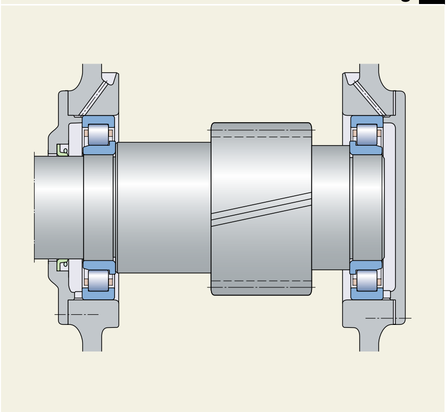


Fig 13



Radial location of bearings

If the load carrying ability of a bearing is to be fully utilized, its rings or washers must be fully supported around their complete circumference and across the entire width of the raceway. The support, which must be firm and even can be provided by a cylindrical or tapered seating or, for thrust bearing washers, by a flat (plane) support surface. This means that bearing seatings must be made with adequate accuracy and that their surface should be uninterrupted by grooves, holes or other features. In addition, the bearing rings must be reliably secured to prevent them from turning on or in their seatings under load.

Generally speaking, satisfactory radial location and adequate support can only be obtained when the rings are mounted with an appropriate degree of interference. Inadequately or incorrectly secured bearing rings generally cause damage to the bearings and associated components. However, when easy mounting and dismounting are desirable, or when axial displacement is required with a non-locating bearing, an interference fit cannot always be used. In certain cases where a loose fit is employed it is necessary to take special precautions to limit the inevitable wear from creep, as for example, by surface hardening of the bearing seating and abutments, lubrication of the mating surfaces via special lubrication grooves and the removal of wear particles, or slots in the bearing ring side faces to accommodate keys or other holding devices.

Selection of fit
When selecting a fit, the factors discussed in this section should be considered, together with the general guidelines given.

1. Conditions of rotation

Conditions of rotation refer to the bearing ring being considered in relation to the direction of the load (➔ **table 1**). Essentially there are three different conditions: “rotating load”, “stationary load” and “direction of load indeterminate”.

“Rotating load” pertains if the bearing ring rotates and the load is stationary, or if the ring is stationary and the load rotates so that all points on the raceway are subjected to load in the course of one revolution. Heavy loads which do not rotate but oscillate, for example, those acting on connecting rod bearings, are generally considered as rotating loads.

A bearing ring subjected to a rotating load will turn (creep or wander) on its seating if mounted with a clearance fit, and wear (fretting corrosion) of the contact surfaces will result. To prevent this, interference fits must be used. The degree of interference needed is dictated by the operating conditions (➔ **points 2** and **4** below).

“Stationary load” pertains if the bearing ring is stationary and the load is also stationary, or if the ring and the load rotate at the same speed, so that the load is always directed towards the same position on the raceway. Under these conditions, a bearing ring will normally not turn on its seating. Therefore, the ring need not necessarily have an interference fit unless this is required for other reasons.

“Direction of load indeterminate” represents variable external loads, shock loads, vibrations and unbalance loads in high-speed machines. These give rise to changes in the direction of load, which cannot be accurately described. When the direction of load is indeterminate and particularly where heavy loads are involved, it is desirable that both rings have an interference fit. For the inner ring the recommended fit for a rotating load is normally used. However, when the outer ring must be free to move axially in the housing, and the load is not heavy, a somewhat looser fit than that recommended for a rotating load may be used.

2. Magnitude of the load

The interference fit of a bearing inner ring on its seating will be loosened with increasing load, as the ring will deform. Under the influence of rotating load the ring may begin to creep. The degree of interference must therefore be related to the magnitude of the load; the heavier the load, particularly if it is a shock load, the greater the interference fit required (→ **fig** 14).

Fig 14

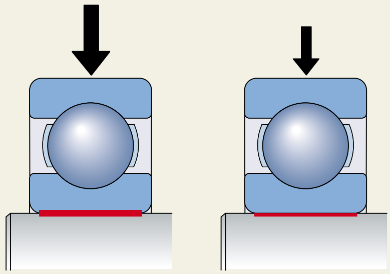
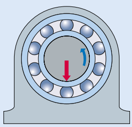
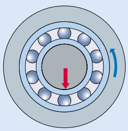
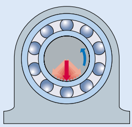
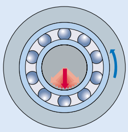


Table 1

Conditions of rotation and loading

Operating conditions	**Schematic illustration**	**Load condition**	**Example**	**Recommended fits**
Rotating inner ring Stationary outer ring Constant load direction		Rotating load on inner ring Stationary load on outer ring	Belt-driven shafts	Interference fit for inner ring Loose fit for outer ring
Stationary inner ring Rotating outer ring Constant load direction		Stationary load on inner ring Rotating load on outer ring	Conveyor idlers Car wheel hub bearings	Loose fit for inner ring Interference fit for outer ring
Rotating inner ring Stationary outer ring Load rotates with inner ring		Stationary load on inner ring Rotating load on outer ring	Vibratory applications Vibrating screens or motors	Interference fit for outer ring Loose fit for inner ring
Stationary inner ring Rotating outer ring Load rotates with outer ring		Rotating load on inner ring. Stationary load on outer ring	Gyratory crusher (Merry-go-round drives)	Interference fit for inner ring Loose fit for outer ring

3. Bearing internal clearance

An interference fit of a bearing on a shaft or in a housing means that the ring is elastically deformed (expanded or compressed) and that the bearing internal clearance is reduced. A certain minimum clearance should remain, however (➔ section “Bearing internal clearance”, starting on **page 137**). The initial clearance and permissible reduction depend on the type and size of the bearing. The reduction in clearance due to the interference fit can be so large that bearings with an initial clearance, which is greater than Normal, have to be used in order to prevent the bearing from becoming preloaded (➔ **fig** 15).

4. Temperature conditions

In many applications the outer ring has a lower temperature in operation than the inner ring. This might lead to reduced internal clearance (➔ **fig** 16).

In service, bearing rings normally reach a temperature that is higher than that of the components to which they are fitted. This can result in an easing of the fit of the inner ring on its seating, while outer ring expansion may prevent the desired axial displacement of the ring in its housing.

Temperature differentials and the direction of heat flow in the bearing arrangement must therefore be carefully considered.

5. Running accuracy requirements

To reduce resilience and vibration, clearance fits should generally not be used for bearings where high demands are placed on running accuracy. Bearing seatings on the shaft and in the housing should be made to narrow dimensional tolerances, corresponding at least to grade 5 for the shaft and at least to grade 6 for the housing. Tight tolerances should also be applied to the cylindricity (➔ **table** 11, **page 196**).

6. Design and material of shaft and housing

The fit of a bearing ring on its seating must not lead to uneven distortion of the ring (out-of-round). This can be caused, for example,

Fig 15

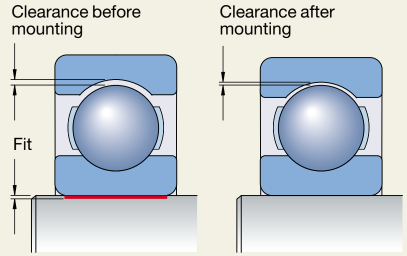
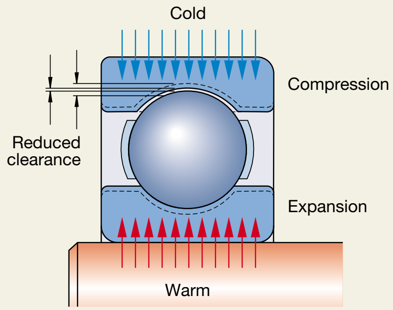


Fig 16



by discontinuities in the seating surface. Split housings are therefore not generally suitable where outer rings are to have a heavy interference fit and the selected tolerance should not give a tighter fit than that obtained with tolerance group H (or at the most K). To provide adequate support for bearing rings mounted in thin-walled housings, light alloy housings or on hollow shafts, heavier interference fits than those normally recommended for thick-walled steel or cast iron housings or for solid shafts should be used (➔ section "Fits for hollow shafts", starting on **page 172**).

7. Ease of mounting and dismounting

Bearings with clearance fits are usually easier to mount or dismount than those with interference fits. Where operating conditions necessitate interference fits and it is essential that mounting and dismounting can be done easily, separable bearings, or bearings with a tapered bore may be used. Bearings with a tapered bore can be mounted either directly on a tapered shaft seating or via adapter or withdrawal sleeves on smooth or stepped cylindrical shafts (➔ **figs** 26 and 27, **page 201**).

8. Displacement of the non-locating bearing

If non-separable bearings are used as non-locating bearings it is imperative that one of the bearing rings is free to move axially at all times during operation. Adopting a clearance fit for the ring that carries a stationary load will provide this (➔ **fig** 20, **page 199**). When the outer ring is under stationary load so that axial displacement is accommodated or takes place in the housing bore, a hardened intermediate bushing or sleeve is often fitted to the outer ring, for example, where light alloy housings are employed. In this way a "hammering out" of the housing seating because of the lower material hardness is avoided; it would otherwise result in the axial displacement being restricted or even prohibited over time.

If cylindrical roller bearings having one ring without flanges, needle roller bearings or CARB toroidal roller bearings are used, both bearing rings may be mounted with an interference fit because axial displacement will take place within the bearing.

Recommended fits

The tolerances for the bore and outside diameters of rolling bearings are internationally standardized (➔ section "Tolerances", starting on **page 120**).

To achieve an interference or a clearance fit for bearings with a cylindrical bore and cylindrical outside diameter, suitable tolerance ranges for the seatings on the shaft and in the housing bore are selected from the ISO tolerance system. Only a limited number of ISO tolerance grades need be considered for rolling bearing applications. The location of the most commonly used grades relative to the bearing bore and outside diameter tolerances are illustrated in **fig** 17**, page 168.**

Bearings with a tapered bore are mounted either directly on tapered shaft seatings or on adapter or withdrawal sleeves, having an external taper, which are fitted to cylindrical shaft seatings. In these cases, the fit of the bearing inner ring is not determined, as for bearings with a cylindrical bore, by the selected shaft tolerance but by the distance through which the ring is driven up on its tapered seating or sleeve. Special precautions with respect to the reduction of the internal clearance must be observed as mentioned in the sections "Self-aligning ball bearings", "Spherical roller bearings" and "CARB toroidal roller bearings".

If the bearings are to be secured using adapter or withdrawal sleeves, larger diameter tolerances are permitted for the sleeve seating, but the tolerances for cylindricity must be reduced (➔ section "Dimensional, form and running accuracy of bearing seatings and abutments", starting on **page 194**).

Tables with recommendations for fits
Recommendations for bearing fits for solid steel shafts will be found in:

Table 2: Radial bearings with cylindrical bore
Table 3: Thrust bearings

and for cast iron and steel housings in:

Table 4: Radial bearings – non-split housings
Table 5: Radial bearings – split or non-split housings
Table 6: Thrust bearings

These recommendations are based on the general selection guidelines described above. Years of experience have shown the recommendations to be correct for a very wide range of applications and bearing arrangements. The tables of housing tolerance recommendations also give information as to whether the outer ring can be axially displaced in the housing bore. Using this information it is possible to check that the chosen tolerance is suitable for non-separable bearings that are to be used as non-locating bearings.

Fig 17

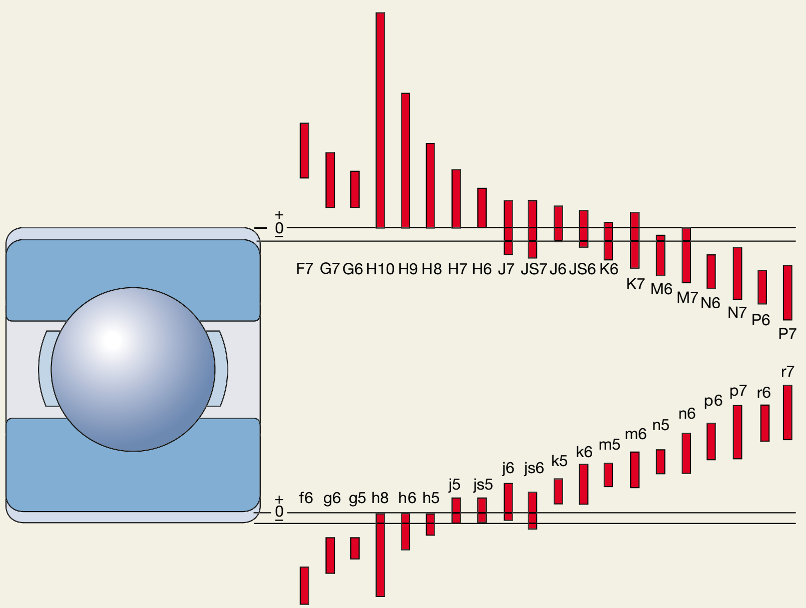


Table 2

Fits for solid steel shafts

Radial bearings with cylindrical bore

Conditions	Examples	Shaft diameter, mm Ball bearings	 Cylindrical and taper roller bearings	 CARB and spherical roller bearings	Tolerance
Rotating inner ring load or direction of load indeterminate					
Light and variable loads (P ≤ 0,06 C)	Conveyors, lightly loaded gearbox bearings	(18) to 100 (100) to 140	≤ 40 (40) to 100	–	j6 k6
Normal and heavy loads (P > 0,06 C)	Bearing applications generally, electric motors, turbines, pumps, internal combustion engines, gearing, woodworking machines	≤ 18 (18) to 100 (100) to 140 (140) to 200 (200) to 280 – – –	– ≤ 40 (40) to 100 (100) to 140 (140) to 200 (200) to 400 – –	– ≤ 40 (40) to 65 (65) to 100 (100) to 140 (140) to 280 (280) to 500 > 500	j5 k5 (k6)[1] m5 (m6)[1] m6 n6 p6 r6[2] r7[2]
Very heavy loads and shock loads with difficult working conditions (P > 0,12 C)	Axleboxes for heavy railway vehicles, traction motors, rolling mills	– – –	(50) to 140 (140) to 200 > 200	(50) to 100 (100) to 140 > 140	n6[2] p6[2] r6[2]
High demands on running accuracy with light loads (P ≤ 0,06 C)	Machine tools	8 to 240 – – – –	– 25 to 40 (40) to 140 (140) to 200 (200) to 500	– – – – –	js4 js4 (j5)[3] k4 (k5)[3] m5[3] n5[3]
Stationary inner ring load					
Easy axial displacement of inner ring on shaft desirable	Wheels on non-rotating axles				g6[4]
Easy axial displacement of inner ring on shaft unnecessary	Tension pulleys, rope sheaves				h6
Axial loads only					
	Bearing applications of all kinds	≤ 250 > 250	≤250 > 250	≤ 250 > 250	j6 js6

1) The tolerances in brackets are generally used for taper roller bearings and single row angular contact ball bearings, they can also be used for other types of bearing where speeds are moderate and the effect of bearing internal clearance variation is not significant

2) Bearings with radial internal clearance greater than Normal may be necessary

3) The tolerances in brackets apply to taper roller bearings. For lightly loaded taper roller bearings adjusted via the inner ring, js5 or js6 should be used

4) Tolerance f6 can be selected for large bearings to provide easy displacement

Table 3

Fits for solid steel shafts

Thrust bearings

Conditions	Shaft diameter, mm	Tolerance
Axial loads only		
Thrust ball bearings	–	h6
Cylindrical roller thrust bearings	–	h6 (h8)
Cylindrical roller and cage thrust assemblies	–	h8
Combined radial and axial loads acting on spherical roller thrust bearings		
Stationary load on shaft washer	≤ 250	j6
	> 250	js6
Rotating load on shaft washer, or direction of load indeterminate	≤200	k6
	(200) to 400	m6
	> 400	n6

Table 4

Fits for cast iron and steel housings

Radial bearings – non-split housings

Conditions	Examples	Tolerance	Displacement of outer ring
Rotating outer ring load			
Heavy loads on bearings in thin-walled housings, heavy shock loads (P > 0,12 C)	Roller bearing wheel hubs, big-end bearings	P7	Cannot be displaced
Normal and heavy loads (P > 0,06 C)	Ball bearing wheel hubs, big-end bearings, crane travelling wheels	N7	Cannot be displaced
Light and variable loads (P ≤ 0,06 C)	Conveyor rollers, rope sheaves, belt tensioner pulleys	M7	Cannot be displaced
Direction of load indeterminate			
Heavy shock loads	Electric traction motors	M7	Cannot be displaced
Normal and heavy loads (P > 0,06 C), axial displacement of outer ring unnecessary	Electric motors, pumps, crankshaft bearings	K7	Cannot be displaced as a rule
Accurate or quiet running[1]			
Ball bearings	Small electric motors	J6[2]	Can be displaced
Taper roller bearings	When adjusted via the outer ring	JS5	–
	Axially located outer ring	K5	–
	Rotating outer ring load	M5	–

[1] For high-precision bearings to tolerance class P5 or better, other recommendations apply (➔ the SKF catalogue “High-precision bearings”)
[2] When easy displacement is required use H6 instead of J6

Table 5

Fits for cast iron and steel housings

Radial bearings – split or non-split housings

Conditions	Examples	Tolerance	Displacement of outer ring
Direction of load indeterminate			
Light and normal loads ($P \leq 0{,}12$ C), axial displacement of outer ring desirable	Medium-sized electrical machines, pumps, crankshaft bearings	J7	Can be displaced as a rule
Stationary outer ring load			
Loads of all kinds	General engineering, railway axleboxes	H7[1)]	Can be displaced
Light and normal loads ($P \leq 0{,}12$ C) with simple working conditions	General engineering	H8	Can be displaced
Heat conduction through shaft	Drying cylinders, large electrical machines with spherical roller bearings	G7[2)]	Can be displaced

1) For large bearings (D > 250 mm) and temperature differences between outer ring and housing > 10 °C, G7 should be used instead of H7

2) For large bearings (D > 250 mm) and temperature differences between outer ring and housing > 10 °C, F7 should be used instead of G7

Table 6

Fits for cast iron and steel housings

Thrust bearings

Conditions	Tolerance	Remarks
Axial loads only		
Thrust ball bearings	H8	For less accurate bearing arrangements there can be a radial clearance of up to 0,001 D
Cylindrical roller thrust bearings	H7 (H9)	
Cylindrical roller and cage thrust assemblies	H10	
Spherical roller thrust bearings where separate bearings provide radial location	–	Housing washer must be fitted with adequate radial clearance so that no radial load whatsoever can act on the thrust bearings
Combined radial and axial loads on spherical roller thrust bearings		
Stationary load on housing washer	H7	See also “Design of associated components” in section “Spherical roller thrust bearings” on **page 877**
Rotating load on housing washer	M7	

Tolerance tables

The values shown in **tables** 7 and 8 for the shaft and housing tolerances enable the character of the fit to be established:

- the upper and lower limits of Normal tolerances for the bearing bore and outside diameter deviations;
- the upper and lower limits of the shaft and housing bore diameter deviations in accordance with ISO 286-2:1988;
- the smallest and largest values of the theoretical interference (+) or clearance (–) in the fit;
- the smallest and largest values of the probable interference (+) or clearance (–) in the fit.

The appropriate values for rolling bearing seatings on shafts are listed for the tolerances:

e7, f5, f6, g5, g6 in **table** 7a, **pages 174** and **175**
h5, h6, h8, h9, j5 in **table** 7b, **pages 176** and **177**
j6, js5, js6, js7, k4 in **table** 7c, **pages 178** and **179**
k5, k6, m5, m6, n5 in **table** 7d, **pages 180** and **181**
n6, p6, p7, r6, r7 in **table** 7e, **pages 182** and **183**

The appropriate values for the rolling bearing housing seatings are listed for the tolerances:

F7, G6, G7, H5, H6 in **table** 8a, **pages 184** and **185**
H7, H8, H9, H10, J6 in **table** 8b, **pages 186** and **187**
J7, JS5, JS6, JS7, K5 in **table** 8c, **pages 188** and **189**
K6, K7, M5, M6, M7 in **table** 8d, **pages 190** and **191**
N6, N7, P6, P7 in **table** 8e, **pages 192** and **193**

The Normal tolerances for the bore and outside diameter for which the limiting values have been calculated are valid for all metric rolling bearings with the exception of metric taper roller bearings when $d \leq 30$ mm and $D \leq 150$ mm and for thrust bearings when $D \leq 150$ mm.

The values for the probable interference or clearance cover 99 % of all the combinations of the theoretical interference or clearance.

When bearings of higher accuracy than Normal are used, the reduced bore and outside tolerances mean that the interference or clearance of the fits is reduced correspondingly. If, in such cases, a more accurate calculation of the limits is required it is advisable to contact the SKF application engineering service.

Fits for hollow shafts

If bearings are to be mounted with an interference fit on a hollow shaft it is generally necessary to use a heavier interference fit than would be used for a solid shaft in order to achieve the same surface pressure between the inner ring and shaft seating. The following diameter ratios are important when deciding on the fit to be used:

$$c_i = \frac{d_i}{d} \text{ and } c_e = \frac{d}{d_e}$$

The fit is not appreciably affected until the diameter ratio of the hollow shaft $c_i \geq 0,5$. If the outside diameter of the inner ring is not known, the diameter ratio c_e can be calculated with sufficient accuracy using the equation

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

where
c_i = diameter ratio of the hollow shaft
c_e = diameter ratio of the inner ring
d = outside diameter of the hollow shaft, bore diameter of bearing, mm
d_i = internal diameter of the hollow shaft, mm
d_e = outside diameter of the inner ring, mm
D = outside bearing diameter, mm
k = a factor for the bearing type
for self-aligning ball bearings in the 22 and 23 series, $k = 0,25$
for cylindrical roller bearings, $k = 0,25$
for all other bearings, $k = 0,3$

To determine the requisite interference fit for a bearing to be mounted on a hollow shaft, use the mean probable interference between the shaft seating and bearing bore obtained for the tolerance recommendation for a solid shaft of the same diameter. If the plastic deformation (smoothing) of the mating surfaces produced during mounting is neglected, then the effective interference can be equated to the mean probable interference.

The interference Δ_H needed for a hollow steel shaft can then be determined in relation to the known interference Δ_V for the solid shaft from **diagram 1**. Δ_V equals the mean value of the smallest and largest values of the probable interference for the solid shaft. The tolerance for the hollow shaft is then selected so that the mean probable interference is as close as possible to the interference Δ_H obtained from **diagram 1**.

Example

A 6208 deep groove ball bearing with $d = 40$ mm and $D = 80$ mm is to be mounted on a hollow shaft having a diameter ratio $c_i = 0,8$. What is the requisite interference and what are the appropriate shaft limits?

If the bearing were to be mounted on a solid steel shaft and subjected to normal loads, a tolerance k5 would be recommended. From **table 7d**, **page 180**, a shaft diameter of 40 mm, the mean probable interference $\Delta_V = (22 + 5)/2 = 13,5$ µm. For $c_i = 0,8$ and

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

so that from **diagram 1** the ratio $\Delta_H/\Delta_V = 1,7$. Thus the requisite interference for the hollow shaft $\Delta_H = 1,7 \times 13,5 = 23$ µm. Consequently, tolerance m6 is selected for the hollow shaft as this gives a mean probable interference of this order.

Diagram 1

Relation of interference Δ_H, needed for a hollow steel shaft, to the known interference Δ_V for a solid steel shaft

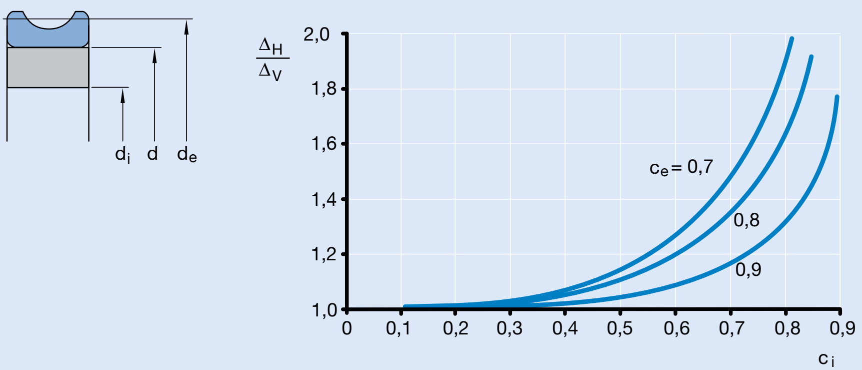
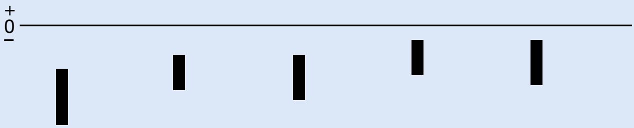


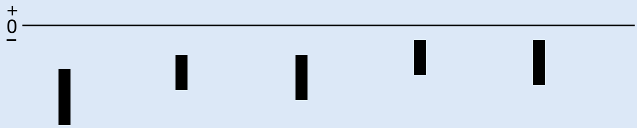
Table 7a

Shaft tolerances and resultant fits



Shaft Nominal diameter d		Bearing Bore diameter tolerance Δ_{dmp}		Deviations of shaft diameter, resultant fits Tolerances									
				e7		f5		f6		g5		g6	
over	incl.	low	high	Deviations (shaft diameter) Theoretical interference (+)/clearance (–) Probable interference (+)/clearance (–)									
mm		µm		µm									
1	**3**	–8	0	–14	–24	–6	–10	–6	–12	–2	–6	–2	–8
				–6	–24	+2	–10	+2	–12	+6	–6	+6	–8
				–8	–22	+1	–9	0	–10	+5	–5	+4	–6
3	**6**	–8	0	–20	–32	–10	–15	–10	–18	–4	–9	–4	–12
				–12	–32	–2	–15	–2	–18	+4	–9	+4	–12
				–14	–30	–3	–14	–4	–16	+3	–8	+2	–10
6	**10**	–8	0	–25	–40	–13	–19	–13	–22	–5	–11	–5	–14
				–17	–40	–5	–19	–5	–22	+3	–11	+3	–14
				–20	–37	–7	–17	–7	–20	+1	–9	+1	–12
10	**18**	–8	0	–32	–50	–16	–24	–16	–27	–6	–14	–6	–17
				–24	–50	–8	–24	–8	–27	+2	–14	+2	–17
				–27	–47	–10	–22	–10	–25	0	–12	0	–15
18	**30**	–10	0	–40	–61	–20	–29	–20	–33	–7	–16	–7	–20
				–30	–61	–10	–29	–10	–33	+3	–16	+3	–20
				–33	–58	–12	–27	–13	–30	+1	–14	0	–17
30	**50**	–12	0	–50	–75	–25	–36	–25	–41	–9	–20	–9	–25
				–38	–75	–13	–36	–13	–41	+3	–20	+3	–25
				–42	–71	–16	–33	–17	–37	0	–17	–1	–21
50	**80**	–15	0	–60	–90	–30	–43	–30	–49	–10	–23	–10	–29
				–45	–90	–15	–43	–15	–49	+5	–23	+5	–29
				–50	–85	–19	–39	–19	–45	+1	–19	+1	–25
80	**120**	–20	0	–72	–107	–36	–51	–36	–58	–12	–27	–12	–34
				–52	–107	–16	–51	–16	–58	+8	–27	+8	–34
				–59	–100	–21	–46	–22	–52	+3	–22	+2	–28
120	**180**	–25	0	–85	–125	–43	–61	–43	–68	–14	–32	–14	–39
				–60	–125	–18	–61	–18	–68	+11	–32	+11	–39
				–68	–117	–24	–55	–25	–61	+5	–26	+4	–32
180	**250**	–30	0	–100	–146	–50	–70	–50	–79	–15	–35	–15	–44
				–70	–146	–20	–70	–20	–79	+15	–35	+15	–44
				–80	–136	–26	–64	–28	–71	+9	–29	+7	–36
250	**315**	–35	0	–110	–162	–56	–79	–56	–88	–17	–40	–17	–49
				–75	–162	–21	–79	–21	–88	+18	–40	+18	–49
				–87	–150	–29	–71	–30	–79	+10	–32	+9	–40
315	**400**	–40	0	–125	–182	–62	–87	–62	–98	–18	–43	–18	–54
				–85	–182	–22	–87	–22	–98	+22	–43	+22	–54
				–98	–169	–30	–79	–33	–87	+14	–35	+11	–43
400	**500**	–45	0	–135	–198	–68	–95	–68	–108	–20	–47	–20	–60
				–90	–198	–23	–95	–23	–108	+25	–47	+25	–60
				–105	–183	–32	–86	–35	–96	+16	–38	+13	–48

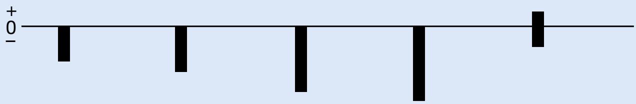
Shaft tolerances and resultant fits



Shaft Nominal diameter d		**Bearing** Bore diameter tolerance Δ_{dmp}		**Deviations of shaft diameter, resultant fits** Tolerances									
				e7		f5		f6		g5		g6	
				Deviations (shaft diameter) Theoretical interference (+)/clearance (–) Probable interference (+)/clearance (–)									
over	incl.	low	high										
mm		µm		µm									
500	**630**	–50	0	–145	–215	–76	–104	–76	–120	–22	–50	–22	–66
				–95	–215	–26	–104	–26	–120	+28	–50	+28	–66
				–111	–199	–36	–94	–39	–107	+18	–40	+15	–53
630	**800**	–75	0	–160	–240	–80	–112	–80	–130	–24	–56	–24	–74
				–85	–240	–5	–112	–5	–130	+51	–56	+51	–74
				–107	–218	–17	–100	–22	–113	+39	–44	+34	–57
800	**1 000**	–100	0	–170	–260	–86	–122	–86	–142	–26	–62	–26	–82
				–70	–260	+14	–122	+14	–142	+74	–62	+74	–82
				–97	–233	0	–108	–6	–122	+60	–48	+54	–62
1 000	**1 250**	–125	0	–195	–300	–98	–140	–98	–164	–28	–70	–28	–94
				–70	–300	+27	–140	+27	–164	+97	–70	+97	–94
				–103	–267	+10	–123	+3	–140	+80	–53	+73	–70
1 250	**1 600**	–160	0	–220	–345	–110	–160	–110	–188	–30	–80	–30	–108
				–60	–345	+50	–160	+50	–188	+130	–80	+130	–108
				–100	–305	+29	–139	+20	–158	+109	–59	+100	–78
1 600	**2 000**	–200	0	–240	–390	–120	–180	–120	–212	–32	–92	–32	–124
				–40	–390	+80	–180	+80	–212	+168	–92	+168	–124
				–90	–340	+55	–155	+45	–177	+143	–67	+133	–89

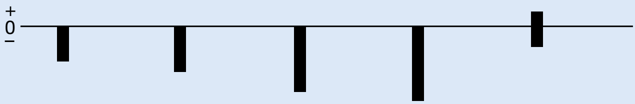
Table **7b**

Shaft tolerances and resultant fits



Shaft Nominal diameter d		Bearing Bore diameter tolerance Δ_{dmp}		Deviations of shaft diameter, resultant fits Tolerances									
				h5		h6		h8		h9		j5	
				Deviations (shaft diameter) Theoretical interference (+)/clearance (–) Probable interference (+)/clearance (–)									
over	incl.	low	high										
mm		µm		µm									
1	**3**	–8	0	0	–4	0	–6	0	–14	0	–25	+2	–2
				+8	–4	+8	–6	+8	–14	+8	–25	+10	–2
				+7	–3	+6	–4	+6	–12	+5	–22	+9	–1
3	**6**	–8	0	0	–5	0	–8	0	–18	0	–30	+3	–2
				+8	–5	+8	–8	+8	–18	+8	–30	+11	–2
				+7	–4	+6	–6	+5	–15	+5	–27	+10	–1
6	**10**	–8	0	0	–6	0	–9	0	–22	0	–36	+4	–2
				+8	–6	+8	–9	+8	–22	+8	–36	+12	–2
				+6	–4	+6	–7	+5	–19	+5	–33	+10	0
10	**18**	–8	0	0	–8	0	–11	0	–27	0	–43	+5	–3
				+8	–8	+8	–11	+8	–27	+8	–43	+13	–3
				+6	–6	+6	–9	+5	–24	+5	–40	+11	–1
18	**30**	–10	0	0	–9	0	–13	0	–33	0	–52	+5	–4
				+10	–9	+10	–13	+10	–33	+10	–52	+15	–4
				+8	–7	+7	–10	+6	–29	+6	–48	+13	–2
30	**50**	–12	0	0	–11	0	–16	0	–39	0	–62	+6	–5
				+12	–11	+12	–16	+12	–39	+12	–62	+18	–5
				+9	–8	+8	–12	+7	–34	+7	–57	+15	–2
50	**80**	–15	0	0	–13	0	–19	0	–46	0	–74	+6	–7
				+15	–13	+15	–19	+15	–46	+15	–74	+21	–7
				+11	–9	+11	–15	+9	–40	+9	–68	+17	–3
80	**120**	–20	0	0	–15	0	–22	0	–54	0	–87	+6	–9
				+20	–15	+20	–22	+20	–54	+20	–87	+26	–9
				+15	–10	+14	–16	+12	–46	+12	–79	+21	–4
120	**180**	–25	0	0	–18	0	–25	0	–63	0	–100	+7	–11
				+25	–18	+25	–25	+25	–63	+25	–100	+32	–11
				+19	–12	+18	–18	+15	–53	+15	–90	+26	–5
180	**250**	–30	0	0	–20	0	–29	0	–72	0	–115	+7	–13
				+30	–20	+30	–29	+30	–72	+30	–115	+37	–13
				+24	–14	+22	–21	+18	–60	+17	–102	+31	–7
250	**315**	–35	0	0	–23	0	–32	0	–81	0	–130	+7	–16
				+35	–23	+35	–32	+35	–81	+35	–130	+42	–16
				+27	–15	+26	–23	+22	–68	+20	–115	+34	–8
315	**400**	–40	0	0	–25	0	–36	0	–89	0	–140	+7	–18
				+40	–25	+40	–36	+40	–89	+40	–140	+47	–18
				+32	–17	+29	–25	+25	–74	+23	–123	+39	–10
400	**500**	–45	0	0	–27	0	–40	0	–97	0	–155	+7	–20
				+45	–27	+45	–40	+45	–97	+45	–155	+52	–20
				+36	–18	+33	–28	+28	–80	+26	–136	+43	–11

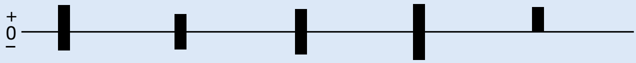
Shaft tolerances and resultant fits



Shaft Nominal diameter d		**Bearing** Bore diameter tolerance Δ_{dmp}		**Deviations of shaft diameter, resultant fits** Tolerances									
				h5		h6		h8		h9		j5	
				Deviations (shaft diameter) Theoretical interference (+)/clearance (–) Probable interference (+)/clearance (–)									
over	incl.	low	high										
mm		µm		µm									
500	**630**	–50	0	0	–28	0	–44	0	–110	0	–175	–	–
				+50	–28	+50	–44	+50	–110	+50	–175	–	–
				+40	–18	+37	–31	+31	–91	+29	–154	–	–
630	**800**	–75	0	0	–32	0	–50	0	–125	0	–200	–	–
				+75	–32	+75	–50	+75	–125	+75	–200	–	–
				+63	–20	+58	–33	+48	–98	+45	–170	–	–
800	**1 000**	–100	0	0	–36	0	–56	0	–140	0	–230	–	–
				+100	–36	+100	–56	+100	–140	+100	–230	–	–
				+86	–22	+80	–36	+67	–107	+61	–191	–	–
1 000	**1 250**	–125	0	0	–42	0	–66	0	–165	0	–260	–	–
				+125	–42	+125	–66	+125	–165	+125	–260	–	–
				+108	–25	+101	–42	+84	–124	+77	–212	–	–
1 250	**1 600**	–160	0	0	–50	0	–78	0	–195	0	–310	–	–
				+160	–50	+160	–78	+160	–195	+160	–310	–	–
				+139	–29	+130	–48	+109	–144	+100	–250	–	–
1 600	**2 000**	–200	0	0	–60	0	–92	0	–230	0	–370	–	–
				+200	–60	+200	–92	+200	–230	+200	–370	–	–
				+175	–35	+165	–57	+138	–168	+126	–296	–	–

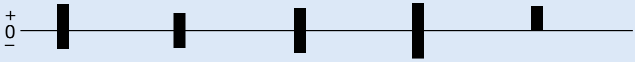
Table 7c

Shaft tolerances and resultant fits



Shaft Nominal diameter d		Bearing Bore diameter tolerance Δ_{dmp}		Deviations of shaft diameter, resultant fits Tolerances									
				j6		js5		js6		js7		k4	
over	incl.	low	high	Deviations (shaft diameter) Theoretical interference (+)/clearance (–) Probable interference (+)/clearance (–)									
mm		µm		µm									
1	**3**	–8	0	+4	–2	+2	–2	+3	–3	+5	–5	+3	0
				+12	–2	+10	–2	+11	–3	+13	–5	+11	0
				+10	0	+9	–1	+9	–1	+11	–3	+10	+1
3	**6**	–8	0	+6	–2	+2,5	–2,5	+4	–4	+6	–6	+5	+1
				+14	–2	+10,5	–2,5	+12	–4	+14	–6	+13	+1
				+12	0	+9	–1	+10	–2	+12	–4	+12	+2
6	**10**	–8	0	+7	–2	+3	–3	+4,5	–4,5	+7,5	–7,5	+5	+1
				+15	–2	+11	–3	+12,5	–4,5	+15,5	–7,5	+13	+1
				+13	0	+9	–1	+11	–3	+13	–5	+12	+2
10	**18**	–8	0	+8	–3	+4	–4	+5,5	–5,5	+9	–9	+6	+1
				+16	–3	+12	–4	+13,5	–5,5	+17	–9	+14	+1
				+14	–1	+10	–2	+11	–3	+14	–6	+13	+2
18	**30**	–10	0	+9	–4	+4,5	–4,5	+6,5	–6,5	+10,5	–10,5	+8	+2
				+19	–4	+14,5	–4,5	+16,5	–6,5	+20,5	–10,5	+18	+2
				+16	–1	+12	–2	+14	–4	+17	–7	+16	+4
30	**50**	–12	0	+11	–5	+5,5	–5,5	+8	–8	+12,5	–12,5	+9	+2
				+23	–5	+17,5	–5,5	+20	–8	+24,5	–12,5	+21	+2
				+19	–1	+15	–3	+16	–4	+20	–8	+19	+4
50	**80**	–15	0	+12	–7	+6,5	–6,5	+9,5	–9,5	+15	–15	+10	+2
				+27	–7	+21,5	–6,5	+24,5	–9,5	+30	–15	+25	+2
				+23	–3	+18	–3	+20	–5	+25	–10	+22	+5
80	**120**	–20	0	+13	–9	+7,5	–7,5	+11	–11	+17,5	–17,5	+13	+3
				+33	–9	+27,5	–7,5	+31	–11	+37,5	–17,5	+33	+3
				+27	–3	+23	–3	+25	–5	+31	–11	+30	+6
120	**180**	–25	0	+14	–11	+9	–9	+12,5	–12,5	+20	–20	+15	+3
				+39	–11	+34	–9	+37,5	–12,5	+45	–20	+40	+3
				+32	–4	+28	–3	+31	–6	+37	–12	+36	+7
180	**250**	–30	0	+16	–13	+10	–10	+14,5	–14,5	+23	–23	+18	+4
				+46	–13	+40	–10	+44,5	–14,5	+53	–23	+48	+4
				+38	–5	+34	–4	+36	–6	+43	–13	+43	+9
250	**315**	–35	0	+16	–16	+11,5	–11,5	+16	–16	+26	–26	+20	+4
				+51	–16	+46,5	–11,5	+51	–16	+61	–26	+55	+4
				+42	–7	+39	–4	+42	–7	+49	–14	+49	+10
315	**400**	–40	0	+18	–18	+12,5	–12,5	+18	–18	+28,5	–28,5	+22	+4
				+58	–18	+52,5	–12,5	+58	–18	+68,5	–28,5	+62	+4
				+47	–7	+44	–4	+47	–7	+55	–15	+55	+11
400	**500**	–45	0	+20	–20	+13,5	–13,5	+20	–20	+31,5	–31,5	+25	+5
				+65	–20	+58,5	–13,5	+65	–20	+76,5	–31,5	+70	+5
				+53	–8	+49	–4	+53	–8	+62	–17	+63	–12

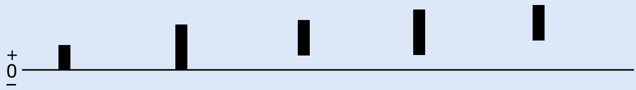
Shaft tolerances and resultant fits



Shaft Nominal diameter d		Bearing Bore diameter tolerance Δ_{dmp}		Deviations of shaft diameter, resultant fits Tolerances									
				j6		js5		js6		js7		k4	
				Deviations (shaft diameter) / Theoretical interference (+)/clearance (–) / Probable interference (+)/clearance (–)									
over	incl.	low	high										
mm		µm		µm									
500	**630**	–50	0	+22	–22	+14	–14	+22	–22	+35	–35	–	–
				+72	–22	+64	–14	+72	–22	+85	–35	–	–
				+59	–9	+54	–4	+59	–9	+69	–19	–	–
630	**800**	–75	0	+25	–25	+16	–16	+25	–25	+40	–40	–	–
				+100	–25	+91	–16	+100	–25	+115	–40	–	–
				+83	–8	+79	–4	+83	–8	+93	–18	–	–
800	**1 000**	–100	0	+28	–28	+18	–18	+28	–28	+45	–45	–	–
				+128	–28	+118	–18	+128	–28	+145	–45	–	–
				+108	–8	+104	–4	+108	–8	+118	–18	–	–
1 000	**1 250**	–125	0	+33	–33	+21	–21	+33	–33	+52	–52	–	–
				+158	–33	+146	–21	+158	–33	+177	–52	–	–
				+134	–9	+129	–4	+134	–9	+145	–20	–	–
1 250	**1 600**	–160	0	+39	–39	+25	–25	+39	–39	+62	–62	–	–
				+199	–39	+185	–25	+199	–39	+222	–62	–	–
				+169	–9	+164	–4	+169	–9	+182	–22	–	–
1 600	**2 000**	–200	0	+46	–46	+30	–30	+46	–46	+75	–75	–	–
				+246	–46	+230	–30	+246	–46	+275	–75	–	–
				+211	–11	+205	–5	+211	–11	+225	–25	–	–

Table **7d**

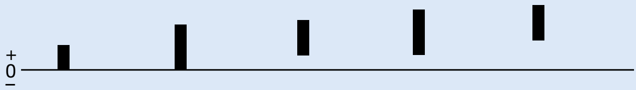
Shaft tolerances and resultant fits



Deviations (shaft diameter)
Theoretical interference (+)/clearance (–)
Probable interference (+)/clearance (–)

Shaft Nominal diameter d		Bearing Bore diameter tolerance Δ_{dmp}		Deviations of shaft diameter, resultant fits Tolerances									
				k5		k6		m5		m6		n5	
over	incl.	low	high										
mm		µm		µm									
1	**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

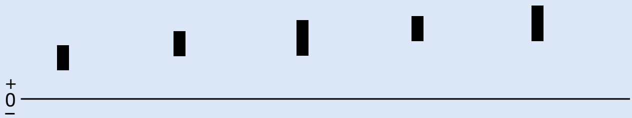
Shaft tolerances and resultant fits



Shaft Nominal diameter d		**Bearing** Bore diameter tolerance Δ_{dmp}		**Deviations of shaft diameter, resultant fits** Tolerances k5		k6		m5		m6		n5	
over	incl.	low	high	Deviations (shaft diameter) / Theoretical interference (+)/clearance (–) / Probable interference (+)/clearance (–)									
mm		µm		µm									
500	**630**	–50	0	+29	0	+44	0	+55	+26	+70	+26	+73	+44
				+78	0	+94	0	+104	+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

Table 7e

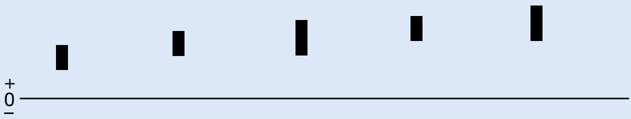
Shaft tolerances and resultant fits



Shaft Nominal diameter d		Bearing Bore diameter tolerance Δ_{dmp}		Deviations of shaft diameter, resultant fits Tolerances									
				n6		p6		p7		r6		r7	
over	incl.	low	high	Deviations (shaft diameter) Theoretical interference (+)/clearance (–) Probable interference (+)/clearance (–)									
mm		µm		µm									
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 **7e**

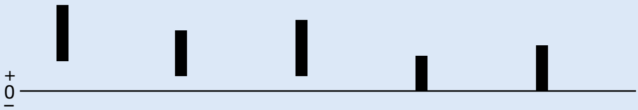
Shaft tolerances and resultant fits



Shaft Nominal diameter d		Bearing Bore diameter tolerance Δ_{dmp}		Deviations of shaft diameter, resultant fits Tolerances									
				n6		p6		p7		r6		r7	
over	incl.	low	high	Deviations (shaft diameter) / Theoretical interference (+)/clearance (–) / Probable interference (+)/clearance (–)									
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

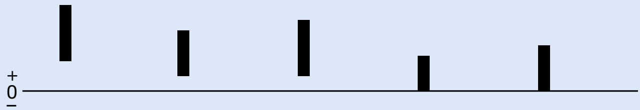
Table 8a

Housing tolerances and resultant fits



Housing Nominal bore diameter D		**Bearing** Outside diameter tolerance Δ_{Dmp}		**Deviations of housing bore diameter, resultant fits** Tolerances									
				F7		G6		G7		H5		H6	
				Deviations (housing bore diameter) Theoretical interference (+)/clearance (–) Probable interference (+)/clearance (–)									
over	incl.	high	low										
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

Housing tolerances and resultant fits

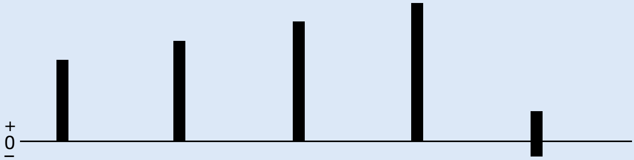


Deviations (housing bore diameter)
Theoretical interference (+)/clearance (–)
Probable interference (+)/clearance (–)

Housing Nominal bore diameter D		Bearing Outside diameter tolerance Δ_{Dmp}		Deviations of housing bore diameter, resultant fits Tolerances									
				F7		G6		G7		H5		H6	
over	incl.	high	low										
mm		µm		µm									
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
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

Table 8b

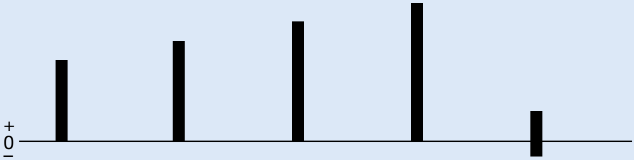
Housing tolerances and resultant fits



Housing Nominal bore diameter D		Bearing Outside diameter tolerance Δ_{Dmp}		Deviations of housing bore diameter, resultant fits Tolerances									
				H7		H8		H9		H10		J6	
				Deviations (housing bore diameter) Theoretical interference (+)/clearance (–) Probable interference (+)/clearance (–)									
over	incl.	high	low										
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	+87	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 8b

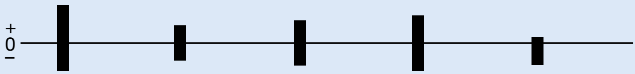
Housing tolerances and resultant fits



Housing Nominal bore diameter D		Bearing Outside diameter tolerance Δ_{Dmp}		Deviations of housing bore diameter, resultant fits Tolerances									
				H7		H8		H9		H10		J6	
				Deviations (housing bore diameter) Theoretical interference (+)/clearance (–) Probable interference (+)/clearance (–)									
over	incl.	high	low										
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	–	–

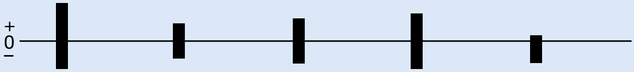
Table 8c

Housing tolerances and resultant fits



Housing Nominal bore diameter D		**Bearing** Outside diameter tolerance Δ_{Dmp}		**Deviations of housing bore diameter, resultant fits** Tolerances									
				J7		JS5		JS6		JS7		K5	
				Deviations (housing bore diameter) Theoretical interference (+)/clearance (–) Probable interference (+)/clearance (–)									
over	incl.	high	low										
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	–	–

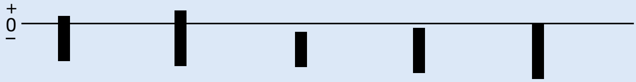
Housing tolerances and resultant fits



Housing Nominal bore diameter D		Bearing Outside diameter tolerance Δ_{Dmp}		Deviations of housing bore diameter, resultant fits Tolerances									
				J7		JS5		JS6		JS7		K5	
				Deviations (housing bore diameter) Theoretical interference (+)/clearance (–) Probable interference (+)/clearance (–)									
over	incl.	high	low										
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	–	–

Table 8d

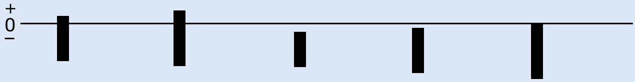
Housing tolerances and resultant fits



Housing Nominal bore diameter D		Bearing Outside diameter tolerance Δ_{Dmp}		Deviations of housing bore diameter, resultant fits Tolerances									
				K6		K7		M5		M6		M7	
				Deviations (housing bore diameter) Theoretical interference (+)/clearance (–) Probable interference (+)/clearance (–)									
over	incl.	high	low										
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

SKF

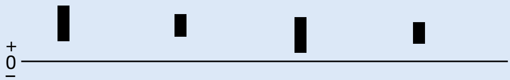
Housing tolerances and resultant fits



Housing Nominal bore diameter D		Bearing Outside diameter tolerance Δ_{Dmp}		Deviations of housing bore diameter, resultant fits Tolerances K6		K7		M5		M6		M7	
over	incl.	high	low	Deviations (housing bore diameter) / Theoretical interference (+)/clearance (–) / Probable interference (+)/clearance (–)									
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

Table 8e

Housing tolerances and resultant fits



Housing Nominal bore diameter D		**Bearing** Outside diameter tolerance Δ_{Dmp}		**Deviations of housing bore diameter, resultant fits** Tolerances							
				N6		N7		P6		P7	
over	incl.	high	low	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

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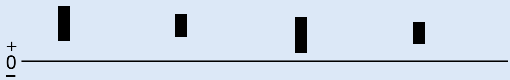
Housing tolerances and resultant fits



Housing Nominal bore diameter D		Bearing Outside diameter tolerance Δ_{Dmp}		Deviations of housing bore diameter, resultant fits Tolerances							
				N6		N7		P6		P7	
over	incl.	high	low	Deviations (housing bore diameter) Theoretical interference (+)/clearance (–) Probable interference (+)/clearance (–)							
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

Dimensional, form and running accuracy of bearing seatings and abutments
The accuracy of cylindrical bearing seatings on shafts and in housing bores, of seatings for thrust bearing washers and of the support surfaces (abutments for bearings provided by shaft and housing shoulders etc.) should correspond to the accuracy of the bearings used. In the following, guideline values for the dimensional, form and running accuracy are provided. These should be followed when machining the seatings and abutments.

Dimensional tolerances

For bearings made to Normal tolerances, the dimensional accuracy of cylindrical seatings on the shaft should be at least to grade 6 and in the housing at least to grade 7. Where adapter or withdrawal sleeves are used, wider diameter tolerances (grades 9 or 10) can be permitted than for bearing seatings (➔ **table 9**). The numerical values of standard tolerance grades IT to ISO 286-1:1988 will be found in **table 10**. For bearings with higher accuracy, correspondingly better grades should be used.

Tolerances for cylindrical form

The cylindricity tolerances as defined in ISO 1101:1983 should be 1 to 2 IT grades better than the prescribed dimensional tolerance, depending on requirements. For example, if a bearing shaft seating has been machined to tolerance m6, then the accuracy of form should be to IT5 or IT4. The tolerance value t_1 for cylindricity is obtained for an assumed shaft diameter of 150 mm from t_1 = IT5/2 = 18/2 = 9 µm. However, the tolerance t_1 is for a radius, hence $2 \times t_1$ applies for the shaft diameter. **Table 11**, **page 196**, gives guideline values for the cylindrical form tolerance and the total runout tolerance for the different bearing tolerance classes.

When bearings are to be mounted on adapter or withdrawal sleeves, the cylindricity of the sleeve seating should be IT5/2 (for h9) or IT7/2 (for h10) (➔ **table 9**).

Tolerances for perpendicularity

Abutments for bearing rings should have a rectangularity tolerance as defined in ISO 1101:1983, which is better by at least one IT grade than the diameter tolerance of the associated cylindrical seating. For thrust bearing washer seatings, the tolerance for perpendicularity should not exceed the values of IT5. Guideline values for the tolerance for rectangularity and for the total axial runout will be found in **table 11**, **page 196**.

Table 9

Shaft tolerances for bearings mounted on sleeves

Shaft diameter d Nominal over	incl.	**Diameter and form tolerances** h9 Deviations high	low	IT5[1)] max	h10 Deviations high	low	IT7[1)] max
mm		µm					
10	**18**	0	–43	8	0	–70	18
18	**30**	0	–52	9	0	–84	21
30	**50**	0	–62	11	0	–100	25
50	**80**	0	–74	13	0	–120	30
80	**120**	0	–87	15	0	–140	35
120	**180**	0	–100	18	0	–160	40
180	**250**	0	–115	20	0	–185	46
250	**315**	0	–130	23	0	–210	52
315	**400**	0	–140	25	0	–230	57
400	**500**	0	–155	27	0	–250	63
500	**630**	0	–175	32	0	–280	70
630	**800**	0	–200	36	0	–320	80
800	**1 000**	0	–230	40	0	–360	90
1 000	**1 250**	0	–260	47	0	–420	105

1) The recommendation is for IT5/2 or IT7/2, because the tolerance zone t is a radius, however in the table above the values relate to a nominal shaft diameter and are therefore not halved

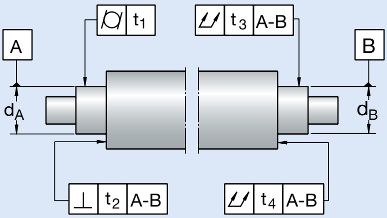
Table 10

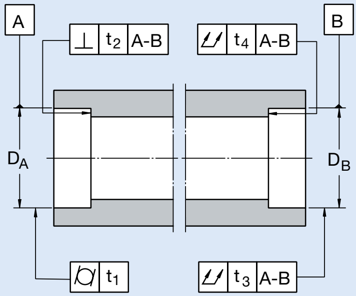
ISO tolerance grades for dimensions (lengths, widths, diameters etc.)

Nominal dimension over	incl.	**Tolerance grades** IT1 max	IT2	IT3	IT4	IT5	IT6	IT7	IT8	IT9	IT10	IT11	IT12
mm		µm											
1	**3**	0,8	1,2	2	3	4	6	10	14	25	40	60	100
3	**6**	1	1,5	2,5	4	5	8	12	18	30	48	75	120
6	**10**	1	1,5	2,5	4	6	9	15	22	36	58	90	150
10	**18**	1,2	2	3	5	8	11	18	27	43	70	110	180
18	**30**	1,5	2,5	4	6	9	13	21	33	52	84	130	210
30	**50**	1,5	2,5	4	7	11	16	25	39	62	100	160	250
50	**80**	2	3	5	8	13	19	30	46	74	120	190	300
80	**120**	2,5	4	6	10	15	22	35	54	87	140	220	350
120	**180**	3,5	5	8	12	18	25	40	63	100	160	250	400
180	**250**	4,5	7	10	14	20	29	46	72	115	185	290	460
250	**315**	6	8	12	16	23	32	52	81	130	210	320	520
315	**400**	7	9	13	18	25	36	57	89	140	230	360	570
400	**500**	8	10	15	20	27	40	63	97	155	250	400	630
500	**630**	–	–	–	–	32	44	70	110	175	280	440	700
630	**800**	–	–	–	–	36	50	80	125	200	320	500	800
800	**1 000**	–	–	–	–	40	56	90	140	230	360	560	900
1 000	**1 250**	–	–	–	–	47	66	105	165	260	420	660	1050
1 250	**1 600**	–	–	–	–	55	78	125	195	310	500	780	1250
1 600	**2 000**	–	–	–	–	65	92	150	230	370	600	920	1 500
2 000	**2 500**	–	–	–	–	78	110	175	280	440	700	1 100	1 750

Table 11

Accuracy of form and position for bearing seatings on shafts and in housings





Surface Characteristic	Symbol for characteristic	tolerance zone	**Permissible deviations** Bearings of tolerance class[1] Normal, CLN	P6	P5	
Cylindrical seating						
Cylindricity	⌭	t_1	IT5/2	IT4/2	IT3/2	IT2/2
Total radial runout	⌰	t_3	IT5/2	IT4/2	IT3/2	IT2/2
Flat abutment						
Rectangularity	⊥	t_2	IT5	IT4	IT3	IT2
Total axial runout	⌰	t_4	IT5	IT4	IT3	IT2

Explanation

For normal demands

For special demands with respect to running accuracy or even support

[1] For bearings of higher accuracy (tolerance class P4 etc.) please refer to SKF catalogue "High-precision bearings"

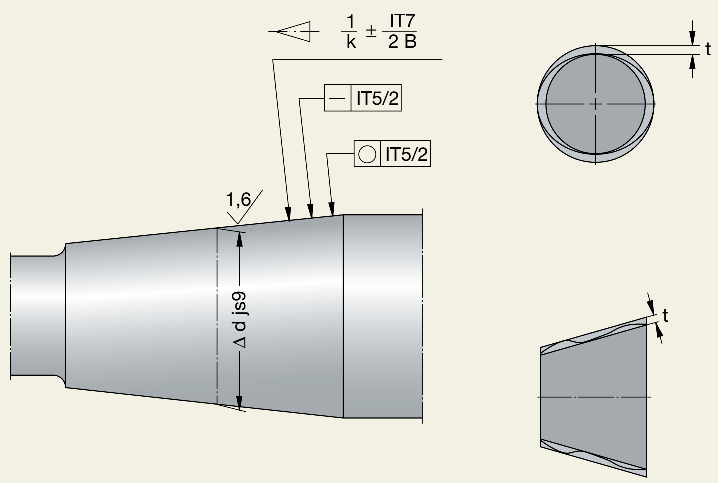
Tolerances for tapered journal seatings

When a bearing is mounted directly onto a tapered shaft seating, the seating diameter tolerance can be wider than in the case of cylindrical seatings. **Fig** 18 shows a grade 9 diameter tolerance, while the form tolerance stipulations are the same as for a cylindrical shaft seating. SKF recommendations for tapered shaft seatings for rolling bearings are as follows.

- The permissible taper deviation for machining the taper seatings is a ± tolerance in accordance with IT7/2 based on the bearing width. The value is determined according to the formula shown in **fig** 18, where
  k = factor for the taper
  12 for taper 1: 12
  30 for taper 1: 30
  B = bearing width
- The straightness tolerance is IT5/2, based on the diameter d and is defined as:
  "In each axial plane through the tapered surface of the shaft, the tolerance zone is limited by two parallel lines a distance "t" apart."
- The radial deviation from circularity is IT5/2, based on the diameter d and is defined as:
  "In each radial plane along the tapered surface of the shaft, the tolerance zone is limited by two concentric circles a distance "t" apart."
  When particularly stringent running accuracy requirements are stipulated, IT4/2 is to apply instead.

The best way to check that the taper is within the recommended tolerances is to measure with dial gauges. A more practical method, but less accurate, is to use ring gauges, special taper gauges or sine bars.

Fig 18



Surface roughness of bearing seatings

The roughness of bearing seating surfaces does not have the same degree of influence on bearing performance as the dimensional, form and running accuracies. However, a desired interference fit is much more accurately obtained the smoother the mating surfaces are. For less critical bearing arrangements relatively large surface roughness is permitted.

For bearing arrangements where demands for accuracy are high, guideline values for the mean surface roughness R_a are given in **table 12** for different dimensional accuracies of the bearing seatings. These recommendations apply to ground seatings, which are normally assumed for shaft seatings.

Raceways on shafts and in housings

Raceways machined in associated components for cylindrical roller bearings with only one ring and for cylindrical roller and cage thrust assemblies, must have a hardness of between 58 and 64 HRC if the load carrying capacity of the bearing or assembly is to be fully exploited.

The surface roughness should be $R_a \leq 0{,}2$ µm or $R_z \leq 1$ µm. For less demanding applications, lower hardness and rougher surfaces may be used.

The out-of-round and deviation from cylindrical form must not exceed 25 and 50 %, respectively, of the actual diameter tolerance of the raceway.

The permissible axial runouts of raceways for thrust assemblies are the same as for the shaft and housing washers of thrust bearings, shown in **table 10**, **page 132**.

Suitable materials for the seatings include through-hardening steels, e.g. 100 Cr 6 to ISO 683-17:1999, case-hardening steels, e.g. 20Cr3 or 17MnCr5 to ISO 683-17:1999) as well as induction hardening steels which can be partially hardened.

The case depth that is recommended for raceways machined in associated components depends on various factors including the dynamic and static load ratios (P/C and P_0/C_0 respectively) as well as the core hardness, and it is difficult to generalize. For example, under conditions of purely static load up to the magnitude of the basic static load rating and with a core hardness of 350 HV, the recommended case depth is in the order of 0,1 times the rolling element diameter. Smaller case depths are permitted for dynamic loads. For additional information, please consult the SKF application engineering service.

Table 12

Guideline values for surface roughness of bearing seatings

Diameter of seatings d (D)[2) over	incl.	Recommended R_a value for ground seatings (Roughness grade numbers) Diameter tolerance to IT7	IT6	IT5
mm		µm		
–	**80**	1,6 (N7)	0,8 (N6)	0,4 (N5)
80	**500**	1,6 (N7)	1,6 (N7)	0,8 (N6)
500	**1 250**	3,2 (N8)[1)	1,6 (N7)	1,6 (N7)

1) When using the oil injection method for mounting R_a should not exceed 1,6 µm
2) For diameters > 1 250 mm consult the SKF application engineering service

Axial location of bearings

An interference fit alone is inadequate for the axial location of a bearing ring. As a rule, therefore, some suitable means of axially securing the ring is needed.

Both rings of a locating bearing should be axially secured on both sides. However, for non-locating bearings, that are of a non-separable design, the ring having the tighter fit – usually the inner ring – should be axially secured; the other ring must be free to move axially with respect to its seating, except for CARB bearings where both the rings are axially secured. For "cross-located" bearings the bearing rings need only be axially secured on one side.

Methods of location

Bearings with cylindrical bore

Bearing rings having an interference fit are generally mounted so that the ring abuts a shoulder on the shaft or in the housing on one side (➔ **fig** 19). On the opposite side, inner rings are normally secured using lock nuts, as shown in the section "Lock nuts", starting on **page 1003**, e.g. of type KM + MB (➔ **fig** 19) or by end plates (➔ **fig** 20) attached to the shaft end. Outer rings are usually retained by the housing end cover (➔ **fig** 21) or possibly, in special cases, by a threaded ring (➔ **fig** 22).

Fig 20

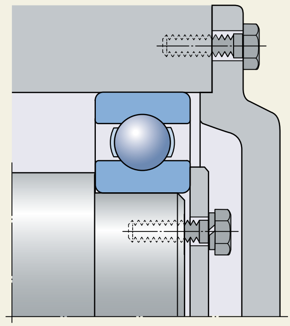


Fig 21

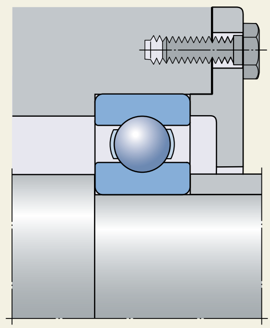


Fig 19

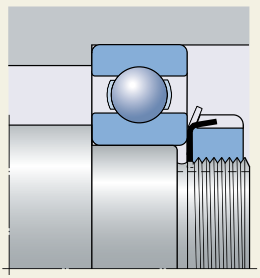
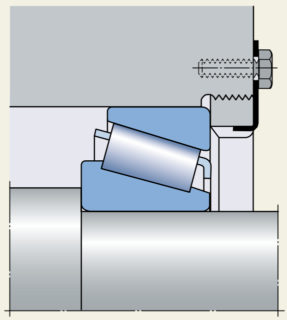


Fig 22



Instead of integral shaft or housing shoulders, it is frequently more convenient to use spacer sleeves or collars between the bearing rings or between a bearing ring and an adjacent component, e.g. a gear (➔ **fig 23**). Location on a shaft can also be accomplished using a split collar that is seated in a groove in the shaft (➔ **fig 25**) and retained either by a second one-piece collar or ring or by the bearing inner ring.

The use of snap rings for the axial location of rolling bearings saves space, permits rapid mounting and dismounting, and simplifies the machining of shafts and housing bores. If moderate or heavy axial loads have to be supported an abutment collar should be inserted between the bearing ring and the snap ring, so that the snap ring is not subjected to excessive bending moments (➔ **fig 24**). The usual axial play between the snap ring and snap ring groove can be reduced, if necessary, by choosing suitable tolerances for the abutment collar or by using shims. Bearings with a snap ring groove in the outer ring (➔ **fig 23**) can be secured in a very simple and space-saving manner using a snap ring (➔ section "Deep groove ball bearings", starting on **page 287**).

Other methods of axial location which are suitable, especially for high precision bearing arrangements involve the use of press fits, e.g. in the form of stepped sleeve arrangements. Additional details are found in the SKF catalogue "High-precision bearings".

Fig 23

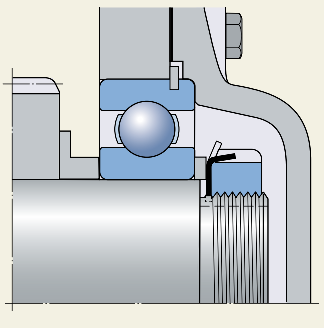
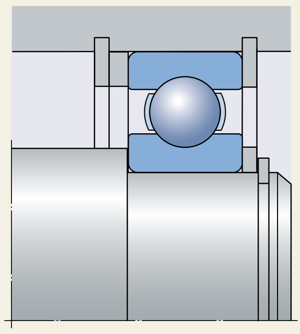


Fig 24



Bearings with tapered bore

Bearings with a tapered bore mounted directly on tapered journals are generally retained on the shaft by a lock nut, or by a lock nut on an externally threaded split ring inserted in a groove in the shaft (➔ **fig** 25).

When using an adapter sleeve on a stepped shaft, the lock nut positions the bearing relative to the sleeve, and a spacer ring is inserted between the shaft shoulder and inner ring on the other side (➔ **fig** 26). Where smooth shafts without integral abutments are used (➔ **fig** 27), the friction between the shaft and sleeve governs the axial load carrying capacity of the bearing, see sections

- “Self-aligning ball bearings” starting on **page 463** and
- “Spherical roller bearings” starting on **page 691**.

Where bearings are mounted on a withdrawal sleeve, an abutment, e.g. a spacer ring, which is frequently designed as a labyrinth ring, must support the inner ring. The withdrawal sleeve itself is axially located by an end plate or a lock nut (➔ **fig** 28).

Fig 26

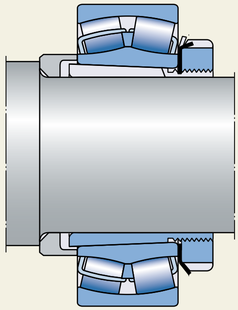


Fig 27

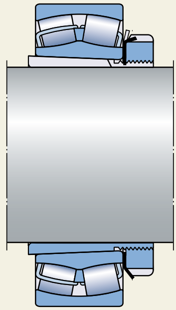


Fig 25

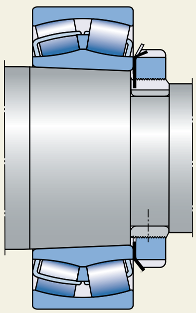
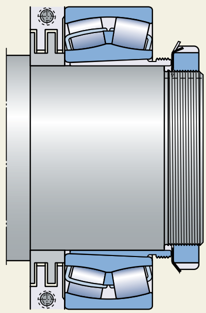


Fig 28



Abutment and fillet dimensions

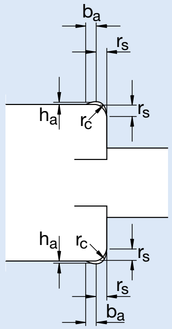
The dimensions of components adjacent to the bearing (shaft and housing shoulders, spacer sleeves etc.) must be such that sufficient support is provided for the bearing rings, but there must be no contact between the rotating parts of the bearing and a stationary component. Appropriate abutment and fillet dimensions are quoted for each bearing listed in the product tables.

The transition between the bearing seating and shaft or housing shoulder, may either take the form of a simple fillet according to the dimensions r_a and r_b in the product tables, or be relieved in the form of an undercut. **Table** 13 gives suitable dimensions for the relieved fillets.

The greater the fillet radius (for the smooth form curve), the more favourable is the stress distribution in the shaft fillet area. For heavily loaded shafts, therefore, a large radius is generally required. In such cases a spacing collar should be provided between the inner ring and shaft shoulder to provide a sufficiently large support surface for the bearing ring. The side of the collar facing the shaft shoulder should be relieved so that it does not contact the shaft fillet (➔ **fig** 29).

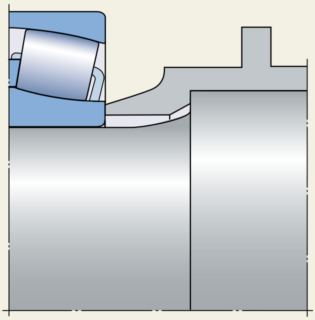
Table 13

Relieved fillets



Bearing chamfer dimension r_s	**Fillet dimensions** b_a	h_a	r_c
mm	mm		
1	2	0,2	1,3
1,1	2,4	0,3	1,5
1,5	3,2	0,4	2
2	4	0,5	2,5
2,1	4	0,5	2,5
3	4,7	0,5	3
4	5,9	0,5	4
5	7,4	0,6	5
6	8,6	0,6	6
7,5	10	0,6	7
9,5	12	0,6	9

Fig 29

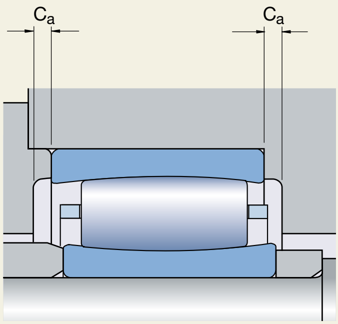


CARB toroidal roller bearings

CARB bearings can accommodate axial expansion of the shaft within the bearing. To be sure that these axial displacements of the shaft with respect to the housing can take place it is necessary to provide space on both sides of the bearing (➔ **fig** 30).

Additional information is found in the section “CARB toroidal roller bearings”, starting on **page 775**.

Fig 30



Designing associated components

Particularly where large bearings are involved, it is often necessary to make provisions during the bearing arrangement design stage, to facilitate mounting and dismounting of the bearing, or even to make it possible at all. If, for example, slots or recesses are machined in the shaft and housing shoulders, it is possible to apply withdrawal tools (➔ **fig** 31). Threaded holes in the housing shoulders also allow the use of screws to push the bearing from its seating (➔ **fig** 32).

If the oil injection method is to be used to mount or dismount bearings on a tapered seating, or to dismount bearings from a cylindrical seating, it is necessary to provide ducts and grooves in the shaft (➔ **fig** 33). The distance of the oil distribution groove from the side of the bearing from which mounting or dismounting is to be undertaken should be about one third of the seating width. Recommended dimensions for the appropriate grooves, ducts and threaded holes to connect the oil supply will be found in **tables** 14 and 15.

Fig 31

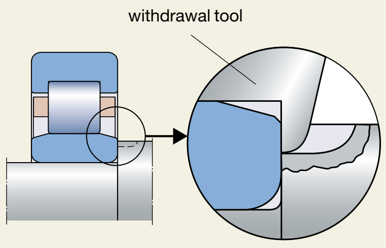


Fig 32

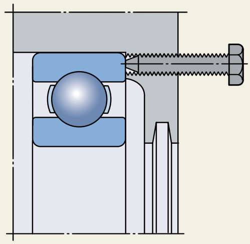


Fig 33

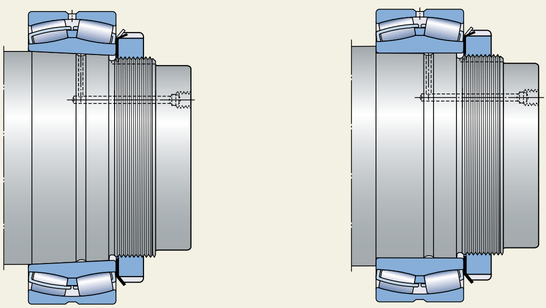
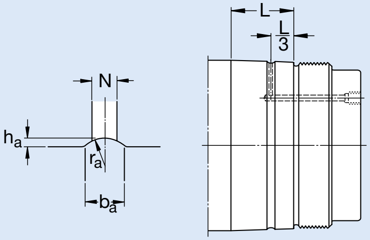


Table 14

Recommended dimensions for oil supply ducts and grooves

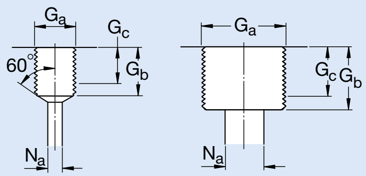


Seating diameter over	incl.	Dimensions b_a	h_a	r_a	N
mm		mm			
	100	3	0,5	2,5	2,5
100	**150**	4	0,8	3	3
150	**200**	4	0,8	3	3
200	**250**	5	1	4	4
250	**300**	5	1	4	4
300	**400**	6	1,25	4,5	5
400	**500**	7	1,5	5	5
500	**650**	8	1,5	6	6
650	**800**	10	2	7	7
800	**1 000**	12	2,5	8	8

L = width of bearing seating

Table 15

Design and recommended dimensions for threaded holes for connecting oil supply



Thread G_a	Design	Dimensions G_b	G_c [1) max	N_a
–	–	mm		
M 6	A	10	8	3
R 1/8	A	12	10	3
R 1/4	A	15	12	5
R 3/8	B	15	12	8
R 1/2	B	18	14	8
R 3/4	B	20	16	8

1) Effective threaded length

Bearing preload

Depending on the application it may be necessary to have either a positive or a negative operational clearance in the bearing arrangement. In the majority of applications, the operational clearance should be positive, i.e. when in operation, the bearing should have a residual clearance, however slight (➔ section “Bearing internal clearance” on **page 137**).

However, there are many cases, e.g. machine tool spindle bearings, pinion bearings in automotive axle drives, bearing arrangements of small electric motors, or bearing arrangements for oscillating movement, where a negative operational clearance, i.e. a preload, is needed to enhance the stiffness of the bearing arrangement or to increase running accuracy. The application of a preload, e.g. by springs, is also recommended where bearings are to operate without load or under very light load and at high speeds. In these cases, the preload serves to provide a minimum load on the bearing and prevent bearing damage as a result of sliding movements of the rolling elements (➔ section “Requisite minimum load” on **page 75**).

Types of preload

Depending on the type of bearing the preload may be either radial or axial. Cylindrical roller bearings, for example, because of their design, can only be preloaded radially, and

Fig 34

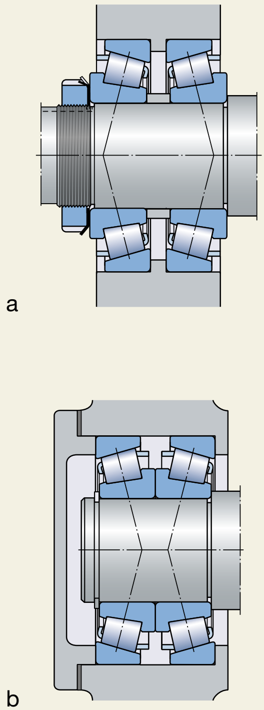
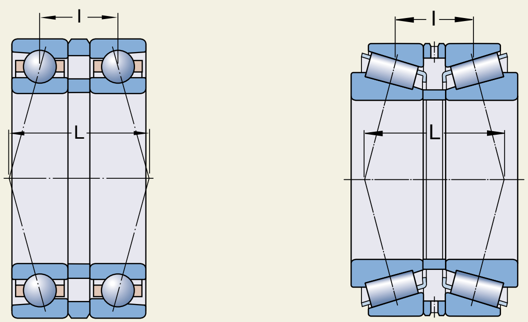


Fig 35

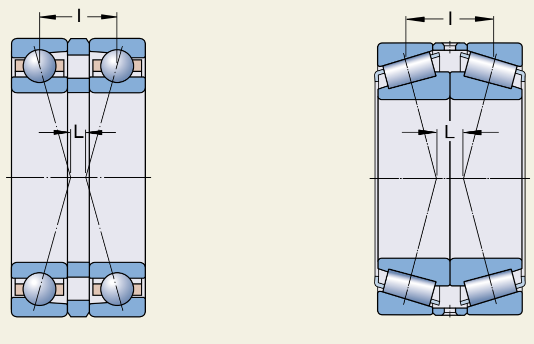


thrust ball and cylindrical roller thrust bearings can only be preloaded axially. Single row angular contact ball bearings and taper roller bearings (➔ **fig** 34), which are normally subjected to axial preload, are generally mounted together with a second bearing of the same type in a back-to-back (**a**) or face-to-face (**b**) arrangement. Deep groove ball bearings are also generally preloaded axially, to do so, the bearings should have a greater radial internal clearance than Normal (e.g. C3) so that, as with angular contact ball bearings, a contact angle which is greater than zero will be produced.

For both taper roller and angular contact ball bearings, the distance L between the pressure centres is longer when the bearings are arranged back-to-back (➔ **fig** 35) and shorter when they are arranged face-to-face (➔ **fig** 36) than the distance l between the bearing centres. This means that the bearings arranged back-to-back can accommodate large tilting moments even if the distance between the bearing centres is relatively short. The radial forces resulting from the moment load and the deformation caused by these in the bearings are smaller than for bearings arranged face-to-face.

If in operation the shaft becomes hotter than the housing, the preload which was adjusted (set) at ambient temperature during mounting will increase, the increase being greater for face-to-face than for back-to-back arrangements. In both cases the thermal expansion in the radial direction serves to reduce clearance or increase preload. This tendency is increased by the thermal expansion in the axial direction when the bearings are face-to-face, but is reduced for back-to-back arrangements. For back-to-back arrangements only, for a given distance between the bearings and when the coefficient of thermal expansion is the same for the bearings and associated components, the radial and axial thermal expansions will cancel each other out so that the preload will not change.

Fig 36



Effects of bearing preload
The main effects of bearing preload are to

- enhance stiffness,
- reduce running noise,
- enhance the accuracy of shaft guidance,
- compensate for wear and settling (bedding down) processes in operation, and
- provide long service life.

High stiffness

Bearing stiffness (in kN/µm) is defined as the ratio of the force acting on the bearing to the elastic deformation in the bearing. The elastic deformation caused by a load in preloaded bearings is smaller for a given load range than in bearings which are not preloaded.

Quiet running

The smaller the operational clearance in a bearing, the better the guidance of the rolling elements in the unloaded zone and the quieter the bearing in operation.

Accurate shaft guidance

Preloaded bearings provide more accurate shaft guidance because preload restricts the ability of the shaft to deflect under load. For example, the more accurate guidance and the increased stiffness afforded by preloaded pinion and differential bearings means that the gear mesh will be kept accurate and remain constant, and that additional dynamic forces will be minimized. As a result, operation will be quiet and the gear mesh will have a long service life.

Compensating for wear and settling

Wear and settling processes in a bearing arrangement during operation increase the clearance but this can be compensated for by preload.

Long service life

In certain applications preloaded bearing arrangements can enhance operational reliability and increase service life. A properly dimensioned preload can have a favourable influence on the load distribution in the bearings and therefore on service life (➔ section “Maintaining the correct preload” on **page 216**).

Determining preload force
Preload may be expressed as a force or as a path (distance), although the preload force is the primary specification factor. Depending on the adjustment method, preload is also indirectly related to the friction torque in the bearing.

Empirical values for the optimum preload can be obtained from proven designs and can be applied to similar designs. For new designs SKF recommends calculating the preload force and checking its accuracy by testing. As generally not all influencing factors of the actual operation are accurately known, corrections may be necessary in practice. The reliability of the calculation depends above all on how well the assumptions made regarding the temperature conditions in operation and the elastic behaviour of the associated components – most importantly the housing – coincide with the actual conditions.

When determining the preload, the preload force required to give an optimum combination of stiffness, bearing life and operational reliability should be calculated first. Then calculate the preload force to be used when adjusting the bearings during mounting. When mounting, the bearings should be at ambient temperature and not subjected to an operating load.

The appropriate preload at normal operating temperature depends on the bearing load. An angular contact ball bearing or a taper roller bearing can accommodate radial and axial loads simultaneously. Under radial load, a force acting in the axial direction will be produced in the bearing, and this must generally be accommodated by a second bearing which faces in the opposite direction to the first one. Purely radial displacement of one bearing ring in relation to the other will mean that half of the bearing circumference (i.e. half of the rolling elements) is under load and the axial force produced in the bearing will be

$F_a = R\ F_r$ for single row angular contact ball bearings or
$F_a = 0{,}5\ F_r/Y$ for single row taper roller bearings.

where F_r is the radial bearing load (➔ **fig 37**).

The values of the variable R which takes into account the contact conditions inside angular contact ball bearings has to be determined according to the guidelines given in the section "Determining axial force for bearings mounted singly or paired in tandem", starting on **page 413**.

The values of the axial factor Y for taper roller bearings will be found in the product tables.

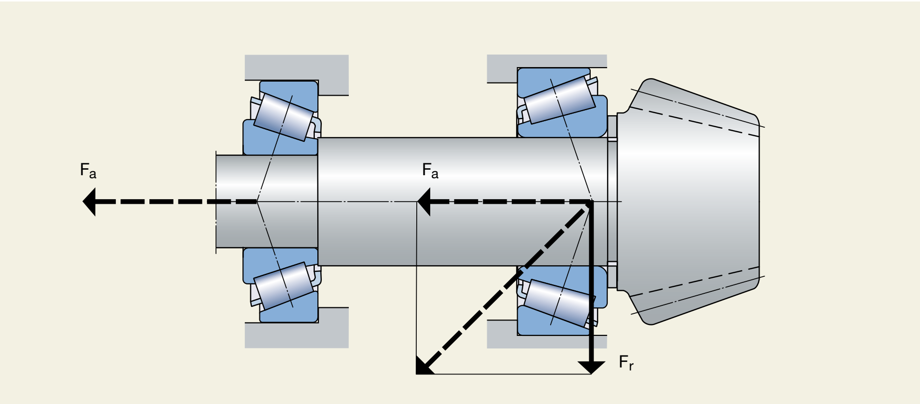
When a single bearing is subjected to a radial load F_r, an external axial force F_a of the above magnitude must be applied if the prerequisite for the basic load ratings (half of the bearing circumference under load) is to be fulfilled. If the applied external force is smaller, the number of rolling elements supporting the load will be smaller and the load carrying capacity of the bearing will be correspondingly reduced.

In a bearing arrangement comprising two single row angular contact ball bearings or two taper roller bearings back-to-back or face-to-face, each bearing must accommodate the axial forces from the other. When the two bearings are the same, the radial load acts centrally between the bearings and if the bearing arrangement is adjusted to zero clearance, the load distribution where half of the rolling elements are under load will be automatically achieved. In other load cases, particularly where there is an external axial load, it may be necessary to preload the bearings to compensate for the play produced as a result of the elastic deformation of the bearing taking the axial load into account and to achieve a more favourable load distribution in the other bearing which is unloaded axially.

Preloading also increases the stiffness of the bearing arrangement. When considering stiffness it should be remembered that it is not only influenced by the resilience of the bearings but also by the elasticity of the shaft and housing, the fits with which the rings are mounted and the elastic deformation of all other components in the force field including the abutments. These all have a considerable impact on the resilience of the total shaft system. The axial and radial resilience of a bearing depend on its internal design, i.e. on the contact conditions (point or line contact), the number and diameter of the rolling elements and the contact angle; the greater the contact angle, the greater the stiffness of the bearing in the axial direction.

If, as a first approximation, a linear dependence of the resilience on the load is assumed, i.e. a constant spring ratio, then a comparison shows that the axial displacement in a bearing arrangement under preload is smaller than for a bearing arrangement

Fig 37



without preload for the same external axial force K_a (➔ **diagram 2**). A pinion bearing arrangement, for example, consists of two taper roller bearings A and B of different size having the spring constants c_A and c_B and is subjected to a preload force F_0. If the axial force K_a acts on bearing A, bearing B will be unloaded, and the additional load acting on bearing A and the axial displacement d_a will be smaller than for a bearing without preload. However, if the external axial force exceeds the value

$$K_a = F_0 \left(1 + \frac{c_A}{c_B}\right)$$

then bearing B will be relieved of the axial preload force and the axial displacement under additional load will be the same as it is for a bearing arrangement without preload, i.e. determined solely by the spring constant of bearing A. To prevent complete unloading of bearing B when bearing A is subjected to load K_a, the following preload force will thus be required

$$F_0 = K_a \frac{c_B}{c_A + c_B}$$

The forces and elastic displacements in a preloaded bearing arrangement as well as the effects of a change in preload force are most easily recognized from a preload force/preload path diagram (➔ **diagram 3**). This consists of the spring curves of the components that are adjusted against each other to preload and enables the following.

- the relationship of the preload force and preload path within the preloaded bearing arrangement;
- the relationship between an externally applied axial force K_a and the bearing load for a preloaded bearing arrangement, as well as the elastic deformation produced by the external force.

In the **diagram 3**, all the components subjected to additional loads by the operational forces are represented by the curves that increase from left to right, and all the unloaded components by the curves that increase from right to left. Curves 1, 2 and 3 are for different preload forces (F_{01}, $F_{02} < F_{01}$ and $F_{03} = 0$). The broken lines refer to the bearings themselves whereas the unbroken lines are for the bearing position in total (bearing with associated components).

Using **diagram 3** it is possible to explain the relationships, for example, for a pinion bearing arrangement (➔ **fig 39, page 213**) where bearing A is adjusted against bearing B via the shaft and housing to give preload. The external axial force K_a (axial component of tooth forces) is superimposed on the preload force F_{01} (curve 1) in such a way that bearing A is subjected to additional load while bearing B is unloaded. The load at bearing position A is designated F_{aA}, that at bearing position B, F_{aB}.

Under the influence of the force K_a, the pinion shaft is axially displaced by the amount δ_{a1}. The smaller preload force F_{02} (curve 2) has been chosen so that bearing B is just unloaded by the axial force K_a, i.e. $F_{aB} = 0$ and $F_{aA} = K_a$. The pinion shaft is displaced in this case by the amount $\delta_{a2} > \delta_{a1}$. When the arrangement is not preloaded (curve 3) the axial displacement of the pinion shaft is at its greatest ($\delta_{a3} > \delta_{a2}$).

Diagram 2

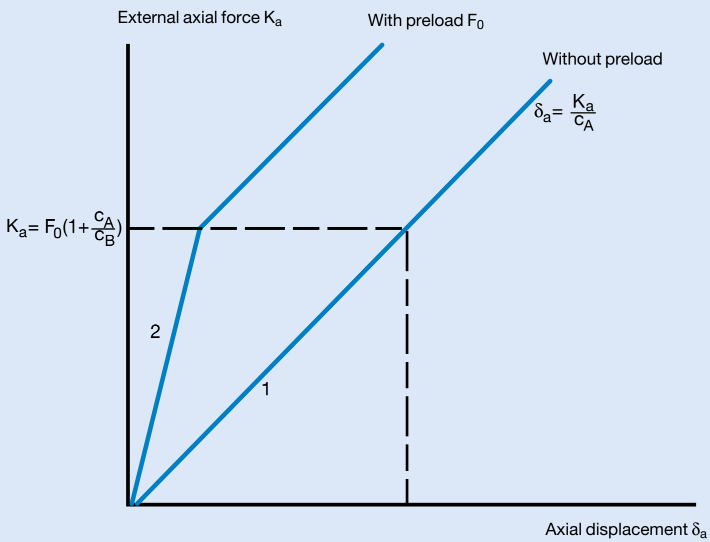
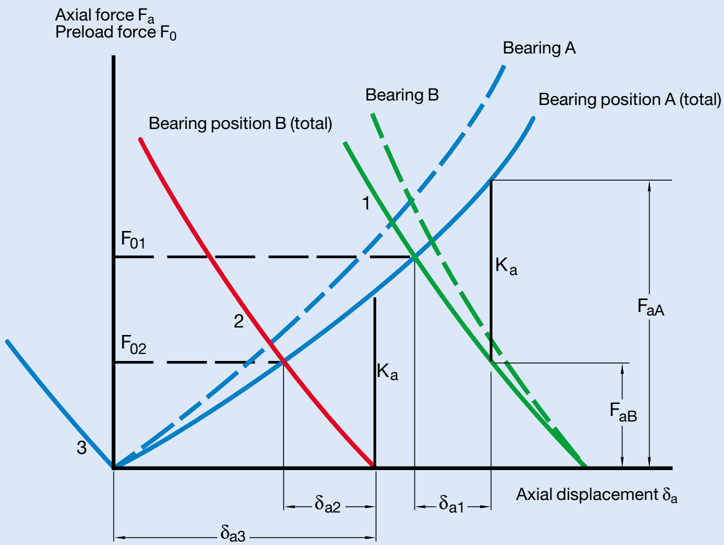


Diagram 3



Adjustment procedures

Adjustment means setting the bearing internal clearance (➔ section “Mounting”, starting on **page 261**) or setting the preload of a bearing arrangement.

The radial preload usually used for cylindrical roller bearings, double row angular contact ball bearings and sometimes for deep groove ball bearings, for example, is achieved by using a sufficient degree of interference for one or both of the bearing rings to reduce the initial internal clearance of the bearing to zero so that in operation there will be a negative clearance, i.e. preload.

Bearings with a tapered bore are particularly suitable for radial preloading since, by driving the bearing up on to its tapered seating, the preload can be applied to within narrow limits.

Axial preload in single row angular contact ball bearings, taper roller bearings and also deep groove ball bearings is produced by displacing one bearing ring axially in relation to the other by an amount corresponding to the desired preload force. There are two main groups of adjustment methods that differ in the principle on which they are based: individual adjustment and collective adjustment.

Individual adjustment

With individual adjustment, each bearing arrangement is adjusted separately using nuts, shims, spacer sleeves, deformable sleeves etc. Measuring and inspection procedures provide that the established nominal preload force is obtained with the least possible deviation. There are different methods depending on the quantity of bearings to be measured:

- adjustment using preload path,
- adjustment using friction torque, and
- adjustment using direct force measurement.

Individual adjustment has the advantage that individual components can be produced to Normal tolerances and the desired preload can be achieved with a reasonably high degree of accuracy.

Adjustment using preload path

This method of adjustment is frequently used when the components of a bearing arrangement are pre-assembled. The preload is achieved, for example, for a pinion bearing arrangement by

- fitting intermediate rings between the outer and inner rings of the two bearings (➔ **fig** 38);
- inserting shims between a housing shoulder and a bearing outer ring or between the casing and the housing (➔ **fig** 39), the housing in this case is the flanged angled insert;
- fitting a spacer ring between a shaft shoulder and one of the bearing inner rings (➔ **fig** 40) or between the inner rings of the two bearings.

Fig 38

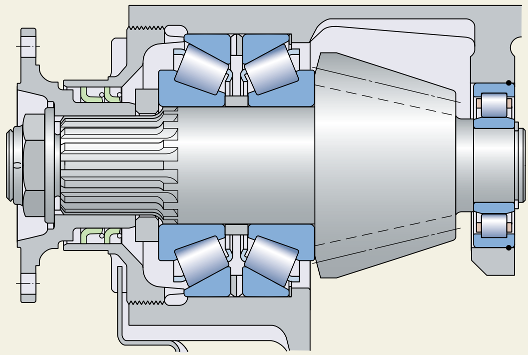


Fig 39

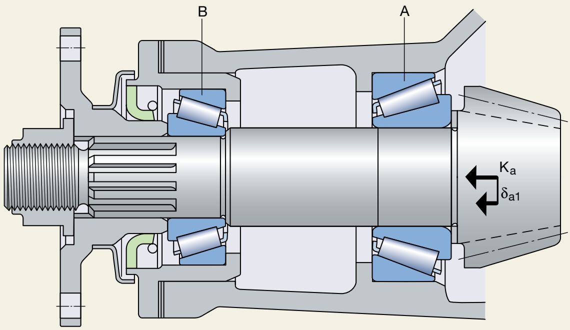
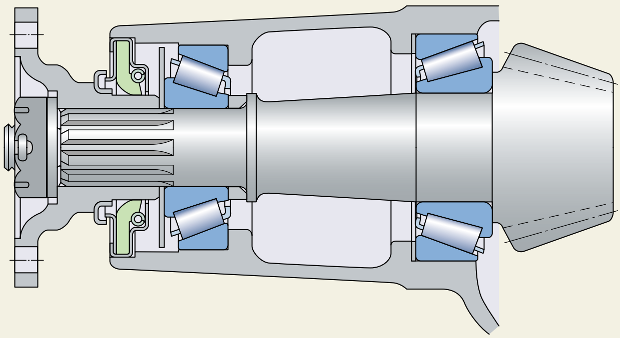


Fig 40

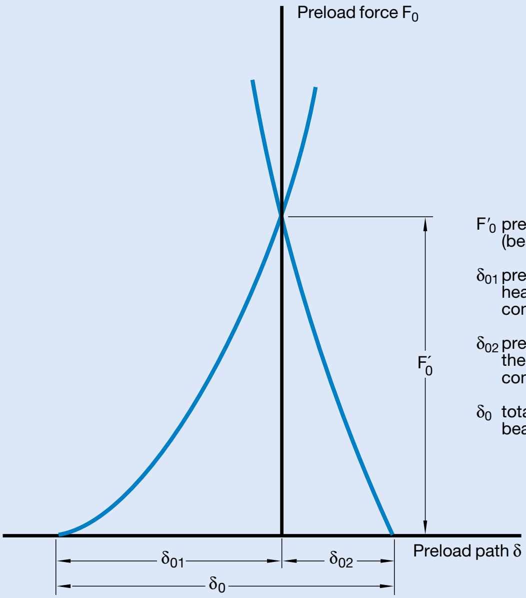


The width of the shims, intermediate rings or spacer rings is determined by

- the distance between the shaft and housing shoulders,
- the total width of both bearings,
- the preload path (axial displacement) corresponding to the desired preload force,
- a correction factor for the preload path to account for thermal expansion in operation,
- the manufacturing tolerances of all components, established by measuring them before mounting, and
- a correction factor to account for a certain loss of preload force after a certain period of operation.

This method of adjustment is based on the relationship between the preload force and the elastic deformations within the preloaded system. The requisite preload can be determined from a preload force/preload path diagram (➔ **diagram 4**).

Diagram 4



F'_0 preload force on the pinion shaft (bearing arrangement)

δ_{01} preload path for the pinion head bearing and surrounding components

δ_{02} preload path for the bearing at the flange side and surrounding components

δ_0 total preload path for pinion bearing arrangement

Adjustment using friction torque

This method is popular in series production because of the short time required and because considerable automation is possible. Since there is a definite relationship between bearing preload and friction torque in the bearing, it is possible to stop adjustment when a friction torque corresponding to the desired preload has been reached if the friction torque is continuously monitored. However, it should be remembered that the friction torque can vary from bearing to bearing, and that it also depends on the preservative used, or on the lubrication conditions and the speed.

Adjustment using direct force measurement

As the purpose of bearing adjustment is to produce a given preload in the bearings, it would seem sensible to use a method either to produce the force directly, or to measure the force directly. However, in practice the indirect methods of adjustment by preload path or friction torque are preferred as they are simple and can be achieved easily and more cost efficiently.

Collective adjustment

With this method of adjustment, which may also be termed “random statistical adjustment”, the bearings, shaft and housing, spacer rings or sleeves etc. are produced in normal quantities and randomly assembled, the components being fully interchangeable. Where taper roller bearings are concerned, this interchangeability also extends to the outer rings and inner ring assemblies. In order not to have to resort to the uneconomic production of very accurate bearings and associated components, it is assumed that the limiting values of the tolerances – statistically – seldom occur together. If, however, the preload force is to be obtained with as little scatter as possible, manufacturing tolerances must be reduced. The advantage of collective adjustment is that no inspection is required and no extra equipment needed when mounting the bearings.

Preloading by springs

By preloading bearings in small electric motors and similar applications it is possible to reduce operating noise. The bearing arrangement in this case comprises a single row deep groove ball bearing at each end of the shaft. The simplest method of applying preload is by a spring or spring "package" (➔ **fig** 42). 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 expansion. The requisite preload force can be estimated from

$$F = k\,d$$

where
F = preload force, kN
k = a factor, see following
d = bearing bore diameter, mm

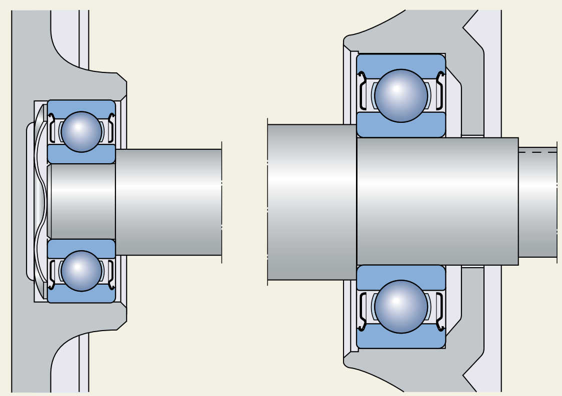
Depending on the design of the electric motor, values of between 0,005 and 0,01 are used for the factor k. If preload is used primarily to protect the bearing from vibration damage when stationary, then greater preload is required and k = 0,02 should be used.

Spring loading is also a common method of applying preload to angular contact ball bearings used in high-speed grinding spindles. The method is not suitable, however, for bearing applications where a high degree of stiffness is required, where the direction of load changes, or where undefined shock loads can occur.

Maintaining the correct preload

When selecting the preload force for a bearing arrangement it should be remembered that the stiffness is only marginally increased when the preload exceeds a given optimum value, whereas friction and consequently heat generation increase and there is a sharp decrease in bearing service life as a result of the additional, constantly acting load. **Diagram** 5 indicates the relationship between bearing life and preload/clearance. Because of the risk which an excessive pre-

Fig 41



load implies for the operational reliability of a bearing arrangement, and because of the complexity of the calculations normally required to establish the appropriate preload force, it is advisable to consult the SKF application engineering service.

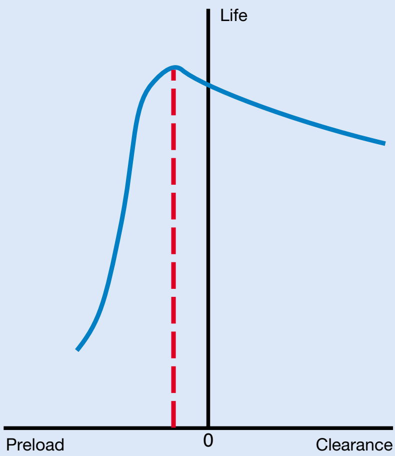
It is also important when adjusting preload in a bearing arrangement, that the established value of the preload force, determined either by calculation or by experience, is achieved with the least possible scatter. This means, for example, for bearing arrangements with taper roller bearings, that the bearings should be turned several times during adjustment so that the rollers do not skew and so that the roller ends are in correct contact with the guide flange of the inner ring. If this is not the case, the results obtained during inspection or by measurements will be false and the final preload can be much smaller than the requisite value.

Bearings for preloaded bearing arrangements

For certain applications, SKF supplies single bearings or matched bearing sets which are specifically made to enable simple and reliable adjustment, or which are matched during manufacture so that after mounting, a predetermined value of the preload will be obtained. These include

- taper roller bearings to the CL7C specifications for automotive pinion and differential bearing arrangements; further details will be found in the section "Single row taper roller bearings", starting on **page 599**,
- single row angular contact ball bearings for universal matching (➔ section "Single row angular contact ball bearings", starting on **page 407**),
- paired single row taper roller bearings, e.g. for industrial gearboxes (➔ section "Paired single row taper roller bearings", starting on **page 667**), and
- paired single row deep groove ball bearings (➔ section "Single row deep groove ball bearings", starting on **page 289**).

Diagram 5



Sealing arrangements

Whatever the bearing arrangement, it consists not only of the bearings but includes associated components. Besides shafts and housings these associated components include the sealing, the performance of which is vital to the cleanliness of the lubricant and the overall service life of the bearing arrangement. For the designer, this means that bearing and sealing arrangement should be viewed as an integrated system and should be treated as such.

Where seals for rolling bearings are concerned, a distinction is made between seals that are integral with the bearing and those that are positioned outside the bearing and are separate from it. Sealed bearings are generally used for arrangements where a sufficiently effective external seal cannot be provided because there is inadequate space or for cost reasons.

Types of seals

The purpose of a seal is to prevent any contaminants from entering into a controlled environment. External seals must be able to prevent media from passing between a stationary and rotating surface, e.g. a housing and shaft. Integral bearing seals must be able to keep contaminants out and lubricant in the bearing cavity.

To be effective, the seal should be sufficiently capable of deformation to be able to compensate for any surface irregularities but also be strong enough to withstand operating pressures. The materials from which the seal is made should also be able to withstand a wide range of operating temperatures, and have appropriate chemical resistance.

There are several seal types; for example, DIN 3750 distinguishes between the following basic types:

- seals in contact with stationary surfaces,
- seals in contact with sliding surfaces,
- non-contact seals,
- bellows and membranes.

The seals in contact with stationary surfaces are known as static seals and their effectiveness depends on the radial or axial deforma-

Fig 42

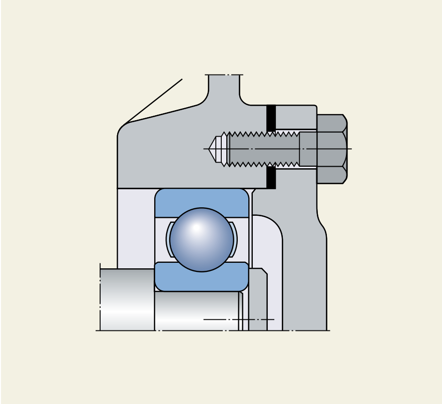


Fig 43

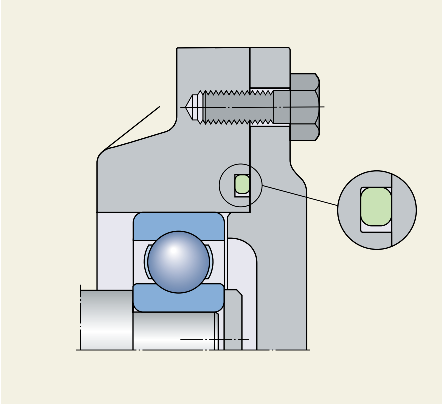
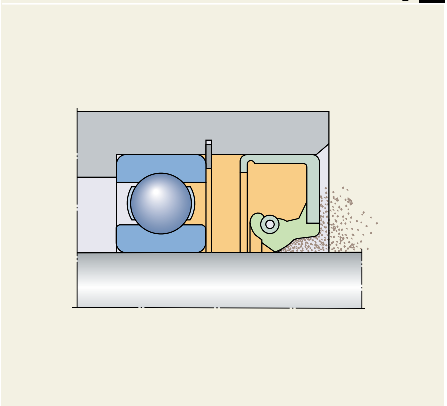


Fig 44



tion of their cross section when installed. Gaskets (➔ **fig** 42) and O-rings (➔ **fig** 43) are typical examples of static seals.

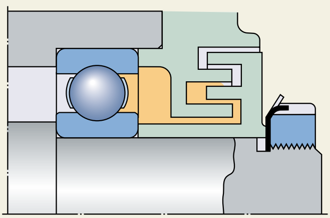
Seals in contact with sliding surfaces are called dynamic seals and are used to seal passages between machine components that move relative to each other either linearly or in the circumferential direction. These dynamic seals have to retain lubricant, exclude contaminants, separate different media and withstand differential pressures. There are various types of dynamic seals, including packing and piston seal rings, which are used for linear or oscillating movements. However, the most common seal is the radial shaft seal (➔ **fig** 44), which is used in a wide variety of applications in all branches of industry.

Non-contact radial shaft seals function by virtue of the sealing effect of a narrow, relatively long gap, which can be arranged axially, radially or in combination. Non-contact seals, which range from simple gap-type seals to multi-stage labyrinths (➔ **fig** 45) are practically without friction and do not wear.

Bellows and membranes are used to seal components that have limited movement relative to each other.

Because of the importance of dynamic radial seals for the efficient sealing of bearing arrangements, the following information deals almost exclusively with radial seals, their various designs and executions.

Fig 45



Selection of seal type

Seals for bearing arrangements should provide a minimum amount of friction and wear while providing maximum protection even under the most arduous conditions. Because bearing performance and service life are so closely tied to the effectiveness of the seal, the influence of contaminants on bearing life is a key design factor. For more information on the influence of contamination on bearing performance, please refer to the section "Selection of bearing size", starting on **page 49**.

Many factors have to be considered when selecting the most suitable seal type for a particular bearing arrangement:

- the lubricant type: oil or grease,
- the peripheral (circumferential) speed at the sealing surface,
- the shaft arrangement: horizontal or vertical,
- possible shaft misalignment,
- available space,
- seal friction and the resulting temperature increase,
- environmental influences, and
- justifiable cost

Selecting the correct seal is of vital importance to the performance of a bearing. It is therefore necessary to accurately specify the sealing requirements and to accurately define the external conditions.

Where full application details are available, reference can be made to the SKF publications:

- Catalogue "CR seals",
- Handbook "Sealing arrangement design guide" or
- "SKF Interactive Engineering Catalogue" on CD-ROM or online at www.skf.com.

If little or no experience is available for a given application, SKF, also one of the largest seal manufacturers in the world, can assist in the selection process or make proposals for suitable seals.

Two types of external sealing devices are normally used with rolling bearings: contact and non-contact seals. The type chosen depends on the needs of the application.

Non-contact seals

The effectiveness of an external non-contact seal depends in principle on the sealing action of the narrow gap between the rotating and stationary components. The gap may be arranged radially, axially or in combination (➔ **fig** 46). These seals can be as simple as a gap-type seal or more complex like a labyrinth seal. In either case, because there is no contact, these seals generate virtually no friction and do not wear. They are generally not easily damaged by solid contaminants and are particularly suitable for high speeds and high temperatures. To enhance their sealing efficiency grease can be pressed into the gap(s) formed by the labyrinth.

Contact seals

The effectiveness of a contact seal depends on the seal's ability to exert a minimum pressure on its counterface by a relatively narrow sealing lip or surface. This pressure (➔ **fig** 47) may be produced either by

- the resilience of the seal, resulting from the elastic properties of the seal material. (**a**)
- the designed interference between the seal and its counterface (**b**) or
- a tangential force exerted by a garter spring incorporated in the seal (**c**).

Contact seals are generally very reliable, particularly when wear is kept to a minimum by producing an appropriate surface finish for the counterface and by lubricating the seal lip/counterface contact area. The friction of the seal on its counterface and the rise in temperature that this generates are a disadvantage and contact seals are therefore only useful for operation up to certain peripheral speeds depending mainly on the seal type and counterface roughness. They are also susceptible to mechanical damage, e.g. as a result of improper mounting, or by solid contaminants. To prevent damage by solid contaminants it is customary to place a non-contact seal in front of a contact seal in order to protect it.

Fig 46

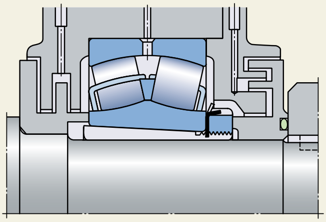
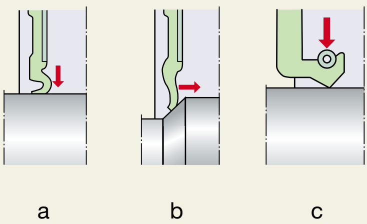


Fig 47



Integral bearing seals
SKF supplies several bearing types fitted with shields or contact seals on one or both sides. These provide an economic and space-saving solution to many sealing problems. Bearings with shields or seals on both sides are supplied already greased and are generally maintenance-free. Actual seal designs are described in detail in the introductory text to the relevant bearing table sections.

Bearings with shields

Bearings fitted with shields (➔ **fig** 48), are used for arrangements where contamination is not heavy and where there is no danger of water, steam etc. coming into contact with the bearing. Shields are also used in applications where reduced friction is important due to speed or operating temperature considerations.

Shields are made from sheet steel and form

- a long sealing gap with the land of the inner ring shoulder (**a**) or
- an efficient labyrinth seal with a recess in the inner ring shoulder (**b**).

Bearings with contact seals

Bearings with contact seals, referred to simply as seals are preferred for arrangements were contamination is moderate and where the presence of moisture or water spray etc. cannot be ruled out, or where a long service life without maintenance is required.

SKF has developed a series of seals (➔ **fig** 49). Depending on the bearing type and/or size the bearings may be equipped with standard seals which seal against

- the inner ring shoulder (**a**) and/or against a recess in the inner ring shoulder (**b**, **c**), or
- the lead-in at the sides of the inner ring raceway (**d**, **e**) or the outer ring (**f**).

Fig 48

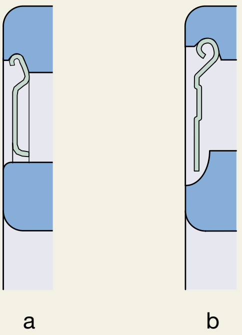
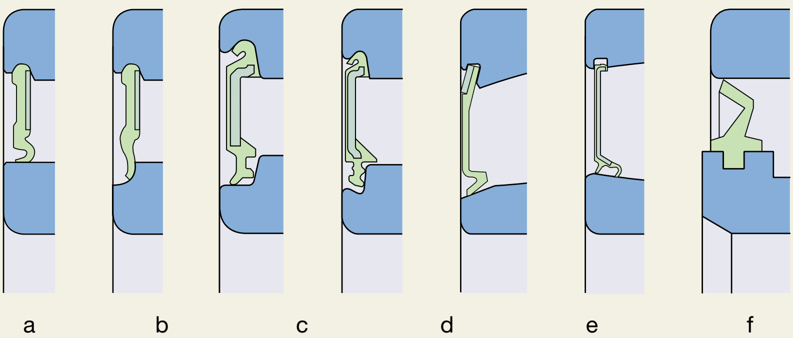


Fig 49



For deep groove ball bearings SKF has developed two additional seal types (➔ **fig** 50), referred to as

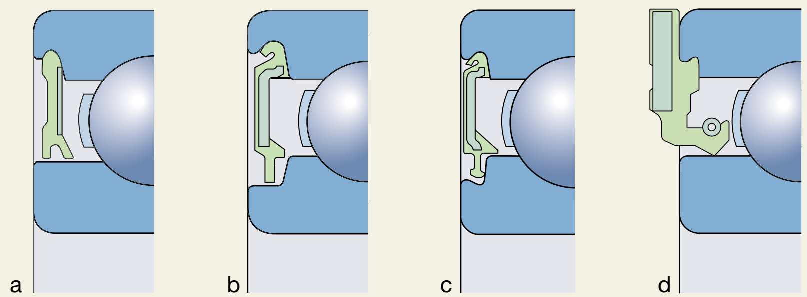
- the low-friction seal (**a**, **b**, **c**), which is practically without contact and fulfils high demands on sealing and the low-friction operation of the bearing.
- the spring-loaded radial shaft Waveseal® (**d**), which is incorporated on one side and together with the bearing, form the ICOS™ oil sealed bearing unit.

Seals integrated in SKF bearings are generally made of elastomer materials and reinforced by sheet steel. Depending on the series, size and the application requirements, the seals are generally produced from

- acrylonitrile butadiene rubber (NBR)
- hydrogenated acrylonitrile butadiene rubber (HNBR)
- fluoro rubber (FPM)

The selection of the appropriate seal material depends on the expected operating temperature and the applied lubricant. Concerning the permissible operating temperatures, please refer to section "Seal materials", starting on **page 142**.

Fig 50



External seals

For bearing arrangements where the efficiency of the seal under the given operating conditions is more important than space considerations or cost, there are several possible seal types to choose from.

The seals offered by SKF are given special attention in the following section. Many ready-to-mount external seals are available commercially. For seals that are not part of the SKF range, the information provided in the following section is to be used as a guideline only. SKF takes no responsibility for the performance of these non-SKF products. Be sure to check with the seal manufacturer before designing any seal into an application.

Non-contact seals

The simplest seal used outside the bearing is the gap-type seal, which creates a small gap between the shaft and housing (➔ **fig** 51). This type of seal is adequate for grease lubricated applications that operate in dry, dust-free environments. To enhance the efficiency of this seal, one or more concentric grooves can be machined in the housing bore at the shaft exit (➔ **fig** 52). The grease emerging through the gap fills the grooves and helps to prevent the entry of contaminants.

With oil lubrication and horizontal shafts, helical grooves – right-hand or left-hand depending on the direction of rotation of the shaft – can be provided in the shaft or housing bore (➔ **fig** 53). These serve to return emerging oil to the bearing position. It is essential here that the direction of shaft rotation does not change.

Fig 51

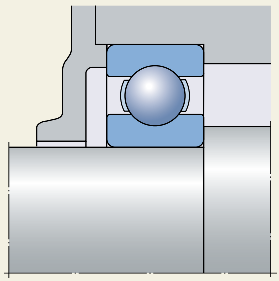


Fig 52

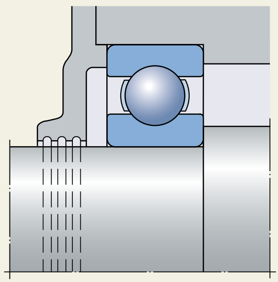


Fig 53

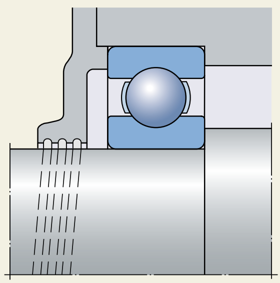
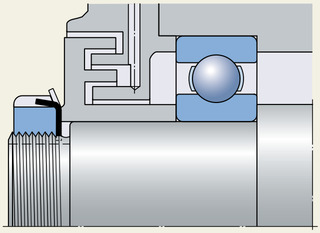


Fig 54



Single or multi-stage labyrinth seals are considerably more effective than simple gap-type seals, but are more expensive to produce. They are chiefly used with grease lubrication. Their efficiency can be further improved by periodically applying a water-insoluble grease, e.g. a grease with a lithium-calcium thickener, via a duct to the labyrinth passages. The tongues of the labyrinth seal are arranged axially (➔ **fig** 54) for one-piece housings and radially (➔ **fig** 55) for split housings. The width of the axial passages of the labyrinth remains unchanged when axial displacement of the shaft occurs in operation and can thus be very narrow. If angular misalignment of the shaft with respect to the housing can occur, labyrinths with inclined passages are used (➔ **fig** 56).

Effective and inexpensive labyrinth seals can be made using commercially available products, e.g. using SKF sealing washers (➔ **fig** 57). Sealing efficiency increases with the number of washer sets used, or can be further enhanced by incorporating flocked washers. Additional information on these sealing washers can be found in the section “Seals” in the “SKF Interactive Engineering Catalogue”, available on CD-ROM or online at www.skf.com.

Fig 55

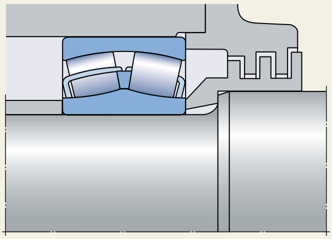


Fig 56

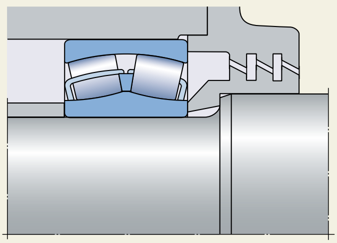
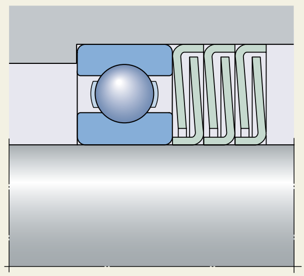


Fig 57



Rotating discs (➔ **fig** 58) are often fitted to the shaft to improve the sealing action of shields, and flinger rings, grooves or discs are used for the same purpose with oil lubrication. The oil from the flinger is collected in a channel in the housing and returned to the inside of the housing through suitable ducts (➔ **fig** 59).

Contact seals

Radial shaft seals are contact seals that are used, above all, for sealing oil-lubricated bearings. These ready-to-mount elastomer sealing components normally have a metal reinforcement or casing. The sealing lip is usually a synthetic rubber and is normally pressed against a counterface on the shaft by a garter spring. Depending on the seal material and medium to be retained and/or excluded, radial shaft seals can be used at temperatures between –60 and +190 °C.

The contact area between the sealing lip and counterface is of vital importance to sealing efficiency. The surface hardness of the counterface should normally be at least 55 HRC and the hardened depth should be at least 0,3 mm, the surface roughness to ISO 4288:1996 should be within the guidelines of R_a = 0,2 to 0,8 µm. In applications, where speeds are low, lubrication is good and contamination is minimal, a lower hardness can be acceptable. To avoid the pumping action produced by helical grinding marks, plunge grinding is recommended. If the main purpose of the radial shaft seal is to prevent lubricant from leaving the housing, the seal should be mounted with the lip facing inwards (➔ **fig** 60). If the main purpose is to exclude contaminants, the lip should face outwards, away from the bearing (➔ **fig** 61, **page 226**).

Fig 58

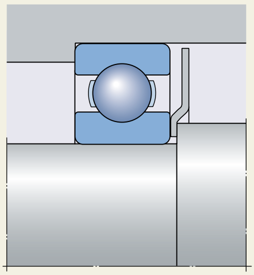


Fig 59

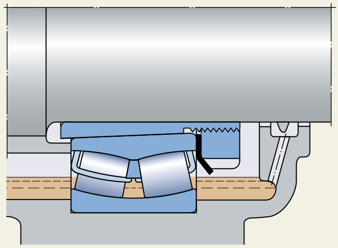


Fig 60

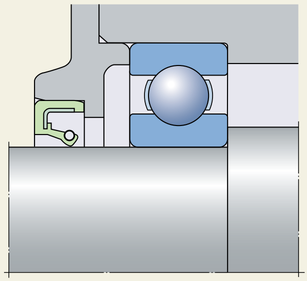


Fig 61

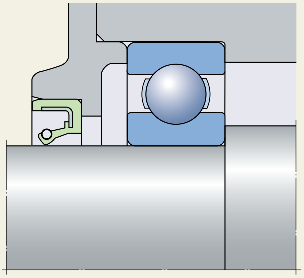


Fig 62

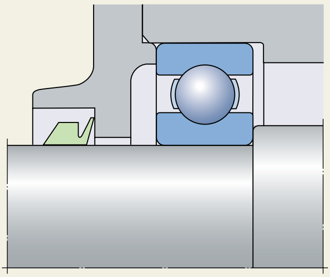
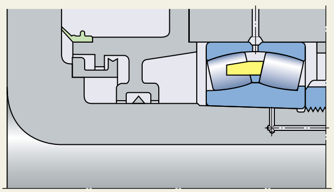


Fig 63



V-ring seals (➔ **fig** 62) can be used both with oil and with grease lubrication. The elastic rubber ring (body) of the seal firmly grips the shaft and rotates with it, while the sealing lip exerts a light axial pressure on the stationary component, e.g. the housing. Depending on the material, V-rings can be used at operating temperatures between –40 and +150 °C. They are simple to install and at low speeds permit relatively large angular misalignments of the shaft. A surface roughness R_a of between 2 and 3 µm is sufficient for the counterface. At peripheral speeds above 8 m/s the V-ring must be axially located on the shaft. At speeds above 12 m/s the ring must be prevented from “lifting” from the shaft by, for example, a sheet metal support ring. When the peripheral speed exceeds 15 m/s the sealing lip will lift from the counterface so that the V-ring becomes a gap-type seal. The good sealing action of the V-ring depends mainly on the fact that the ring body acts as a flinger, repelling dirt and fluids. Therefore, with grease lubrication the seal is generally arranged outside the housing, whereas for oil lubrication it is normally arranged inside the housing with the lip pointing away from the bearing position. Used as a secondary seal, V-rings protect the primary seal from excessive contaminants and moisture.

Axial clamp seals (➔ **fig** 63) 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 roughness R_a of 2,5 µm.

Mechanical seals (➔ **fig** 64) are used to seal grease or oil lubricated bearing positions where speeds are relatively low and operating conditions difficult and arduous. They consist of two sliding steel rings with finely finished sealing surfaces and two plastic cup springs (Belleville washers), which position the sliding rings in the housing bore and provide the necessary preload force to the sealing surfaces. There are no special demands on the mating surfaces in the housing bore.

Felt seals (➔ **fig** 65) are generally used with grease lubrication. They are simple and inexpensive and can be used at peripheral speeds of up to 4 m/s and at operating temperatures up to +100 °C. The counterface should be ground to a surface roughness $R_a \leq 3,2$ µm. The efficiency of a felt seal can be much improved by mounting a simple labyrinth seal as a secondary seal. Before being inserted in the housing groove, the felt rings or strips should be soaked in oil at about 80 °C.

Spring washers (➔ **fig** 66) provide simple, inexpensive and space-saving seals for grease lubricated rigid bearings, particularly deep groove ball bearings. The washers are clamped against either the outer ring or the inner ring and exert a resilient pressure axially against the other ring. After a certain running-in period these seals become non-contact seals by forming a very narrow gap-type seal.

More detailed information on seals supplied by SKF will be found in the SKF catalogue “CR seals” and seals incorporated in SKF products are also described in detail in literature dealing with these products.

Fig 64

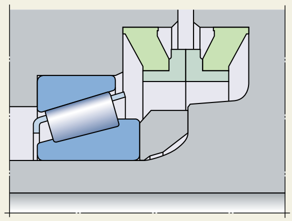


Fig 65

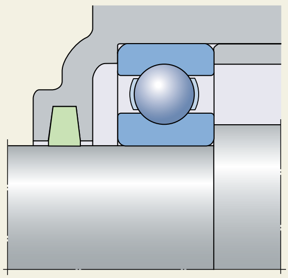
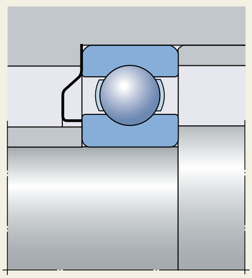
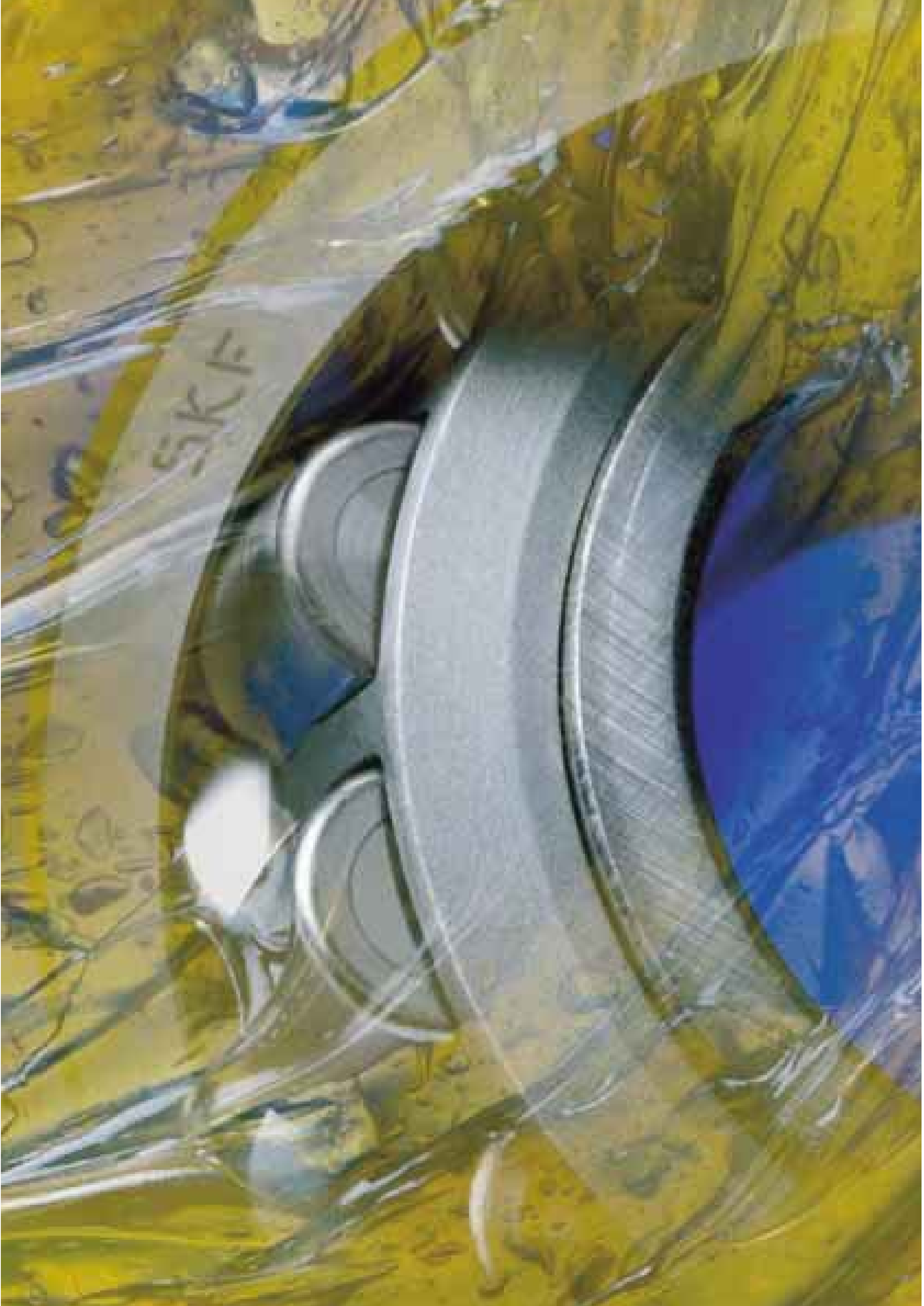


Fig 66




SKF

Lubrication

If rolling bearings are to operate reliably they must be adequately lubricated to prevent direct metal-to-metal contact between the rolling elements, raceways and cages. The lubricant also inhibits wear and protects the bearing surfaces against corrosion. The choice of a suitable lubricant and method of lubrication for each individual bearing application is therefore important, as is correct maintenance.

A wide selection of greases and oils is available for the lubrication of rolling bearings and there are also solid lubricants, e.g. for extreme temperature conditions. The actual choice of a lubricant depends primarily on the operating conditions, i.e. the temperature range and speeds as well as the influence of the surroundings.

The most favourable operating temperatures will be obtained when the minimum amount of lubricant needed for reliable bearing lubrication is provided. However, when the lubricant has additional functions, such as sealing or the removal of heat, additional amounts of lubricant may be required.

The lubricant in a bearing arrangement gradually loses its lubricating properties as a result of mechanical work, ageing and the build-up of contamination. It is therefore necessary for grease to be replenished or renewed and for oil to be filtered and changed at regular intervals.

The information and recommendations in this section relate to bearings without integral seals or shields.

SKF bearings and bearing units with integral seals and shields at both sides are supplied greased. Information about the greases used by SKF as standard for these products can be found in the text preceding the relevant product tables together with a brief description of the performance data.

The service life of the grease in sealed bearings most often exceeds bearing life so that, with some exceptions, no provision is made for the relubrication of these bearings.

Note:
Differences in the lubricating properties of seemingly identical lubricants – particularly grease – produced at different locations can exist. Therefore, SKF cannot accept liability for any lubricant or its performance. The user is therefore advised to specify lubricant properties in detail so as to obtain the most suitable lubricant for the application.

Grease lubrication

Grease can be used to lubricate rolling bearings under normal operating conditions in the majority of applications.

Grease has the advantage over oil that it is more easily retained in the bearing arrangement, particularly where shafts are inclined or vertical, and it also contributes to sealing the arrangement against contaminants, moisture or water.

Excessive amounts of grease will cause the operating temperature within the bearing to rise rapidly, particularly when running at high speeds. As a general rule, when starting up only the bearing should be completely filled, while the free space in the housing should be partly filled with grease. Before operating at full speed, the excess grease in the bearing must be allowed to settle or escape during a running-in period. At the end of the running-in period the operating temperature will drop considerably indicating that the grease has been distributed in the bearing arrangement.

However, where bearings are to operate at very low speeds and good protection against contamination and corrosion is required, it is advisable to fill the housing completely with grease.

Lubricating greases

Lubricating greases consist of a mineral or synthetic oil combined with a thickener. The thickeners are usually metallic soaps. However, other thickeners, e.g. polyurea can be used for superior performance in certain areas, i.e. high temperature applications. Additives can also be included to enhance certain properties of the grease. The consistency of the grease depends largely on the type and concentration of the thickener used and on the operating temperature of the application. When selecting a grease, the consistency, operating temperature range, viscosity of the base oil, rust inhibiting properties and the load carrying ability are the most important factors to be considered. Detailed information on these properties follows.

Base oil viscosity

The importance of the oil viscosity for the formation of an oil film to separate the bearing surfaces and thus for the life of the bearing is dealt with in the section “Lubrication conditions – the viscosity ratio κ” on **page 59**; the information applies equally to the base oil viscosity of greases.

The base oil viscosity of the greases normally used for rolling bearings lies between 15 and 500 mm^2/s at 40 °C. Greases based on oils having higher viscosities than 1 000 mm^2/s at 40 °C bleed oil so slowly that the bearing will not be adequately lubricated. Therefore, if a calculated viscosity well above 1 000 mm^2/s at 40 °C is required because of low speeds, it is better to use a grease with a maximum viscosity of 1 000 mm^2/s and good oil bleeding properties or to apply oil lubrication.

The base oil viscosity also governs the maximum recommended speed at which a given grease can be used for bearing lubrication. The permissible rotational speed for grease is also influenced by the shear strength of the grease, which is determined by the thickener. To indicate the speed capability, grease manufacturers often quote a "speed factor"

$$A = n\, d_m$$

where

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

For applications operating at very high speeds, e.g. at A > 700 000 for ball bearings, the most suitable greases are those incorporating base oils of low viscosity.

Consistency
Greases are divided into various consistency classes according to the National Lubricating Grease Institute (NLGI) scale. The consistency of grease used for bearing lubrication should not change drastically when operated within its specified temperature range after mechanical working. Greases that soften at elevated temperatures may leak from the bearing arrangement. Those that stiffen at low temperatures may restrict rotation of the bearing or have insufficient oil bleeding.

Metallic soap thickened greases, with a consistency of 1, 2 or 3 are used for rolling bearings. The most common greases have a consistency of 2. Lower consistency greases are preferred for low temperature applications, or for improved pumpability. Consistency 3 greases are recommended for bearing arrangements with a vertical shaft, where a baffle plate is arranged beneath the bearing to prevent the grease from leaving the bearing.

In applications subjected to vibration, the grease is heavily worked as it is continuously thrown back into the bearing by vibration. Higher consistency greases may help here, but stiffness alone does not necessarily provide adequate lubrication. Therefore mechanically stable greases should be used instead.

Greases thickened with polyurea can soften or harden depending on the shear rate in the application. In applications with vertical shafts there is a danger that a polyurea grease will leak under certain conditions.

Temperature range – the SKF traffic light concept
The temperature range over which a grease can be used depends largely on the type of base oil and thickener used as well as the additives. The relevant temperatures are schematically illustrated in **diagram 1** in the form of a "double traffic light".

The extreme temperature limits, i.e. low temperature limit and the high temperature limit, are well defined.

- The low temperature limit (LTL), i.e. the lowest temperature at which the grease will allow the bearing to be started up without difficulty, is largely determined by the type of base oil and its viscosity.
- The high temperature limit (HTL) is determined by the type of thickener and for soap base greases it is given by the dropping point. The dropping point indicates the temperature at which the grease loses its consistency and becomes a fluid.

It is evident that operation below the low temperature limit and above the high temperature limit is not advised as shown in **diagram 1** by the red zones. Although grease suppliers indicate the specific values for the low and high temperature limits in their product information, the really important temperatures for reliable operation are given by the SKF values for

- the low temperature performance limit (LTPL) and
- the high temperature performance limit (HTPL).

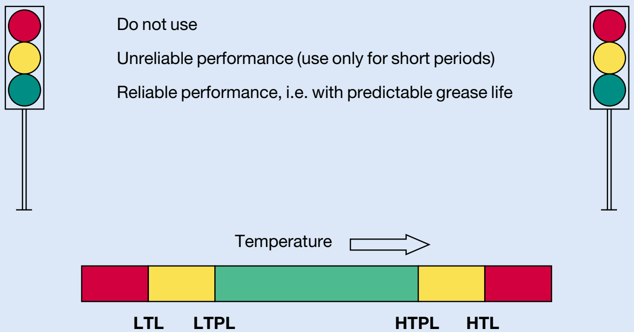
It is within these two limits, the green zone in **diagram** 1, where the grease will function reliably and grease life can be determined accurately, Since the definition of the high temperature performance limit is not standardized internationally care must be taken when interpreting suppliers' data.

At temperatures above the high temperature performance limit (HTPL), grease will age and oxidize with increasing rapidity and the by-products of the oxidation will have a detrimental effect on lubrication. Therefore, temperatures in the amber zone, between the high temperature performance limit and the high temperature limit (HTL) should occur only for very short periods.

An amber zone also exists for low temperatures. With decreasing temperature, the tendency of grease to bleed decreases and the stiffness (consistency) of the grease increases. This will ultimately lead to an insufficient supply of lubricant to the contact surfaces of the rolling elements and raceways. In **diagram** 1, this temperature limit is indicated by the low temperature performance limit (LTPL). Values for the low temperature performance limit are different for roller and ball bearings. Since ball bearings are easier to lubricate than roller bearings, the low temperature performance limit is less important for ball bearings. For roller bearings, however, serious damage will result when the bearings are operated continuously below this limit. Short periods in this zone e.g. during a cold start, are not harmful since the heat caused by friction will bring the bearing temperature into the green zone.

Diagram 1

The SKF traffic light concept



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

Note:
The SKF traffic light concept is applicable for any grease; however, the temperature zones differ from grease to grease and can only be determined by functional bearing testing. The traffic light limits for

- grease types normally used for rolling bearings are shown in **diagram** 2 and for
- SKF greases are shown in **diagram** 3.

The values shown in these diagrams are based on extensive tests conducted in SKF laboratories and may differ from those quoted by lubricant manufacturers. They are valid for commonly available NLGI 2 greases without EP additives. The temperatures in the diagrams relate to the observed self-induced bearing temperature (usually measured on the non-rotating ring). Since the data for each grease type is a summary of many greases of more or less similar composition, the transitions for each group are not sharp but fall within a small range.

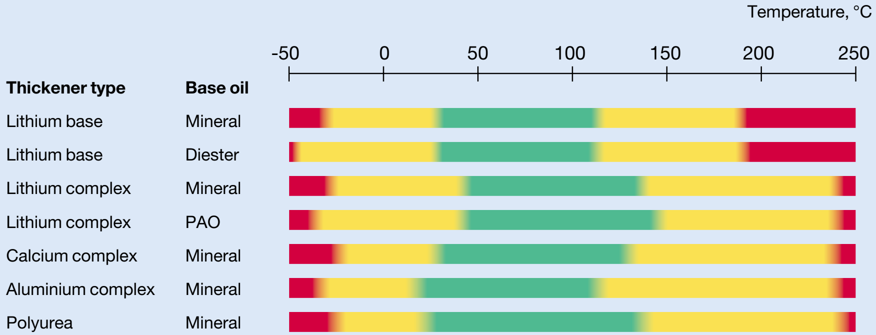
Protection against corrosion, behaviour in the presence of water
Grease should protect the bearing against corrosion and should not be washed out of the bearing in cases of water penetration. The thickener type solely determines the resistance to water: lithium complex, calcium complex and polyurea greases offer usually very good resistance. The type of rust inhibitor additive mainly determines the rust inhibiting properties of greases.

At very low speeds, a full grease pack is beneficial for corrosion protection and for the prevention of water ingress.

Load carrying ability: EP and AW additives
Bearing life is shortened if the lubricant film thickness is not sufficient to prevent metal-to-metal contact of the asperities on the contact surfaces. One option to overcome this is to use so-called EP (Extreme Pressure) additives. High temperatures induced by local asperity contact, activate these additives promoting mild wear at the points of contact. The result is a smoother surface, lower contact stresses and an increase in service life.

Diagram 2

The SKF traffic light concept – standard greases



Many modern EP additives are of the sulphur/phosphorus type. Unfortunately these additives may have a negative effect on the strength of the bearing steel matrix. If such additives are used then the chemical activity may not be restricted to the asperity contacts. If the operating temperature and contact stresses are too high, the additives may become chemically reactive even without asperity contact. This can promote corrosion/diffusion mechanisms in the contacts and may lead to accelerated bearing failure usually initiated by micro pitting. Therefore, SKF recommends the use of less reactive EP additives for operating temperatures above 80 °C. Lubricants with EP additives should not be used for bearings operating at temperatures higher than 100 °C. For very low speeds, solid lubricant additives such as graphite and molybdenum disulphide (MoS_2) are sometimes included in the additive package to enhance the EP effect. These additives should have a high purity level and a very small particle size; otherwise dents due to overrolling of the particles might reduce bearing fatigue life.

AW (Anti-Wear) additives have a function similar to that of EP additives, i.e. to prevent severe metal-to-metal contact. Therefore EP and AW additives are very often not differentiated between. However, the way they work is different. The main difference is that an AW additive builds a protective layer that adheres to the surface. The asperities are then shear-

Diagram 3

The SKF traffic light concept – SKF greases

For operating temperatures above 150 °C, SKF LGET 2 is recommended

ing over each other rather than through each other. The roughness is not reduced by mild wear as in the case of EP additives. Here too special care has to be taken; AW additives may contain elements that, in the same way as the EP additives, can migrate into the bearing steel and weaken the structure.

Certain thickeners (e.g. calcium sulphonate complex) also provide an EP/AW effect without chemical activity and the resulting effect on bearing fatigue life. Therefore, the operating temperature limits for EP additives do not apply for these greases.

If the lubricant film thickness is sufficient, SKF does not generally recommend the use of EP and AW additives. However there are circumstances where EP/AW additives may be useful. If excessive sliding between the rollers and raceways is expected they may be beneficial. Contact the SKF application engineering service for further information.

Miscibility

If it becomes necessary to change from one grease to another, the miscibility or the ability to mix greases without adverse effects should be considered. If incompatible greases are mixed, the resulting consistency can change dramatically so that bearing damage e.g. due to severe leakage, could result.

Greases having the same thickener and similar base oils can generally be mixed without any detrimental consequences, e.g. a lithium thickener/mineral oil grease can generally be mixed with another lithium thickener/mineral oil grease. Also, some greases with different thickeners e.g. calcium complex and lithium complex greases, are miscible with each other.

In bearing arrangements where a low grease consistency might lead to grease escaping from the arrangement, the next relubrication should involve purging all the old grease from the arrangement rather than replenishing it (➔ section “Relubrication”, starting on **page 237**).

The preservative with which SKF bearings are treated is compatible with the majority of rolling bearing greases with the possible exception of polyurea greases (➔ section “Preparations for mounting and dismounting” on **page 258**). Modern polyurea greases (e.g. SKF grease LGHP 2) tend to be more compatible with preservatives than some of the older polyurea greases. Note that greases with a PTFE base oil, e.g. SKF LGET 2 grease are not compatible with standard preservatives and the preservatives must be removed before applying grease. Contact the SKF application engineering service for further information.

SKF greases

The SKF range of lubricating greases for rolling bearings comprises many types of grease and covers virtually all application requirements. These greases have been developed based on the latest information regarding rolling bearing lubrication and have been thoroughly tested both in the laboratory and in the field. Their quality is continuously monitored by SKF.

The most important technical specifications on SKF greases are given in **table 2** on **pages 246** and **247** together with a quick selection guide. The temperature ranges where the SKF greases can be used are schematically illustrated in **diagram 3**, **page 235**, according to the SKF traffic light concept.

Further information on SKF greases can be found in the catalogue “SKF Maintenance and Lubrication Products” or online at www.mapro.skf.com.

For a more detailed selection of the appropriate grease for a specific bearing type and application, use the Internet based SKF grease selection program “LubeSelect”. This program can be found online at www.aptitudexchange.com.

Relubrication

Rolling bearings have to be relubricated if the service life of the grease is shorter than the expected service life of the bearing. Relubrication should always be undertaken at a time when the condition of the existing lubricant is still satisfactory.

The time at which relubrication should be undertaken depends on many related factors. These include bearing type and size, speed, operating temperature, grease type, space around the bearing and the bearing environment. It is only possible to base recommendations on statistical rules; the SKF relubrication intervals are defined as the time period, at the end of which 99 % of the bearings are still reliably lubricated. This represents the L_1 grease life.

SKF recommends using experience based on data from actual applications and tests, together with the estimated relubrication intervals provided hereafter.

Relubrication intervals

The relubrication intervals t_f for bearings on horizontal shafts under normal and clean conditions can be obtained from **diagram** 4 as a function of

- the speed factor A multiplied by the relevant bearing factor b_f where
  A = n d_m
  n = rotational speed, r/min
  d_m = bearing mean diameter
  = 0,5 (d + D), mm
  b_f = bearing factor depending on bearing type and load conditions (➔ **table** 1, **page 239**)
- the load ratio C/P

The relubrication interval t_f is an estimated value, valid for an operating temperature of 70 °C, using good quality lithium thickener/mineral oil greases. When bearing operating conditions differ, adjust the relubrication intervals obtained from **diagram** 4 according to the information given under “Deviating operating conditions and bearing types”.

If the speed factor A exceeds a value of 70 % of the recommended limit according to **table** 1, or if ambient temperatures are high, then the use of the calculations presented in the section “Speeds and vibration”, starting on **page 107**, is recommended to check the operating temperature and the proper lubrication method.

When using high performance greases, a longer relubrication interval and grease life may be possible. Contact the SKF application engineering service for additional information

Diagram 4

Relubrication intervals at operating temperatures of 70 °C

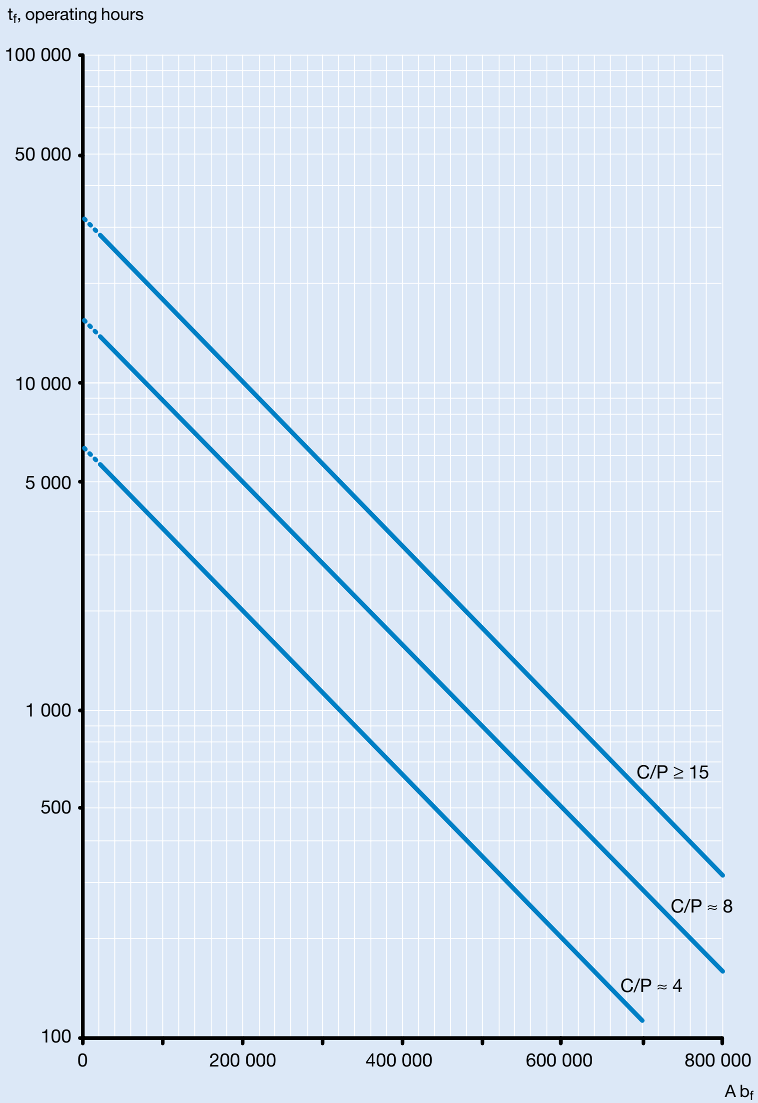


Table 1

Bearing factors and recommended limits for speed factor A

Bearing type[1]	**Bearing factor b_f**	**Recommended limits for speed factor A 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
– full complement[2]	4	NA[3]	NA[3]	20 000
Taper roller bearings	2	350 000	300 000	200 000
Spherical roller bearings				
– when load ratio $F_a/F_r < e$ and $d_m \leq 800$ mm				
series 222, 238, 239	2	350 000	200 000	100 000
series 213, 223, 230, 231, 232, 240, 248, 249	2	250 000	150 000	80 000
series 241	2	150 000	80 000[4]	50 000[4]
– when load ratio $F_a/F_r < e$ 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[4]	30 000[4]
– when load ratio $F_a/F_r > e$				
all series	6	150 000	50 000[4]	30 000[4]
CARB toroidal roller bearings				
– with cage	2	350 000	200 000	100 000
– without 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
Spherical roller thrust bearings				
– rotating shaft washer	4	200 000	170 000	150 000

[1] The bearing factors and recommended practical speed factor “A” limits apply to bearings with standard internal geometry and standard cage execution. For alternative internal bearing design and special cage execution, please contact the SKF application engineering service
[2] The t_f value obtained from **diagram 4** needs to be divided by a factor of 10
[3] Not applicable, for these C/P values a caged bearing is recommended instead
[4] For higher speeds oil lubrication is recommended

Deviating operating conditions and bearing types

Operating temperature

To account for the accelerated ageing of grease with increasing temperature, it is recommended halving the intervals obtained from the **diagram** 4 for every 15 °C increase in operating temperature above 70 °C, remembering that the high temperature performance limit for the grease (➔ **diagram** 1, HTPL) should not be exceeded.

The relubrication interval t_f may be extended at temperatures below 70 °C if the temperature is not close to the lower temperature performance limit (➔ **diagram** 1, LTPL). A total extension of the relubrication interval t_f by more than a factor of two never is recommended. In case of full complement bearings and thrust roller bearings, t_f values obtained from **diagram** 4 should not be extended.

Moreover, it is not advisable to use relubrication intervals in excess of 30 000 hours.

For many applications, there is a practical grease lubrication limit, when the bearing ring with the highest temperature exceeds an operating temperature of 100 °C. Above this temperature special greases should be used. In addition, the temperature stability of the bearing and premature seal failure should be taken into consideration.

For high temperature applications please consult the SKF application engineering service.

Vertical shaft

For bearings on vertical shafts, the intervals obtained from **diagram** 4 should be halved. The use of a good sealing or retaining shield is a prerequisite to prevent grease leaking from the bearing arrangement.

Vibration

Moderate vibration will not have a negative effect on grease life, but high vibration and shock levels, such as those in vibrating screen applications, will cause the grease to churn. In these cases the relubrication interval should be reduced. If the grease becomes too soft, grease with a better mechanical stability, e.g. SKF grease LGHB 2 or grease with higher stiffness up to NLGI 3 should be used.

Outer ring rotation

In applications where the outer ring rotates, the speed factor A is calculated differently: in this case use the bearing outside diameter D instead of d_m. The use of a good sealing mechanism is a prerequisite in order to avoid grease loss.

Under conditions of high outer ring speeds (i.e. > 40 % of the reference speed listed in the product tables), greases with a reduced bleeding tendency should be selected.

For spherical roller thrust bearings with a rotating housing washer oil lubrication is recommended.

Contamination

In case of ingress of contamination, more frequent relubrication than indicated by the relubrication interval will reduce the negative effects of foreign particles on the grease while reducing the damaging effects caused by overrolling the particles. Fluid contaminants (water, process fluids) also call for a reduced interval. In case of severe contamination, continuous relubrication should be considered.

Very low speeds

Bearings that operate at very low speeds under light loads call for a grease with low consistency while bearings that operate at low speeds and heavy loads need to be lubricated by high viscosity greases, and if possible, with very good EP characteristics. Solid additives such as graphite and molybdenum disulphide (MoS_2) can be considered for a speed factor A < 20 000. Selecting the proper grease and grease fill is very important in low speed applications.

High speeds

Relubrication intervals for bearings used at high speeds i.e. above the recommended speed factor A given in **table** 1, **page 239**, only apply when using special greases or modified bearing executions, e.g. hybrid bearings. In these cases continuous relubrication techniques such as circulating oil, oil-spot etc are more suitable than grease lubrication.

Very heavy loads

For bearings operating at a speed factor A > 20 000 and subjected to a load ratio C/P < 4 the relubrication interval is further reduced. Under these very heavy load conditions, continuous grease relubrication or oil bath lubrication is recommended.

In applications where the speed factor A < 20 000 and the load ratio C/P = 1–2 reference should be made to the information under "Very low speeds" on **page 240**. For heavy loads and high speeds circulating oil lubrication with cooling is generally recommended.

Very light loads

In many cases the relubrication interval may be extended if the loads are light (C/P = 30 to 50). To obtain satisfactory operation the bearings should be subjected at least to the minimum load as stated in the text preceding the relevant product tables.

Misalignment

A constant misalignment within the permissible limits does not adversely affect the grease life in spherical roller bearings, self-aligning ball bearings or toroidal roller bearings.

Large bearings

To establish a proper relubrication interval for line contact bearings, in particular large roller bearings (d > 300 mm) used in critical bearing arrangements in process industries an interactive procedure is recommended. In these cases it is advisable to initially relubricate more frequently and adhere strictly to the recommended regreasing quantities (➔ section "Relubrication procedures" on **page 242**).

Before regreasing, the appearance of the used grease and the degree of contamination due to particles and water should be checked. Also the seal should be checked completely, looking for wear, damage and leaks. When the condition of the grease and associated components is found to be satisfactory, the relubrication interval can be gradually increased.

A similar procedure is recommended for spherical roller thrust bearings, prototype machines and upgrades of high-density power equipment or wherever application experience is limited.

Cylindrical roller bearings

The relubrication intervals from **diagram** 4 are valid for cylindrical roller bearings fitted with

- a moulded glass fibre reinforced polyamide 6,6 cage, designation suffix P
- a two-piece machined brass cage guided by the rollers, designation suffix M.

For bearings with a pressed steel cage, designation suffix J, or with an inner or outer ring centred cage, designation suffixes MA, ML and MP, the value for the relubrication interval from **diagram** 4 should be halved. Moreover, grease with good oil bleeding properties should be applied. For bearings with MA, MB, ML or MP cage generally oil lubrication should be preferred.

Observations

If the determined value for the relubrication interval t_f is too short for a particular application, it is recommended to

- check the bearing operating temperature,
- check whether the grease is contaminated by solid particles or fluids,
- check the bearing application conditions such as load or misalignment

and, last but not least, a more suitable grease should be considered.

Relubrication procedures

The choice of the relubrication procedure generally depends on the application and on the relubrication interval t_f obtained.

- Replenishment is a convenient and preferred procedure if the relubrication interval is shorter than six months. It allows uninterrupted operation and provides, when compared with continuous relubrication, a lower steady state temperature.
- Renewing the grease fill is generally recommended when the relubrication intervals are longer than six months. This procedure is often applied as part of a bearing maintenance schedule e.g. in railway applications.
- Continuous relubrication is used when the estimated relubrication intervals are short, e.g. due to the adverse effects of contamination, or when other procedures of relubrication are inconvenient because access to the bearing is difficult. Continuous relubrication is not recommended for applications with high rotational speeds since the intensive churning of the grease can lead to very high operating temperatures and destruction of the grease thickener structure.

When using different bearings in a bearing arrangement it is common practice to apply the lowest estimated relubrication interval for both bearings. The guidelines and grease quantities for the three alternative procedures are given below.

Replenishment
As mentioned in the introduction of the grease lubrication section, the bearing should initially be completely filled, while the free space in the housing should be partly filled. Depending on the intended method of replenishment, the following grease fill percentages for this free space in the housing are recommended:

- 40 % when replenishing is made from the side of the bearing (➔ **fig 1**);
- 20 % when replenishing is made through the annular groove and lubrication holes in the bearing outer or inner ring (➔ **fig 2**).

Suitable quantities for replenishment from the side of a bearing can be obtained from

$$G_p = 0{,}005\ D\ B$$

and for replenishment through the bearing outer or inner ring from

$$G_p = 0{,}002\ D\ B$$

Fig 1

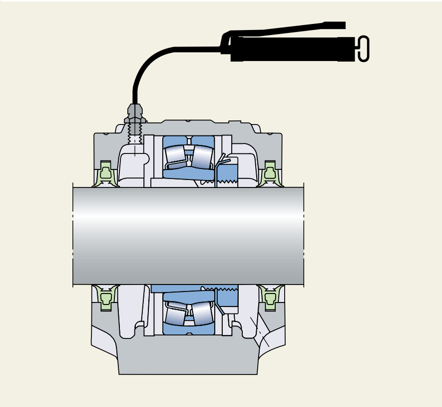
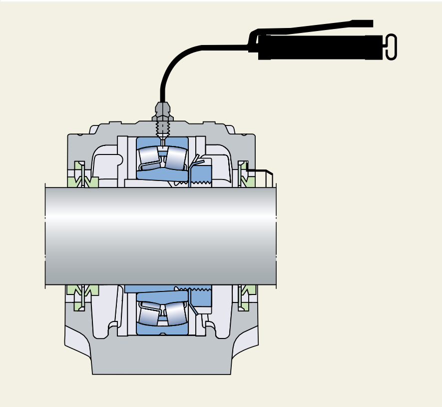


Fig 2



where

G_p = grease quantity to be added when replenishing, g
D = bearing outside diameter, mm
B = total bearing width (for thrust bearings use height H), mm

To facilitate the supply of grease using a grease gun, a grease nipple must be provided on the housing. If contact seals are used, an exit hole in the housing should also be provided so that excessive amounts of grease will not build up in the space surrounding the bearing (➔ **fig** 1) as this might cause a permanent increase in bearing temperature. The exit hole should be plugged when high-pressure water is used for cleaning.

The danger of excess grease collecting in the space surrounding the bearing and causing temperature peaks, with its detrimental effect on the grease as well as the bearing, is more pronounced when bearings operate at high speeds. In these cases it is advisable to use a grease escape valve rather than an exit hole. This prevents over-lubrication and allows relubrication to be performed while the machine is in operation. A grease escape valve consists basically of a disc that rotates with the shaft and which forms a narrow gap together with the housing end cover (➔ **fig** 3). Excess and used grease are thrown out by the disc into an annular cavity and leaves the housing through an opening on the underside of the end cover. Additional information regarding the design and dimensioning of grease escape valves can be supplied on request.

To be sure that fresh grease actually reaches the bearing and replaces the old grease, the lubrication duct in the housing should either feed the grease adjacent to the outer ring side face (➔ **figs** 1 and 4) or, better still, into the bearing. To facilitate efficient lubrication some bearing types, e.g. spherical roller bearings, are provided with an annular groove and/or lubrication holes in the outer or inner ring (➔ **figs** 2 and 5).

Fig 3

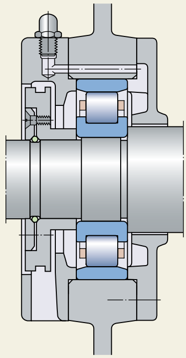


Fig 4

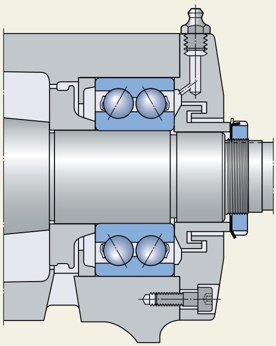
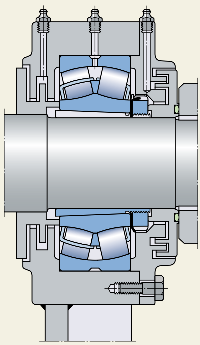


Fig 5



To be effective in replacing old grease, it is important that grease is replenished while the machine is operating. In cases where the machine is not in operation, the bearing should be rotated during replenishment. When lubricating the bearing directly through the inner or outer ring, the fresh grease is most effective in replenishment; therefore, the amount of grease needed is reduced when compared with relubricating from the side. It is assumed that the lubrication ducts were already filled with grease during the mounting process. If not, a greater relubrication quantity during the first replenishment is needed to compensate for the empty ducts.

Where long lubrication ducts are used, check whether the grease can be adequately pumped at the prevailing ambient temperature.

The complete grease fill should be replaced when the free space in the housing can no longer accommodate additional grease, e.g. approximately above 75 % of the housing free volume. When relubricating from the side and starting with 40 % initial fill of the housing, the complete grease fill should be replaced after approximately five replenishments. Due to the lower initial fill of the housing and the reduced topping-up quantity during replenishment in the case of relubricating the bearing directly through inner or outer ring, renewal will only be required in exceptional cases.

Renewing the grease fill

When renewal of the grease fill is made at the estimated relubrication interval or after a certain number of replenishments, the used grease in the bearing arrangement should be completely removed and replaced by fresh grease.

Filling the bearing and housing with grease should be done in accordance with the guidelines given under “Replenishment”.

To enable renewal of the grease fill the bearing housing should be easily accessible and easily opened. The cap of split housings and the covers of one-piece housings can usually be removed to expose the bearing. After removing the used grease, fresh grease should first be packed between the rolling elements. Great care should be taken to see that contaminants are not introduced into the bearing or housing when relubricating, and the grease itself should be protected. The use of grease resistant gloves is recommended to prevent any allergic skin reactions.

When housings are less accessible but are provided with grease nipples and exit holes, it is possible to completely renew the grease fill by relubricating several times in close succession until it can be assumed that all old grease has been pressed out of the housing. This procedure requires much more grease than is needed for manual renewal of the grease fill. In addition, this method of renewal has a limitation with respect to rotational speeds: at high speeds it will lead to undue temperature increases caused by excessive churning of the grease.

Continuous relubrication
This procedure is used when the calculated relubrication interval is very short, e.g. due to the adverse effects of contamination, or when other procedures of relubrication are inconvenient, e.g. access to the bearing is difficult.

Due to the excessive churning of the grease, which can lead to increased temperature, continuous lubrication is only recommended when rotational speeds are low, i.e. at speed factors

- $A < 150\,000$ for ball bearings and
- $A < 75\,000$ for roller bearings.

In these cases the initial grease fill of the housing may be 100 % and the quantity for relubrication per time unit is derived from the equations for G_p under “Replenishment” by spreading the relevant quantity over the relubrication interval.

When using continuous relubrication, check whether the grease can be adequately pumped through the ducts at the prevailing ambient temperature.

Continuous lubrication can be achieved via single-point or multi-point automatic lubricators, e.g. SKF SYSTEM 24 or SYSTEM MultiPoint. For additional information refer to section on “Maintenance and lubrication products”, starting on **page 1065**.

Table 2

SKF greases – technical specification and characteristics

Part 1: Technical specification

Desig-nation	Description	NGLI class	Thickener/ base oil	Base oil viscosity at 40 °C	100 °C	Temperature limits LTL[1]	HTPL[2]
–	–	–	–	mm^2/s		°C	
LGMT 2	All purpose industrial and automotive	2	Lithium soap/ mineral oil	110	11	– 30	+ 120
LGMT 3	All purpose industrial and automotive	3	Lithium soap/ mineral oil	120	12	– 30	+ 120
LGEP 2	Extreme pressure, high load	2	Lithium soap/ mineral oil	200	16	– 20	+ 110
LGLT 2	Low load and temperature, high speed	2	Lithium soap/ diester oil	15	3,7	– 55	+ 100
LGHP 2	High performance and high temperature	2–3	Di-urea/ mineral oil	96	10,5	– 40	+ 150
LGFP 2	Food compatible	2	Aluminium complex/ medical white oil	130	7,3	– 20	+ 110
LGGB 2	Biodegradable and low toxicity	2	Lithium-calcium soap/ ester oil	110	13	– 40	+ 120
LGLC 2	Low temperature and high speed	2	Calcium complex soap/ ester-mineral oil	24	4,7	– 40	+ 120
LGWA 2	Wide temperature range	2	Lithium complex soap/ mineral oil	185	15	– 30 peaks:	+ 140 + 220
LGHB 2	High viscosity and high temperature	2	Calcium complex sulphonate/mineral oil	450	26,5	– 20 peaks:	+ 150 + 200
LGET 2	Extreme temperature	2	PTFE/synthetic (fluorinated polyether)	400	38	– 40	+ 260
LGEM 2	High viscosity with solid lubricants	2	Lithium soap/ mineral oil	500	32	– 20	+ 120
LGEV 2	Extreme high viscosity with solid lubricants	2	Lithium-calcium soap/ mineral oil	1 000	58	–10	+ 120
LGWM 1	Extreme pressure, low temperature	1	Lithium soap/ mineral oil	200	16	– 30	+ 110

1) LTL: low temperature limit
2) HTPL: high temperature performance limit

Table 2

SKF greases – technical specification and characteristics

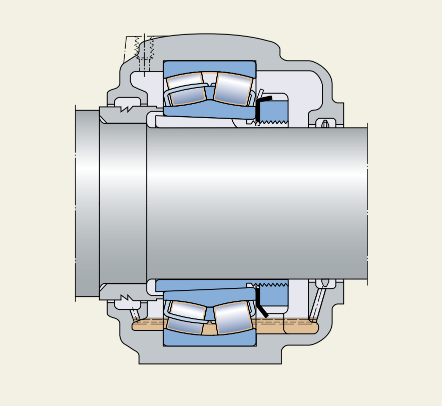
Part 2: Characteristics

Designation	High temperature, above +120 °C	Low temperature	Very high speed	Very low speed or oscillations	Low torque, low friction	Severe vibrations	Heavy loads	Rust inhibiting properties	Water resistance
LGMT 2			o	–	+	+	o	+	+
LGMT 3			o	–	o	+	o	o	+
LGEP 2			o	o	–	+	+	+	+
LGLT 2		+	+	–	+	–	–	o	o
LGHP 2	+	o	+	–	o	–	o	+	+
LGFP 2			o	–	o	o		+	+
LGGB 2		o	o	o	o	+	+	o	+
LGLC 2		+	+	–	+	–	o	+	+
LGWA 2	+		o	o	o	+	+	+	+
LGHB 2	+		o	+	–	+	+	+	+
LGET 2	Contact the SKF application engineering service								
LGEM 2			–	+	–	+	+	+	+
LGEV 2		–	–	+	–	+	+	+	+
LGWM 1		+	o	o	o	–	+	+	+

Symbols: + Recommended
o Suitable
– Not suitable

Where no symbol is indicated the relevant grease may be used – however it is not recommended.
For further information please contact the SKF application engineering service

Fig 6



Oil lubrication

Oil is generally used for rolling bearing lubrication when high speeds or operating temperatures preclude the use of grease, when frictional or applied heat has to be removed from the bearing position, or when adjacent components (gears etc.) are lubricated with oil.

Methods of oil lubrication

Oil bath

The simplest method of oil lubrication is the oil bath (➔ **fig** 6). 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. The oil level should be such that it almost reaches the centre of the lowest rolling element when the bearing is stationary. The use of oil levellers such as the SKF LAHD 500 is recommended to provide the correct oil level. When operating at high speed the oil level can drop significantly and the housing can become overfilled by the oil leveller, under these conditions, please consult the SKF application engineering service.

Oil pick-up ring

For bearing applications where speeds and operating temperature are such that oil lubrication is necessary and high reliability is required the oil pick-up ring lubrication method is recommended (➔ **fig 7**). The pick-up ring serves to bring about oil circulation. The ring hangs loosely on a sleeve on the shaft at one side of the bearing and dips into the oil in the lower half of the housing. As the shaft rotates, the ring follows and transports oil from the bottom to a collecting trough. The oil then flows through the bearing back into the reservoir at the bottom. SKF plummer block housings in the SONL series are designed for the oil pick-up ring lubrication method. For additional information please consult the SKF application engineering service.

Circulating oil

Operation at high speeds will cause the operating temperature to increase and will accelerate ageing of the oil. To avoid frequent oil changes and to achieve a fully flooded condition, the circulating oil lubrication method is generally preferred (➔ **fig 8**). Circulation is usually produced with the aid of a pump. After the oil has passed through the bearing, it generally settles in a tank where it is filtered and, if required, cooled before being returned to the bearing. Proper filtering leads to high values for the factor η_c and thus to long bearing service life (➔ section "SKF rating life", starting on **page 52**).

Cooling the oil enables the operating temperature of the bearing to be kept at a low level.

Fig 7

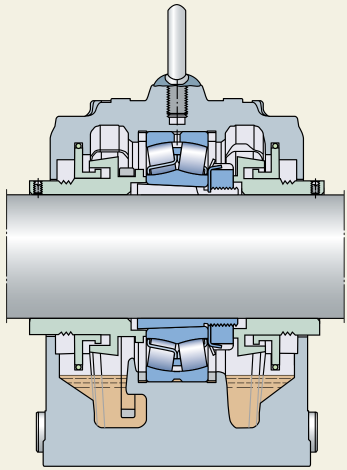


Fig 8

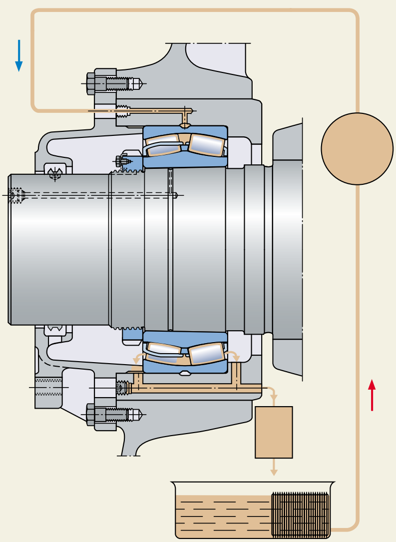


Fig 9

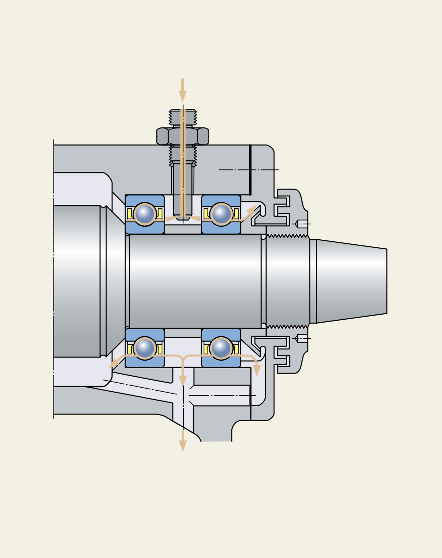
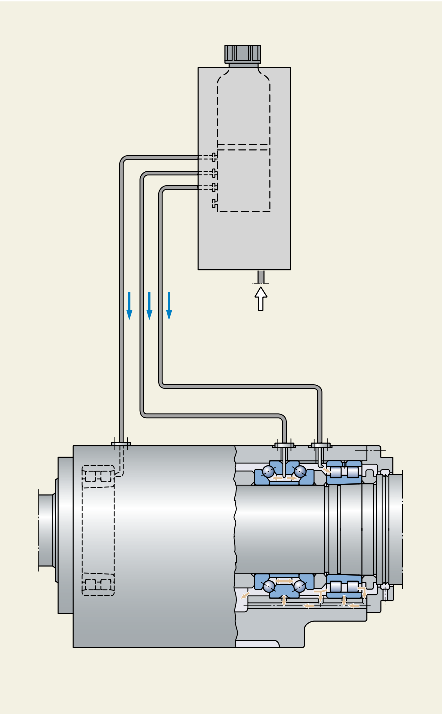


Fig 10



Oil jet

For very high-speed operation a sufficient but not excessive amount of oil must be supplied to the bearing to provide adequate lubrication without increasing the operating temperature more than necessary. One particularly efficient method of achieving this is the oil jet method (➔ **fig** 9) where a jet of oil under high pressure is directed at the side of the bearing. The velocity of the oil jet must be sufficiently high (at least 15 m/s) to penetrate the turbulence surrounding the rotating bearing.

Oil-spot

With the oil-spot method (➔ **fig** 10) – also called the oil-air method – very small, accurately metered quantities of oil are directed at each individual bearing by compressed air. This minimum quantity enables bearings to operate at lower temperatures or at higher speeds than any other method of lubrication. The oil is supplied to the leads by a metering unit, such as the SKF TOS-EX2, at given intervals. The oil is transported by compressed air; it coats the inside of the leads and "creeps" along them. It is projected to the bearing via a nozzle or it just flows to the bearing raceways by a surface tension effect. The compressed air serves to cool the bearing and also produces an excess pressure in the bearing arrangement to prevent contaminants from entering.

When using the circulating oil and oil-spot methods, adequately dimensioned ducts must be provided so that the oil flowing from the bearing can leave the arrangement.

In order to increase bearing service life, all methods of bearing lubrication that use clean oil are preferred, i.e. well filtered circulating oil lubrication, oil jet method and the oil-spot method with filtered air and oil.

Oil mist

Oil mist lubrication has not been recommended for some time due to possible negative environmental effects.

A new generation of oil mist generators permits to produce oil mist with 5 ppm oil. New designs of special seals also limit the amount of stray mist to a minimum. In case synthetic non-toxic oil is used, the environmental effects are even further reduced. Oil mist lubrication today is used in very specific applications, like the petroleum industry.

Lubricating oils

Straight mineral oils without EP additives are generally favoured for rolling bearing lubrication. Oils containing EP, antiwear and other additives for the improvement of certain lubricant properties are generally only used in special cases. The remarks covering EP additives in the section "Load carrying ability: EP and AW additives" on **page 234** also apply to those additives in oils.

Synthetic versions of many of the popular lubricant classes are available. Synthetic oils are generally only considered for bearing lubrication in extreme cases, e.g. at very low or very high operating temperatures. The term synthetic oil covers a wide range of different base stocks. The main ones are polyalphaolefins (PAO), esters and polyalkylene glycols (PAG). These synthetic oils have different properties to mineral oils (➔ **table 3**).

With respect to bearing fatigue life the actual lubricant film thickness plays a major role. The oil viscosity, the viscosity index and the pressure-viscosity coefficient influence the actual film thickness in the contact area for a fully flooded condition. For most mineral oil based lubricants, the pressure-viscosity coefficient is similar and generic values obtained from literature can be used without large error. However, the response of viscosity to increasing pressure is determined by the chemical structure of the base stocks used. As a result of this there is considerable variation in pressure-viscosity coefficients for the different types of synthetic base stocks. Due to the differences in the viscosity index and pressure-viscosity coefficient, it should be remembered that the lubricant film formation, when using synthetic oil, may differ from that of a mineral oil having the same viscosity. Accurate information should always be sought from the individual lubricant supplier.

In addition, additives play a role in the film formation. Due to differences in solubility, different types of additives are applied in synthetic oils when compared with the mineral oil based counterparts.

Table 3

Properties of oil types

Properties	**Base oil type** Mineral	**PAO**	**Ester**	**PAG**
Pour point (°C)	–30 .. 0	–50 .. –40	–60 .. –40	appr. – 30
Viscosity index	low	moderate	low to moderate	high
Pressure-viscosity coefficient	high	moderate	low	high

Selection of lubricating oils

Selecting oil is primarily based on the viscosity required to provide adequate lubrication for the bearing at normal operating temperature. The viscosity of oil is temperature dependent, becoming lower as the temperature rises. The viscosity-temperature relationship of oil is characterized by the viscosity index VI. For rolling bearing lubrication, oils having a high viscosity index (little change with temperature) of at least 85 are recommended.

In order for a sufficiently thick oil film to be formed in the contact area between rolling elements and raceways, the oil must retain a minimum viscosity at the operating temperature. The rated kinematic viscosity ν_1 required at the operating temperature to provide adequate lubrication can be determined from **diagram 5**, **page 254**, provided a mineral oil is used. When the operating temperature is known from experience or can otherwise be determined, the corresponding viscosity at the internationally standardized reference temperature of 40 °C can be obtained from **diagram 6**, **page 255**, which is compiled for a viscosity index of 95. Certain bearing types, e.g. spherical roller bearings, toroidal roller bearings, taper roller bearings, and spherical roller thrust bearings, normally have a higher operating temperature than other bearing types, e.g. deep groove ball bearings and cylindrical roller bearings, under comparable operating conditions.

When selecting the oil the following aspects should be considered.

- Bearing life may be extended by selecting an oil where the actual operating viscosity (ν) is somewhat higher than the rated viscosity (ν_1). However, since increased viscosity raises the bearing operating temperature there is frequently a practical limit to the lubrication improvement that can be obtained by this means.
- If the viscosity ratio $\kappa = \nu/\nu_1$ is less than 1, an oil containing EP additives is recommended and if κ is less than 0,4 an oil with EP additives must be used. An oil with EP additives may also enhance operational reliability in cases where κ is greater than 1 and medium and large-size roller bearings are concerned. It should be remembered that some EP additives may have adverse effects (➔ "Load carrying ability: EP and AW additives " on **page 234**).
- For exceptionally low or high speeds, for critical loading conditions, or for unusual lubricating conditions please consult the SKF application engineering service.

Example

A bearing having a bore diameter d = 340 mm and outside diameter D = 420 mm is required to operate at a speed n = 500 r/min. Therefore $d_m = 0{,}5\,(d + D) = 380$ mm. From **diagram 5,** the minimum kinematic viscosity ν_1 required for adequate lubrication at the operating temperature is approximately 11 mm²/s. From **diagram 6**, assuming that the operating temperature of the bearing is 70 °C, it is found that a lubricating oil to an ISO VG 32 viscosity class, with an actual viscosity ν of at least 32 mm²/s at the reference temperature of 40 °C will be required.

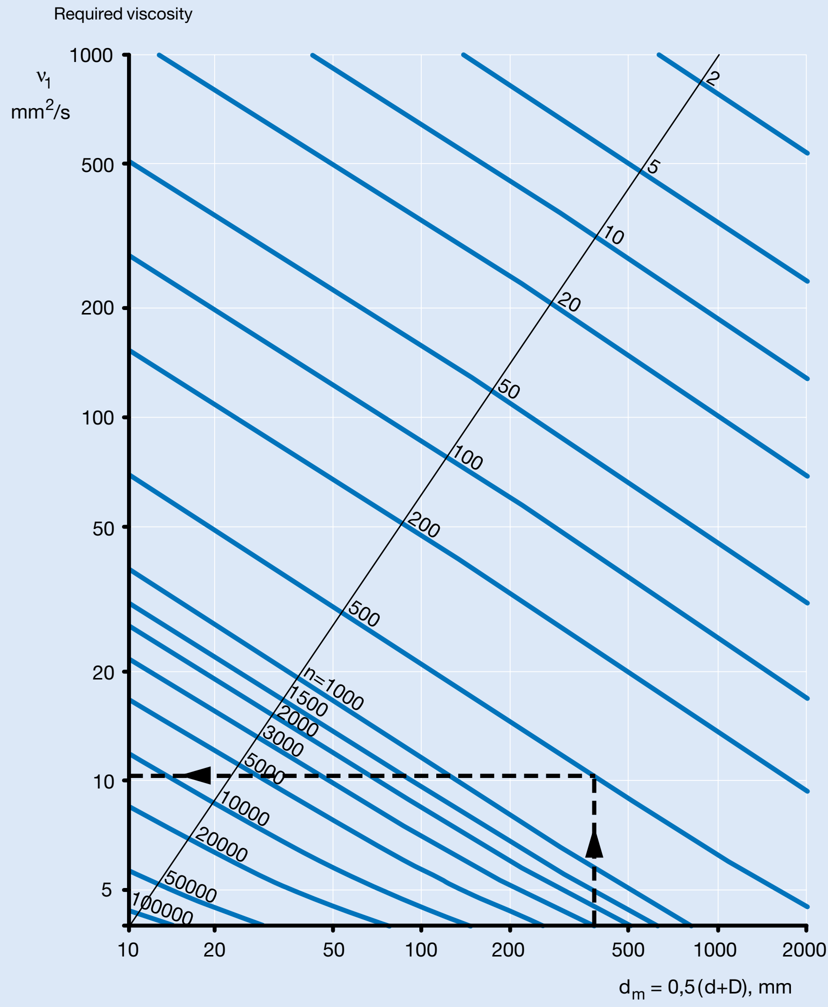
Oil change
The frequency with which it is necessary to change the oil depends mainly on the operating conditions and the quantity of oil.

With oil bath lubrication it is generally sufficient to change the oil once a year, provided the operating temperature does not exceed 50 °C and there is little risk of contamination. Higher temperatures call for more frequent oil changes, e.g. for operating temperatures around 100 °C, the oil should be changed every three months. Frequent oil changes are also needed if other operating conditions are arduous.

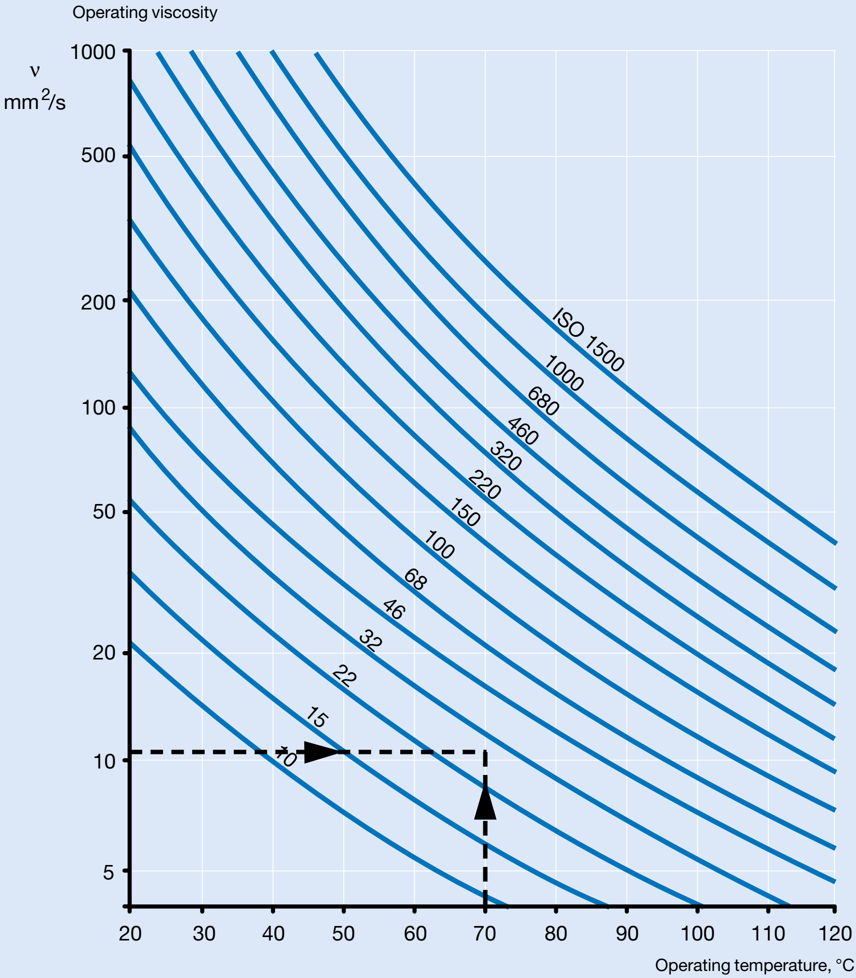
With circulating oil lubrication, the period between two oil changes is also determined by how frequently the total oil quantity is circulated and whether or not the oil is cooled. It is generally only possible to determine a suitable interval by test runs and by regular inspection of the condition of the oil to see that it is not contaminated and is not excessively oxidized. The same applies for oil jet lubrication. With oil spot lubrication the oil only passes through the bearing once and is not recirculated.

Diagram 5

Estimation of the rated viscosity ν_1 required for adequate lubrication



Estimation of the actual operating viscosity ν




SKF

Mounting and dismounting

General information

To provide proper bearing performance and prevent premature failure, skill and cleanliness when mounting ball and roller bearings are necessary.

As precision components, rolling bearings should be handled carefully when mounting. It is also important to choose the correct method of mounting and to use the correct tool for the job. The comprehensive SKF range of maintenance products includes mechanical and hydraulic tools and heating equipment as well as other products for mounting and maintenance. This full line of products will facilitate and speed the work, giving professional results. Brief information can be found in the section "Maintenance and lubrication products", starting on **page 1065**.

To realize maximum bearing service life, a bearing must be installed correctly – which often is more difficult than it appears, especially where large size bearings are concerned. To be sure that bearings are mounted and maintained properly, SKF offers seminars and hands-on training courses as part of the SKF Reliability Systems concept. Installation and maintenance assistance may also be available from your local SKF company.

The information provided in the following section is quite general and is intended primarily to indicate what must be considered by machine and equipment designers in order to facilitate bearing mounting and dismounting. More detailed descriptions of the actual mounting and dismounting procedures will be found in the publication "SKF Bearing Maintenance Handbook" which is available through your local SKF representative on request, or online at www.skf.com/mount or www.aptitudexchange.com.

Where to mount

Bearings should be installed in a dry, dust-free room away from metalworking or other machines producing swarf and dust. When bearings have to be mounted in an unprotected area, which is often the case with large bearings, steps need to be taken to protect the bearing and mounting position from contamination by dust, dirt and moisture until installation has been completed. This can be done by covering or wrapping bearings, machine components etc. with waxed paper or foil.

Preparations for mounting and dismounting

Before mounting, all the necessary parts, tools, equipment and data need to be at hand. It is also recommended that any drawings or instructions be studied to determine the correct order in which to assemble the various components.

Housings, shafts, seals and other components of the bearing arrangement need to be checked to see that they are clean, particularly any threaded holes, leads or grooves where remnants of previous machining operations might have collected. The unmachined surfaces of cast housings need to be free of core sand and any burrs need to be removed.

The dimensional and form accuracy of all components of the bearing arrangement needs to be checked. The bearings will only perform satisfactorily if the associated components have the requisite accuracy and if the prescribed tolerances are adhered to. The diameter of cylindrical shaft and housing seatings are usually checked using a stirrup or internal gauge at two cross-sections and in four directions (➔ **fig 1**). Tapered bearing seatings are checked using ring gauges, special taper gauges or sine bars.

It is advisable to keep a record of the measurements. When measuring it is important that the components being measured and the measuring instruments have approximately the same temperature. This means that it is necessary to leave the components and measuring equipment together in the same place sufficiently long for them to reach the same temperature. This is particularly important where large bearings and their associated components, which are correspondingly large and heavy, are concerned.

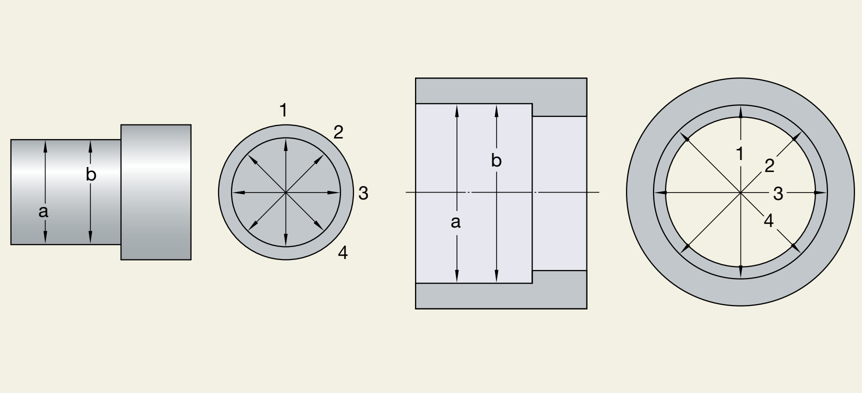
The bearings need to be left in their original packages until immediately before mounting so that they will not be exposed to any contaminants, especially dirt. Normally, the preservative with which new bearings are coated before leaving the factory does not need to be removed; it is only necessary to wipe off the outside cylindrical surface and bore. If, however, the bearing is to be grease lubricated and used at very high or very low temperatures, or if the grease is not compatible with the preservative, it is necessary to wash and carefully dry the bearing. This is to avoid any detrimental effect on the lubricating properties of the grease.

Bearings should be washed and dried before mounting if there is a risk that they have become contaminated because of improper handling (damaged packaging etc.).

When taken from its original packaging, any bearing that is covered by a relatively thick, greasy layer of preservative should also be washed and dried. This might be the case for some large-size bearings with an outside diameter larger than 420 mm. Suitable agents for washing rolling bearings include white spirit and paraffin.

Bearings that are supplied ready greased and which have integral seals or shields on both sides should not be washed before mounting.

Fig 1



Bearing handling

It is generally a good idea to use gloves as well as carrying and lifting tools, which have been specially designed for mounting and dismounting bearings. This will save not only time and money but the work will also be less tiring, less risky and less injurious to health.

For these reasons, the use of heat and oil resistant gloves is recommended when handling hot or oily bearings. These gloves should have a durable outside and a soft non-allergenic inside, as, for example, SKF TMBA gloves.

Heated and/or larger or heavier bearings often cause problems because they cannot be handled in a safe and efficient manner by one or two persons. Satisfactory arrangements for carrying and lifting these bearings can be made on site in a workshop. The bearing handling tool TMMH from SKF (➔ **fig** 2) is one such arrangement, which solves most of the problems and facilitates handling, mounting and dismounting bearings on shafts.

If large, heavy bearings are to be moved or held in position using lifting tackle, they should not be suspended at a single point, but a steel band or fabric belt should be used (➔ **fig** 3). A spring between the hook of the lifting tackle and the belt facilitates positioning the bearing when it is to be pushed onto a shaft.

To ease lifting, large bearings can be provided on request with threaded holes in the ring side faces to accommodate eye bolts. The hole size is limited by the ring thickness. It is therefore only permissible to lift the bearing itself or the individual ring by the bolts. Care also needs be taken to see that the eye bolts are only subjected to load in the direction of the shank axis (➔ **fig** 4). If the load is to be applied at an angle, suitable adjustable attachments are required.

When mounting a large housing over a bearing that is already in position on a shaft it is advisable to provide three-point suspension for the housing, and for the length of one sling to be adjustable. This enables the housing bore to be exactly aligned with the bearing.

Fig 2

Fig 3

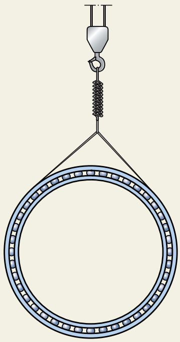


Fig 4

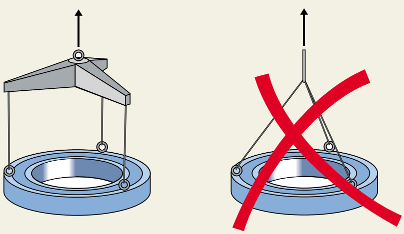


Fig 5

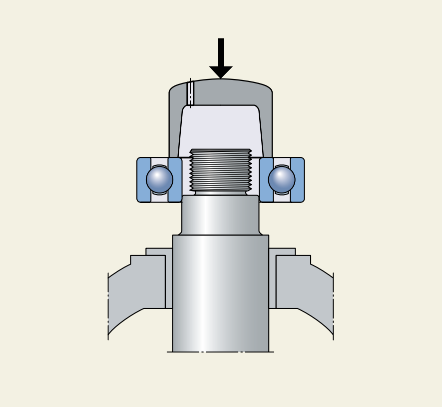


Fig 6

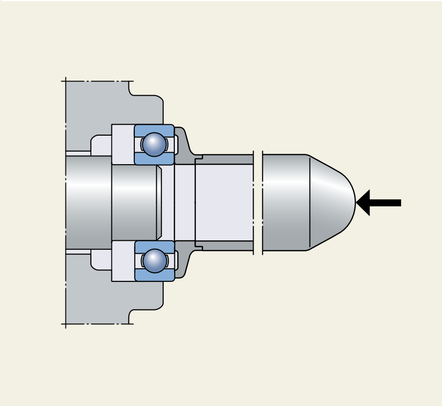
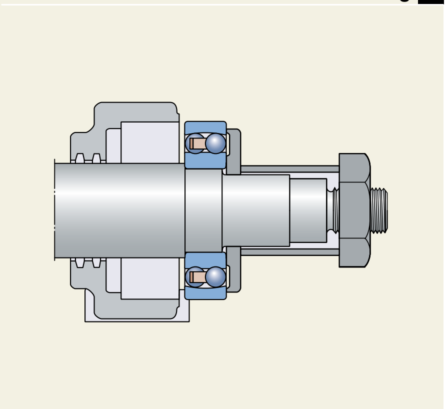


Fig 7



Mounting

Depending on the bearing type and size, mechanical, thermal or hydraulic methods are used for mounting. In all cases it is important that the bearing rings, cages and rolling elements or seals do not receive direct blows and that the mounting force must never be directed through the rolling elements.

Some parts may be mounted with a loose fit. To avoid any fretting corrosion between the mating surfaces, it is recommended to apply a thin layer of SKF anti-fretting agent LGAF 3 E.

Mounting bearings with a cylindrical bore
With non-separable bearings, the ring that is to have the tighter fit should generally be mounted first. The seating surface should be lightly oiled before mounting.

Cold mounting

If the fit is not too tight, small bearings may be driven into position by applying light hammer blows to a sleeve placed against the bearing ring face. The blows should be evenly distributed around the ring to prevent the bearing from tilting or skewing. The use of a mounting dolly instead of a sleeve allows the mounting force to be applied centrally (➔ **fig** 5).

If a non-separable bearing is to be pressed onto the shaft and into the housing bore at the same time the mounting force has to be applied equally to both rings and the abutment surfaces of the mounting tool must lie in the same plane. In this case a bearing fitting tool should be used, where an impact ring abuts the side faces of the inner and outer rings and the sleeve allows the mounting forces to be applied centrally (➔ **fig** 6).

With self-aligning bearings, the use of an intermediate mounting ring prevents the outer ring from tilting and swivelling when the bearing with shaft is introduced into the housing bore (➔ **fig** 7). It should be remembered that the balls of some sizes of self-aligning ball bearings protrude from the side faces of the bearing, so that the intermediate mounting ring should be recessed in order not to damage the balls. Large numbers of bearings are generally mounted using mechanical or hydraulic presses.

With separable bearings, the inner ring can be mounted independently of the outer ring, which simplifies mounting, particularly where both rings are to have an interference fit. When installing the shaft with the inner ring already in position, into the housing containing the outer ring, care must be taken to see that they are correctly aligned to avoid scoring the raceways and rolling elements. When mounting cylindrical and needle roller bearings with an inner ring without flanges or a flange at one side, SKF recommends using a mounting sleeve (➔ **fig** 8). The outside diameter of the sleeve should be equal to the raceway diameter F of the inner ring and should be machined to a d10 tolerance.

Hot mounting

It is generally not possible to mount larger bearings in the cold state, as the force required to mount a bearing increases very considerably with increasing bearing size. The bearings, the inner rings or the housings (e.g. hubs) are therefore heated prior to mounting.

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 seating. Bearings should not be heated to more than 125 °C as otherwise dimensional changes caused by alterations in the structure of the bearing material may occur. Bearings fitted with shields or seals should not be heated above 80 °C because of their grease fill or seal material.

When heating bearings local overheating must be avoided. To heat bearings evenly, SKF electric induction heaters (➔ **fig** 9) are recommended. If hotplates are used, the bearing must be turned over a number of times. Hotplates should not be used for heating sealed bearings

Bearing adjustment
The internal clearance of single row angular contact ball bearings and taper roller bearings is only established, in contrast to other radial bearings with cylindrical bore, when one bearing is adjusted against a second bearing. Usually these bearings are arranged in pairs either back-to-back or face-to-face, and one bearing ring is axially displaced until a given clearance or preload is attained. The choice of clearance or preload depends on the demands placed on the performance of the bearing arrangement and on the operating conditions. Additional information about bearing preloads can be found under the heading "Bearing preload" so that the recommendations in the following refer only to the

Fig 8

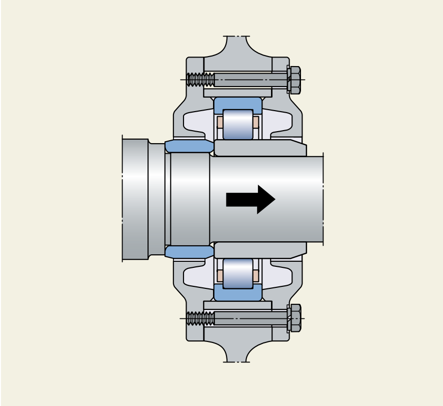


Fig 9

adjustment of internal clearance in bearing arrangements with angular contact ball bearings and taper roller bearings.

The appropriate value for the clearance to be obtained when mounting is determined by the conditions when the bearing is under load and at the operating temperature. Depending on the size and arrangement of the bearings, the materials from which the shaft and housing are made and the distance between the two bearings, the initial clearance obtained on mounting may be smaller or larger in actual operation. If, for example, differential thermal expansion of inner and outer rings will cause a reduction in clearance during operation, the initial clearance must be sufficiently large so that distortion of the bearings and the detrimental consequences of this are avoided.

Since there is a definite relationship between the radial and axial internal clearance of angular contact ball bearings and taper roller bearings, it is sufficient to specify one value, generally the axial internal clearance. This specified value is then obtained, from a condition of zero clearance, 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 methods used to adjust the clearance and measure the set clearance are determined by whether a few or many bearings are to be mounted.

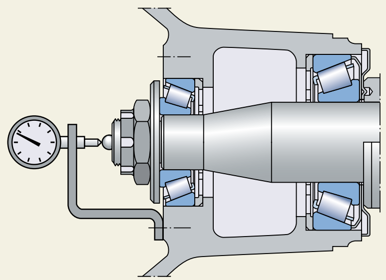
One method is to check the set axial clearance, for example, of a hub bearing arrangement, using a dial gauge attached to the hub (➔ **fig 10**). It is important when adjusting taper roller bearings and measuring the clearance that the shaft, or housing, is turned through several revolutions in both directions to be sure that there is proper contact of the roller ends with the guide flange on the inner ring. If the contact is not correct, the measured result will be inaccurate and the desired adjustment will not be achieved.

Mounting bearings with a tapered bore

For bearings having a tapered bore, inner rings are always mounted with an interference fit. The degree of interference is not determined by the chosen shaft tolerance, as with bearings having a cylindrical bore, but by how far the bearing is driven up onto the tapered shaft seating, or onto the adapter or withdrawal sleeve. As the bearing is driven up the tapered seating, its radial internal clearance is reduced. This reduction can be measured to determine the degree of interference and the proper fit.

When mounting self-aligning ball bearings, CARB toroidal roller bearings, spherical roller bearings, as well as high-precision cylindrical roller bearings with tapered bore either the reduction in radial internal clearance or the axial drive-up onto the tapered seating is determined and used as a measure of the degree of interference. Guideline values of clearance reduction and axial drive-up are given in the text preceding the relevant product table sections.

Fig 10



Small bearings

Small bearings may be driven up onto a tapered seating using a nut. In the case of adapter sleeves the sleeve nut is used. Small withdrawal sleeves may be driven into the bearing bore using a nut. A hook or impact spanner can be used to tighten the nut. The seating surfaces of the shaft and sleeve should be lightly oiled with thin oil before mounting is started.

Medium and large sized bearings

For larger bearings, considerably more force is required and

- SKF hydraulic nuts should be used and/or
- the oil injection method should be employed.

In either case, the mounting process will be considerably easier. The oil injection equipment required for both, operating the hydraulic nut as well as for applying the oil injection method, is available from SKF. Additional information about these products can be found in the section "Maintenance and lubrication products", starting on **page 1065.**

When using an SKF hydraulic nut for mounting it has to be positioned onto a threaded section of the journal or onto the thread of the sleeve so that its annular piston abuts the inner ring of the bearing, a nut on the shaft, or a disc attached to the end of the shaft. Pumping oil into the hydraulic nut displaces the piston axially with the force needed for accurate and safe mounting. Mounting of a spherical roller bearing with the aid of a hydraulic nut on

- a tapered seating is shown in **fig** 11,
- an adapter sleeve is shown in **fig** 12,
- a withdrawal sleeve is shown in **fig** 13.

With the oil injection method, oil under high pressure is injected between the bearing and bearing seating to form an oil film. This oil film separates the mating surfaces and appreciably reduces the friction between them. This method is typically used when mounting bearings directly on tapered journals (➔ **fig** 14), but is also used to mount

Fig 11

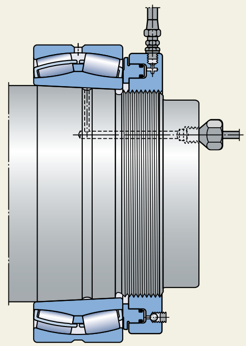


Fig 12

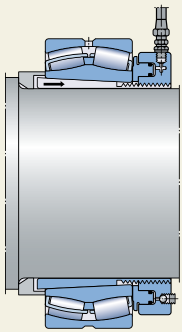


Fig 13

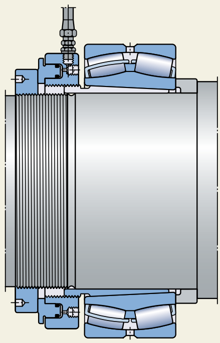
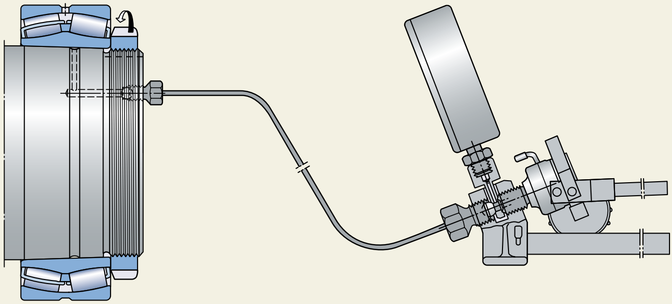
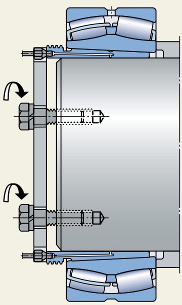


Fig 14



bearings on adapter and withdrawal sleeves that have been prepared for the oil injection method. A pump or oil injector produces the requisite pressure, the oil is injected between the mating surfaces via ducts and distributor grooves in the shaft or sleeve. The necessary ducts and grooves in the shaft must be considered when designing the bearing arrangement. A spherical roller bearing mounted on a withdrawal sleeve with oil ducts is shown in **fig** 15. The withdrawal sleeve is pressed into the bearing bore by injecting oil between the mating surfaces and tightening the screws in turn.

Fig 15



Determination of the interference fit

Bearings with a tapered bore are always mounted with an interference fit. The reduction in radial internal clearance, or the axial displacement of the inner ring on its tapered seating is used to determine and measure the degree of interference.

Different methods can be used to measure the degree of interference.

1. Measuring the clearance reduction with a feeler gauge.
2. Measuring lock nut tightening angle.
3. Measuring the axial drive-up.
4. Measuring the inner ring expansion.

A brief description of these four different methods is provided in the following. More detailed information about these methods will be found in the relevant product sections.

Measuring clearance reduction with a feeler gauge

The method using feeler gauges for measuring the radial internal clearance before and after mounting bearings is applicable for medium and large-sized spherical and toroidal roller bearings. The clearance should preferably be measured between the outer ring and an unloaded roller (➔ **fig** 16).

Measuring the lock nut tightening angle

Measuring the lock nut tightening angle is a proven method to determine the correct degree of interference in small to medium-sized bearings on tapered seatings (➔ **fig** 17). Guideline values for the tightening angle α have been established, providing accurate positioning of the bearing on its tapered seating.

Measuring the axial drive-up

Mounting bearings with a tapered bore can be done by measuring the axial drive-up s of the inner ring on its seating. Guideline values for the required axial drive-up are given in the text preceding the relevant product table sections.

However, a more suitable method is the “SKF Drive-up Method”. This mounting 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 predetermined position. The method incorporates the use of an SKF hydraulic nut fitted with a dial indicator, and a specially calibrated digital gauge mounted on a selected pump (➔ **fig** 18). Determined values of the requisite oil pressure and the axial displacement for the individual bearings provide accurate positioning of the bearings. These values can be found

- in the handbook “SKF Drive-up Method” on CD-ROM,
- in the “SKF Interactive Engineering Catalogue” on CD-ROM or online at www.skf.com or
- online at www.skf.com/mount.

Fig 16

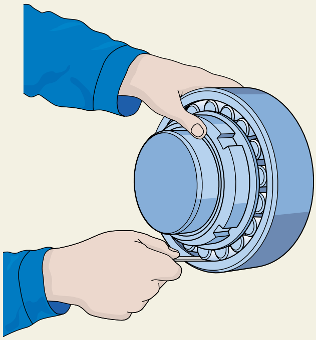


Fig 17

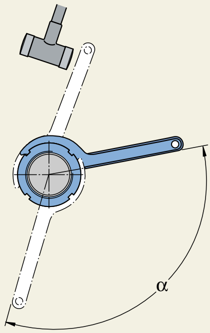
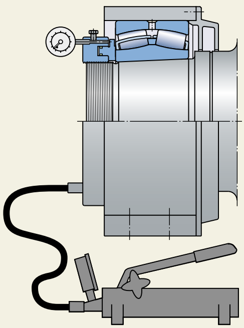


Fig 18



Measuring the inner ring expansion

Measuring inner ring expansion is a simple and very accurate method to determine the correct position of large-size spherical and toroidal roller bearings on their seatings. For this kind of measurement the SKF SensorMount® is now available, using a sensor, integrated with the bearing inner ring, a dedicated hand-held indicator and common hydraulic mounting tools (➔ **fig 19**). Aspects such as bearing size, shaft smoothness, material or design – solid or hollow – do not need to be considered

Test running

After mounting a bearing, the prescribed lubricant is applied and a test run made so that noise and bearing temperature can be checked.

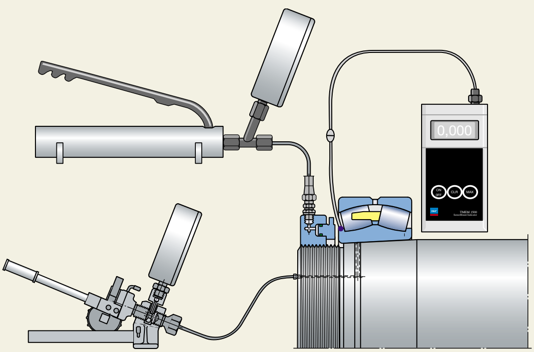
The test run should be carried out under partial load and – where there is a wide speed range – at slow or moderate speed. Under no circumstances should a rolling bearing be allowed to start up unloaded and accelerated to high speed, as there is a danger that the rolling elements would slide on the raceways and damage them, or that the cage would be subjected to inadmissible stresses. Reference should be made to the section "Minimum load" in the text preceding the relevant product table sections.

Any noise or vibration can be checked using an SKF electronic stethoscope. Normally, bearings produce an even "purring" noise. Whistling or screeching indicates inadequate lubrication. An uneven rumbling or hammering is due in most cases to 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. For example, in the case of grease lubrication, the temperature will not drop until the grease has been evenly distributed in the bearing arrangement, after which an equilibrium temperature will be reached. Unusually high temperatures or constant peaking indicates that there is too much lubricant in the arrangement or that the bearing is radially or axially distorted. Other causes are that the associated components have not been correctly made or mounted, or that the seals have excessive friction.

During the test run, or immediately afterwards, the seals should be checked to see that they perform correctly and any lubrication equipment as well as the oil level of an oil bath should be checked. It may be necessary to sample the lubricant to determine whether the bearing arrangement is contaminated or components of the arrangement have become worn.

Fig 19



Dismounting

If bearings are to be used again after removal, 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 seating first. To dismount a bearing having an interference fit, the tools described in the following section may be used, the choice of tools will depend on bearing type, size and fit.

Dismounting bearings with a cylindrical bore

Cold dismounting

Small bearings may be removed from their seatings by applying light hammer blows via a suitable drift to the ring face, or preferably by using a puller. The claws of the puller should be placed around the side face of the ring to be removed, or an adjacent component (➔ **fig** 20), e.g. a labyrinth ring etc. Dismounting is made easier if

- provision is made for slots in the shaft and housing shoulders to take the claws of the puller or
- tapped holes are provided in the housing shoulders to take withdrawal screws (➔ **fig** 21).

Larger bearings mounted with an interference fit generally require greater force to remove them, particularly if, after a long period of service, fretting corrosion has occurred. Use of the oil injection method considerably facilitates dismounting in such cases. This presupposes that the necessary oil supply ducts and distributor grooves have been designed into the arrangement (➔ **fig** 22).

Fig 20

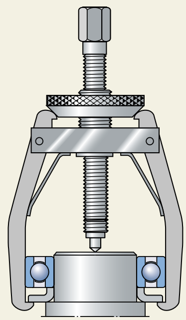


Fig 21

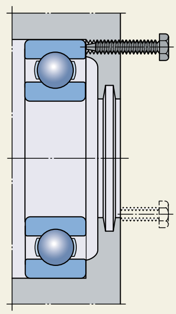


Fig 22

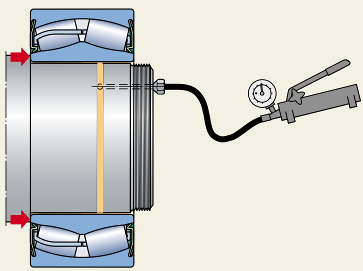


Fig 23

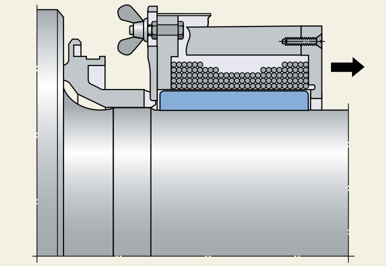
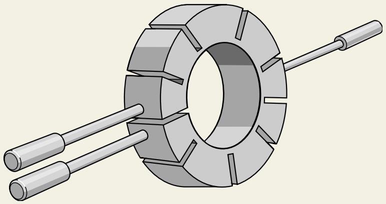


Fig 24



Hot dismounting

Special induction heaters have been developed to dismount the inner rings of cylindrical roller bearings having no flanges or only one flange. They heat the inner ring rapidly without heating the shaft to any degree, so that the expanded ring can easily be removed. These electrical induction heaters (➔ **fig** 23) have one or more coils energized by alternating current. It is necessary to demagnetize the inner rings after heating and removal. The use of electric withdrawal tools becomes economic when bearings of the same size are frequently mounted and dismounted.

When flangeless inner rings of cylindrical roller bearings, or those with only one flange, which have not to be removed frequently, or if larger sizes of inner rings (up to approximately 400 mm bore diameter) have to be dismounted it is less costly and also easier to use a so-called thermo-withdrawal ring also referred to as a heating ring. This is a slotted ring, generally of light alloy, with handles (➔ **fig** 24). The above-mentioned heaters and heating rings are available from SKF. Additional information can be found in the section “Maintenance and lubrication products”, starting on **page 1065.**

Dismounting bearings with a tapered bore

Dismounting bearing on a tapered journal

Small and medium-sized bearings on a tapered journal can be dismounted using conventional pullers, which engage the inner ring (➔ **fig** 25). Preferably a self-centring puller should be used to avoid damage to the bearing seating. Bearings on tapered seatings normally loosen very quickly. Therefore, it is necessary to provide a stop of some kind, a lock nut for example, to prevent the bearing from being completely withdrawn from the shaft.

The dismounting of large bearings from tapered journals is greatly eased if the oil injection method is employed. After injecting pressurised oil between the mating surfaces, the bearing will separate suddenly from its seating. A stop must therefore be provided, for example, a shaft nut or end plate, to limit the axial movement of the bearing (➔ **fig** 26).

Fig 25

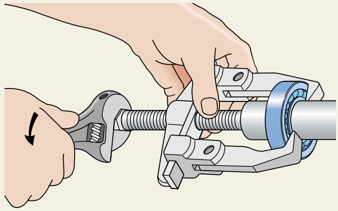


Fig 26

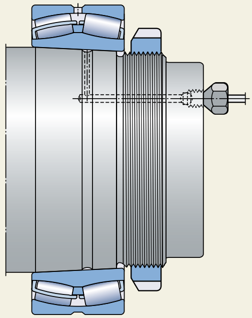


Fig 27

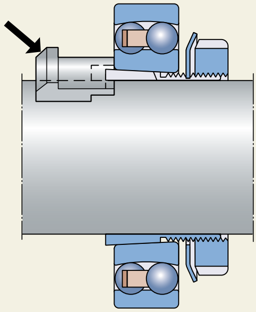
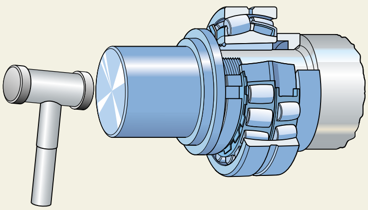


Fig 28



Dismounting bearing on an adapter sleeve

Small and medium-sized bearings on adapter sleeve and smooth shafts can be dismounted by hammer blows directed to a drift (➔ **fig** 27) until the bearing becomes free. But first the sleeve nut has to be loosened a few turns.

Small and medium-sized bearings on adapter sleeve and stepped shafts can be dismounted by using a dolly abutting the sleeve nut, which has been released by a few turns (➔ **fig** 28).

Dismounting large bearings from an adapter sleeve with a hydraulic nut has proved easy to do. To use this technique however, the bearing must be mounted against a shoulder (➔ **fig** 29). If the sleeves are provided with oil supply ducts and distributor grooves the dismounting becomes easier because the oil injection method can be employed.

Dismounting bearing on a withdrawal sleeve

When dismounting bearings on withdrawal sleeves, the axial locking device: a locking nut, end cover etc., has to be removed. Small and medium-sized bearings can be dismounted using a lock nut and a hook or impact spanner to free the bearing (➔ **fig** 30).

Fig 29

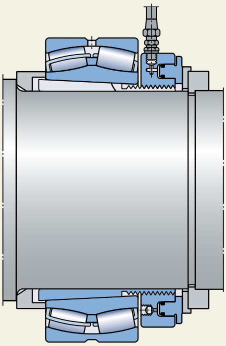
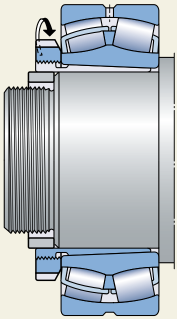


Fig 30



The preferred means of dismounting large bearings is by using a hydraulic nut (➔ **fig** 31). If the threaded section of the sleeve protrudes beyond the shaft end or shaft shoulder, a support ring having the greatest possible wall thickness should be inserted in the sleeve bore to prevent distortion and damage to the thread when the nut is tightened.

Withdrawal sleeves for large bearings are generally provided with distributor ducts and grooves for the oil injection method to save considerable time when mounting as well as dismounting large bearings (➔ **fig** 32).

Fig 31

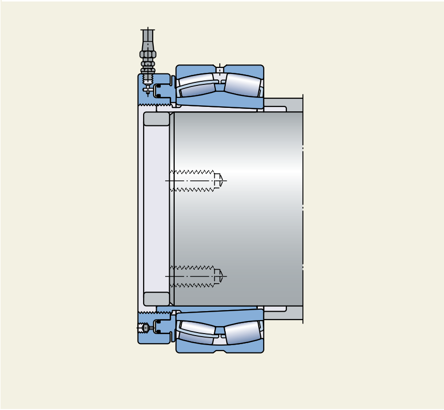
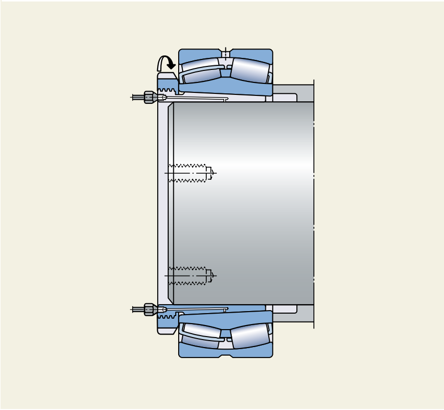


Fig 32



Bearing storage

Bearings can be stored in their original packaging for many years, provided that the relative humidity in the storeroom does not exceed 60 % and there are no great fluctuations in temperature. With sealed or shielded bearings it may be found that the lubricating properties of the grease with which they are filled may have deteriorated if the bearings have been stored for a long time. Bearings that are not stored in their original packaging should be well protected against corrosion and contamination.

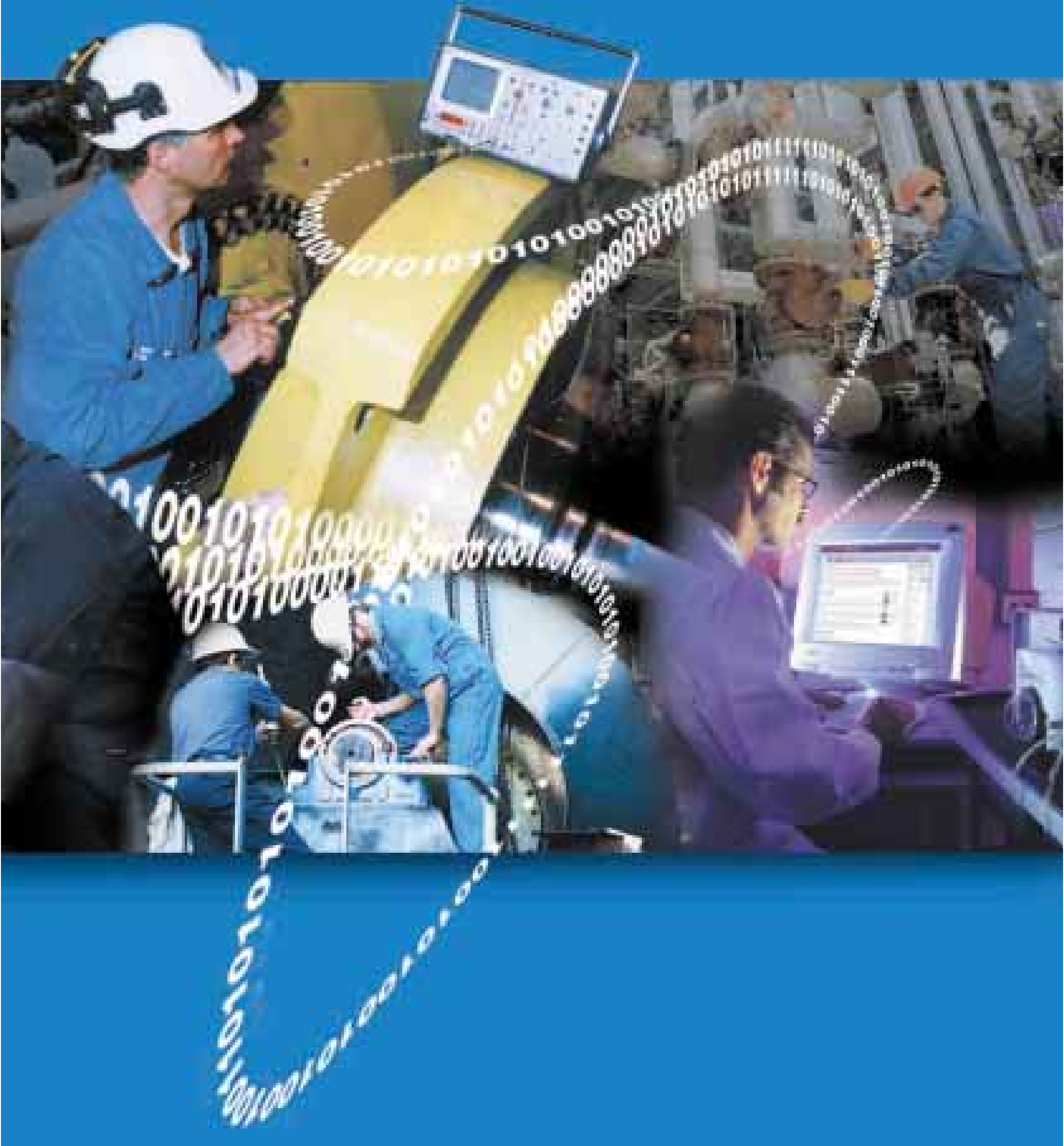
Large rolling bearings should only be stored lying down, and preferably with support for the whole extent of the side faces of the rings. If kept in a standing position, the weight of the rings and rolling elements can give rise to permanent deformation because the rings are relatively thin-walled.

Inspection and cleaning

As with all other important machine components, ball and roller bearings must be frequently cleaned and examined. The intervals between such examinations depend entirely on the operating conditions.

If it is possible to ascertain the condition of the bearing during service, e.g. by listening to the sound of the bearing when it is running and measuring the temperature or examining the lubricant, then it is usually found sufficient if the bearings (rings, cage and rolling elements) and other parts of the bearing arrangement are thoroughly cleaned and inspected annually. Where the load is heavy, the frequency of inspection must be increased, e.g. rolling mill bearings are often inspected when the rolls are changed.

After the bearing components have been cleaned with a suitable solvent (white spirit, paraffin etc.) they should be oiled or greased immediately to prevent corrosion. This is particularly important for bearings in machines that are left to stand for considerable periods.



Reliability and services

SKF has been a leader and innovator in bearing technology since 1907. The evolution of SKF expertize in machine reliability stems from the very nature of bearings and their applications. SKF's understanding of a bearing's performance in an application requires an equally extensive knowledge of the machines and the processes. The thorough understanding of machine components, systems and related processes, enables SKF to create and provide realistic solutions for optimum machine and process reliability and productivity.

Close working partnerships with customers worldwide has provided SKF with an extensive knowledge of applications in virtually every industry. As a result, SKF has learned to apply the most relevant of today's emerging technologies to industry-specific applications.

Through SKF Reliability Systems SKF provides a single source for a complete productivity solution. The goal is to help customers reduce total machine related costs, enhance productivity and strengthen profitability. Whatever the requirements SKF Reliability Systems offers the knowledge, services and products needed to achieve specific business goals.

An integrated platform

SKF's range of products and services provides the solutions that will ultimately lead to increased bottom line profitability. The focus on technology and seamless interface with plantwide systems supports four key areas.

Decision support

SKF can assist customers in retention, storage and utilization of crucial information with its @ptitude industrial decision support software (➔ **page 279**).

Condition monitoring

As a leading supplier of condition monitoring products, SKF offers a complete range – from hand-held data collectors/analysers to online surveillance and machine protection systems. These products provide interface with condition monitoring analysis software and other plantwide systems and are listed starting on **page 280**.

Tools and lubricants

SKF has developed a range of tools and lubricants to provide safe and damage-free machine maintenance. Brief information on these products is given in the section "Maintenance and lubrication products", starting on **page 1065**.

Component innovations

Component innovations are needed to achieve productivity goals that were never intended by original equipment manufacturers. SKF has developed bearing products designed to run faster, longer and cooler without maintenance in many difficult applications. A selection of such products is listed in the sections "Engineering products", starting on **page 889,** and "Mechatronics", starting on **page 951**.

The Asset Efficiency Optimization concept

The Asset Efficiency Optimization™ (AEO) concept from SKF picks up where most plan asset management programmes typically stop. Using this concept enables a plant to produce the same amount for less cost, or to produce more for the same costs. It is a system for organizing and applying assets – from personnel to machinery – bringing together knowledge and technology to achieve the greatest return on investment.

By applying the power of SKF's technology and service solutions, you can benefit from a programme that assists in achieving your organization's overall business objectives. These include reduced costs, greater productivity, better utilization of resources, and as a result, increased bottom line profitability (➔ **diagram** 1).

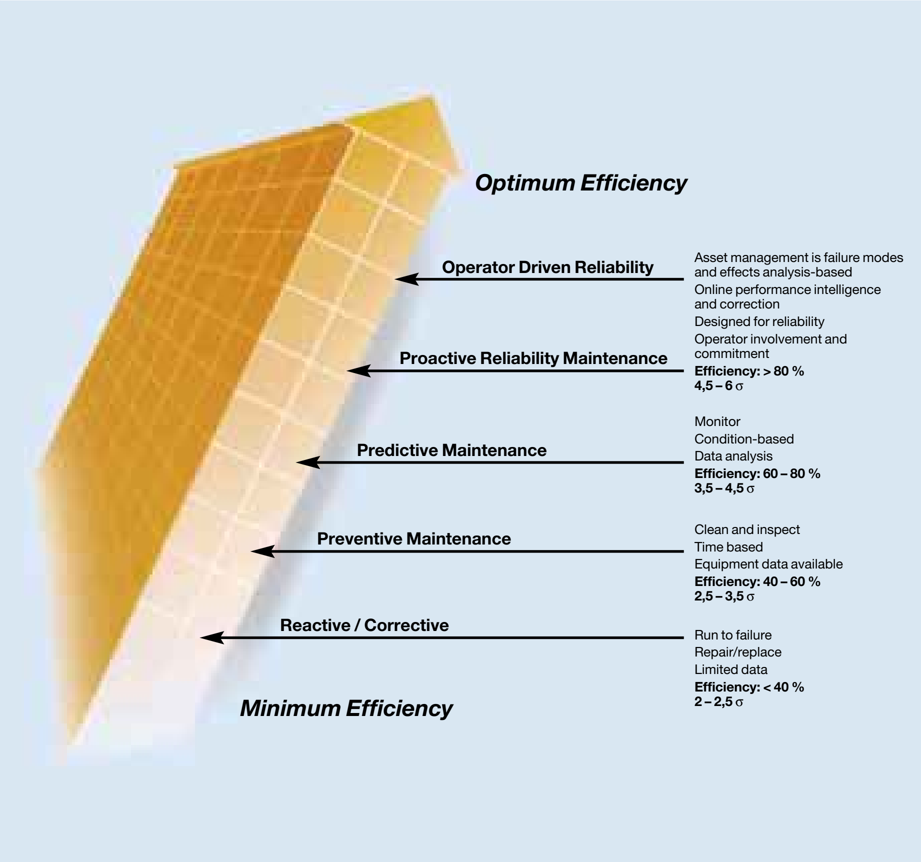
SKF technology and service solutions

The following summarizes the most important services and products that SKF Reliability Systems offers to provide solutions to the real-life application conditions. For detailed information on the SKF Reliability Systems program please refer to publication 5160 E "The Guide to Asset Efficiency Optimization™ for Improved Profitability" or visit www.skfreliability.com to see the latest information on strategies and services.

Assessment
An assessment can include one or all of the following areas.

- Determination of current situation
- Maintenance
- Supply and stores processes
- Predictive maintenance

Diagram 1



Maintenance strategy
SKF can help to establish a comprehensive maintenance strategy, designed to make sure that productivity as well as safety and integrity issues receive the attention they require. **Diagram 1** on **page 277** illustrates the range and ranking of maintenance practices.

The latest and innovative approach to maintenance is called Operator Driven Reliability (ODR). This maintenance concept is simply a framework for organizing the activities of plant operations personnel in concert with a company's reliability maintenance practices. SKF has the knowledge and equipment to initiate and support this approach.

Maintenance engineering
Maintenance engineering is putting the strategy to work and includes for example the implementation of a "Computerized Maintenance Management System" (CMMS) with all the data and process information needed to achieve maintenance strategy goals.

Supply process
This service is an integral part of increasing profitability by reducing transaction costs, releasing capital tied up in spare inventory and making sure that the spares are available when needed.

Proactive reliability maintenance
Following the Proactive Reliability Maintenance process helps to provide best return on plant assets. It addresses failures and implements the processes necessary to prevent recurrence. The SKF Proactive Reliability process is based on four key steps:

- Predictive maintenance, a multi-faceted process that uses sophisticated technology systems to gather comprehensive intelligence on machine conditions and production processes.
- Diagnostics and Root Cause Analysis (RCA) to identify problems and necessary actions, such as machine alignment, balancing etc.
- Key performance indicators are performance improvement targets, established jointly between SKF and the customer.
- Periodic operational reviews between SKF and the plant management to analyze performance.

Machine maintenance
SKF Reliability Systems has developed its most comprehensive service programme for rotating equipment to drive machine maintenance in the most cost effective ways. This programme includes products and services such as

- machine alignment
- precision balancing
- lubrication management
- bearing analysis
- technology advice and machine upgrades
- bearing installation

Machine improvement
To remain competitive, plants must keep pace with new machine technologies. SKF can help to keep pace – without the need to invest in new machines. Recommendations can include one, or a combination of actions:

- Upgrade, rebuild and re-design
- Design engineering
- Refurbishment of bearings
- Repair and upgrade machine tool spindles
- Instrument/equipment calibrations

Training
SKF Reliability Services offers comprehensive machine reliability and asset management training – from the shop floor to the highest level of management.

Integrated Maintenance Solutions
An Integrated Maintenance Solution (IMS) agreement brings together all areas of expertize offered by SKF, establishing a continuous process of maintenance monitoring, analysis and improvement. It provides a planned skills transfer programme for maintenance and operations personnel, and technology upgrades where required.

With an IMS agreement, SKF Reliability Systems will manage every key component of a machine asset management strategy, providing a total system for improving efficiency. Each agreement is customized to specific business needs. The user can choose which areas need to be included, based on internal resources and current supplier contracts. With an IMS contract, SKF shares some of the risk as well as the savings, while the user receives agreed-upon financial returns with little to no capital investment.

@ptitude Industrial Decision Support System

The @ptitude Industrial Decision Support System from SKF is a knowledge management system that incorporates today's most advanced technologies to integrate data from multiple sources into an easy to use reliability maintenance application. It enhances the user team ability to make the right decision at the right time, providing a structured approach to capturing and applying knowledge. A key element of the @ptitude system is its online, web-enabled asset management knowledge bank: @ptitudeXchange subscribers have access to articles, technical handbooks, white papers, best practices and benchmarking information, interactive decision-support programs and an information network for expert advice and services. For additional information, please visit www.aptitudexchange.com.

Condition monitoring products

At the core of the SKF product range are the vibration detection, analysis and diagnostic products, which enable process monitoring as an added benefit. Some of those products are shown below. More information about SKF condition monitoring products can be found online at www.skf.com.

Microlog family of data collectors
The SKF family of Microlog data collector/analysers is designed so that users can easily establish a comprehensive periodic condition monitoring programme. As a diagnostic tool, the Microlog is unequalled in its class. Embedded intelligence provides step-by-step instructions for performing critical analysis functions like basic and advanced balancing, cyclic analysis, run-up/coast-down, bump test, tracking filter or motor current analysis. The frequency analysis module enables overlay defect frequencies on collected spectra to detect bearing defects, gear mesh, misalignment, unbalance or looseness problems.

MARLIN™ family of data managers
The MARLIN data management system is designed to be the frontline tool for operators, building the communications/technology bridge between operations, maintenance, engineering and plant management. This rugged high performance data collector provides a simple, convenient and portable way to collect and store machine vibration, process, and inspection data for quick downloading and analysis.

Vibration Pen plus

The Vibration Pen plus offers users a way to begin a cost-effective condition monitoring program, or to expand the responsibility for machine reliability to operators throughout the plant. A multi-parameter vibration monitoring tool, the Vibration Pen plus operates with the press of a button, measuring vibration according to ISO standards and utilizing acceleration enveloping technology to identify a range of bearing, gear mesh and other machinery problems.

Inspector 400 ultrasonic probe

The inspector 400 ultrasonic probe senses high frequency sounds produced by leaks, electrical discharges and equipment as it operates. It electronically translates these signals using a heterodyning process, making them audible through a headset and "visible" as increments on a meter. This enables maintenance personnel to detect pressure and vacuum leaks, arcing, tracking and corona in electric apparatus or test bearings, pumps, motors, compressors etc.

Infrared temperature probe

This laser sighted non-contact thermometer senses the temperature of an object with an infrared detector, enabling maintenance personnel to take temperature readings in locations that might otherwise be difficult to access.

Machine condition transmitter, online monitoring units
SKF machine condition transmitters provide vital information about bearing performance and overall machine condition. This information can be used to make sure that essential production equipment is kept running. This cost effective system offers two adjustable warning levels (alert and alarm) via two independent set points with LED alarm indicators and output relay contacts.

Online monitoring units provide around-the-clock automated data collection and a powerful array of analysis tools to optimize condition monitoring efforts. If a machine starts to develop a problem, the system helps to detect, analyse and track the defect so that maintenance costs are minimized. A "live" mode feature enables detailed online analysis, while event logs provide a history of events that may have occurred while the system was unattended.

Vibration sensors, eddy probes
SKF's in-depth bearing, machinery, monitoring and signal processing knowledge was included in the development of the CMSS2100 and CMSS2200 vibration sensor series. These single units can be used instead of the wide range of accelerometers typically needed to meet a variety of conditions.

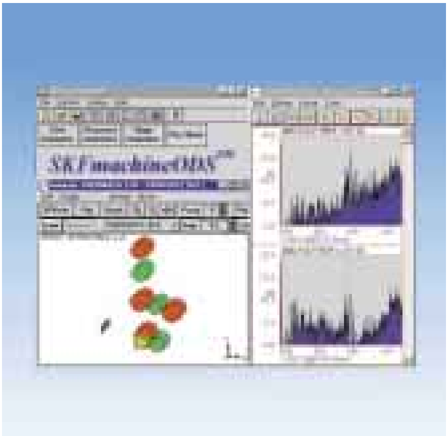
In addition to a full line of vibration sensors SKF also offers eddy probe systems for the measurement of relative motion in sleeve bearing machines.

Wireless sensors
Wireless sensors developed by SKF are ideal for online monitoring of machine condition in rotating equipment. Since they are wireless, difficult access is no problem and data can be collected safely from a distance. At the heart of the system is an SKF vibration sensor that is connected to a battery-powered unit transmitting the signals wirelessly to a base station. The wireless system is available as a stand-alone package or as a complement to the SKF cable based online system.

Machinery protection systems
DYMAC, an SKF Group company, offers total system integration by bringing advanced condition monitoring and protection systems into a plant-wide control platform. The end result is not only improved profitability, but also a better and safer environment. For example, the VM600 Machinery Monitoring System, a digital, modular, scaleable hardware and software solution for Plant Usage Optimization provides integrated machinery protection, condition and advanced performance monitoring from a single source.

For more information, please visit www.skf.com.

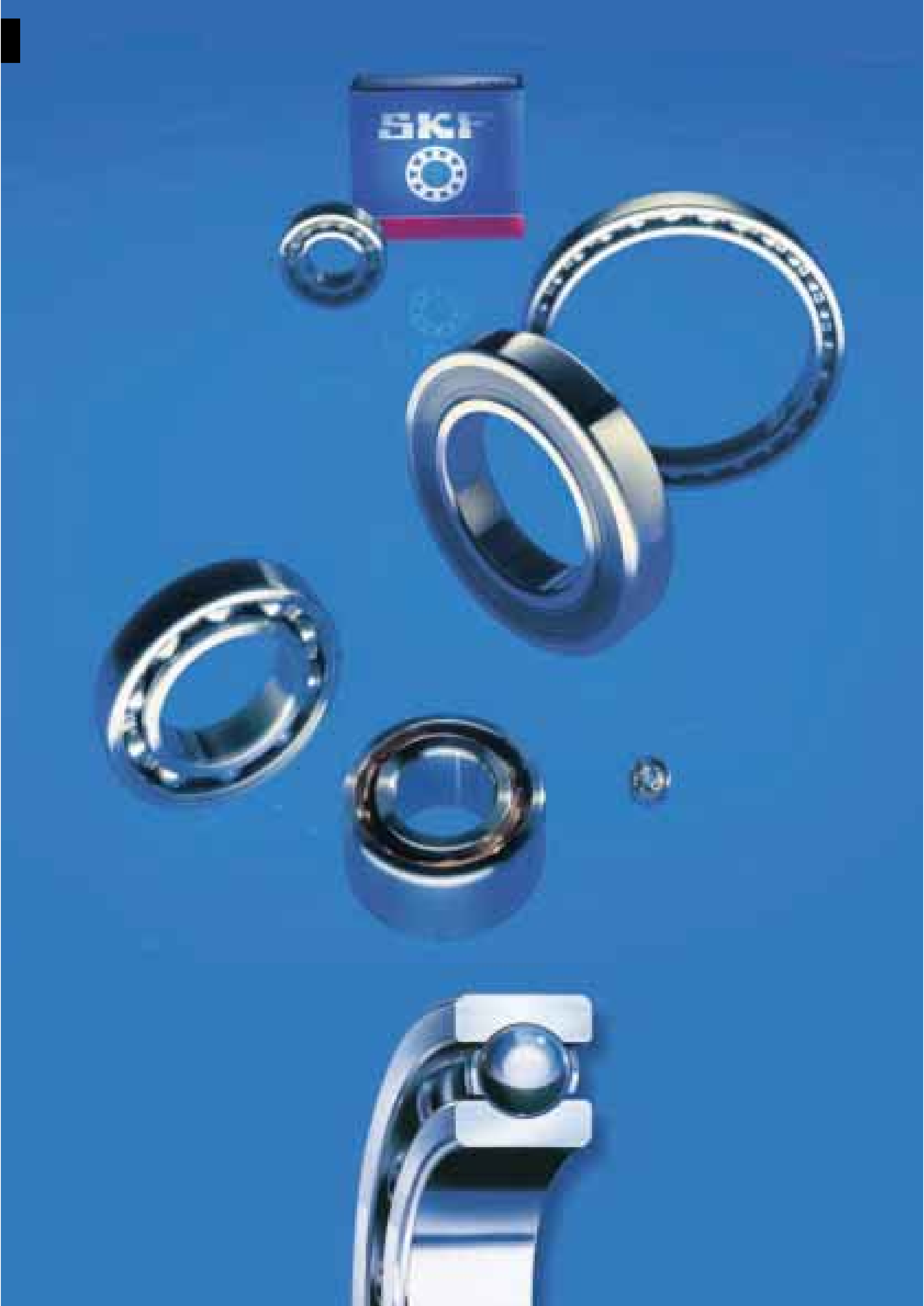
SKFmachine Operating Deflection Shape/Modal Analysis Software
SKFmachine ODS and SKFmachine SHAPE are easy to use, window-based software programmes to observe, analyse and document the dynamic behaviour of machinery. They help to easily identify and correct problems related to structural weakness and resonances in machinery.



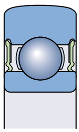
SKF Machine Analyst™
SKF Machine Analyst is the core component in a suite of software applications that provide a comprehensive reliability solution for process and manufacturing plants. It is the follow-up software solution of the successful PRISM4 suite. Written from the ground up using Component Object Model (COM) architecture, SKF Machine Analyst can be easily and effectively integrated with third party plug-ins, as well as systems such as computerized maintenance management systems, enterprise resource planning and others. Several versions are available, e.g. for online monitoring systems or to work with the MARLIN data mangement system. SKF Machine Analyst takes full advantage of the Microsoft Windows® functionality and features including multi-tasking, context-sensitive help, right-click functionality and the Windows Explorer™ graphical user interface.

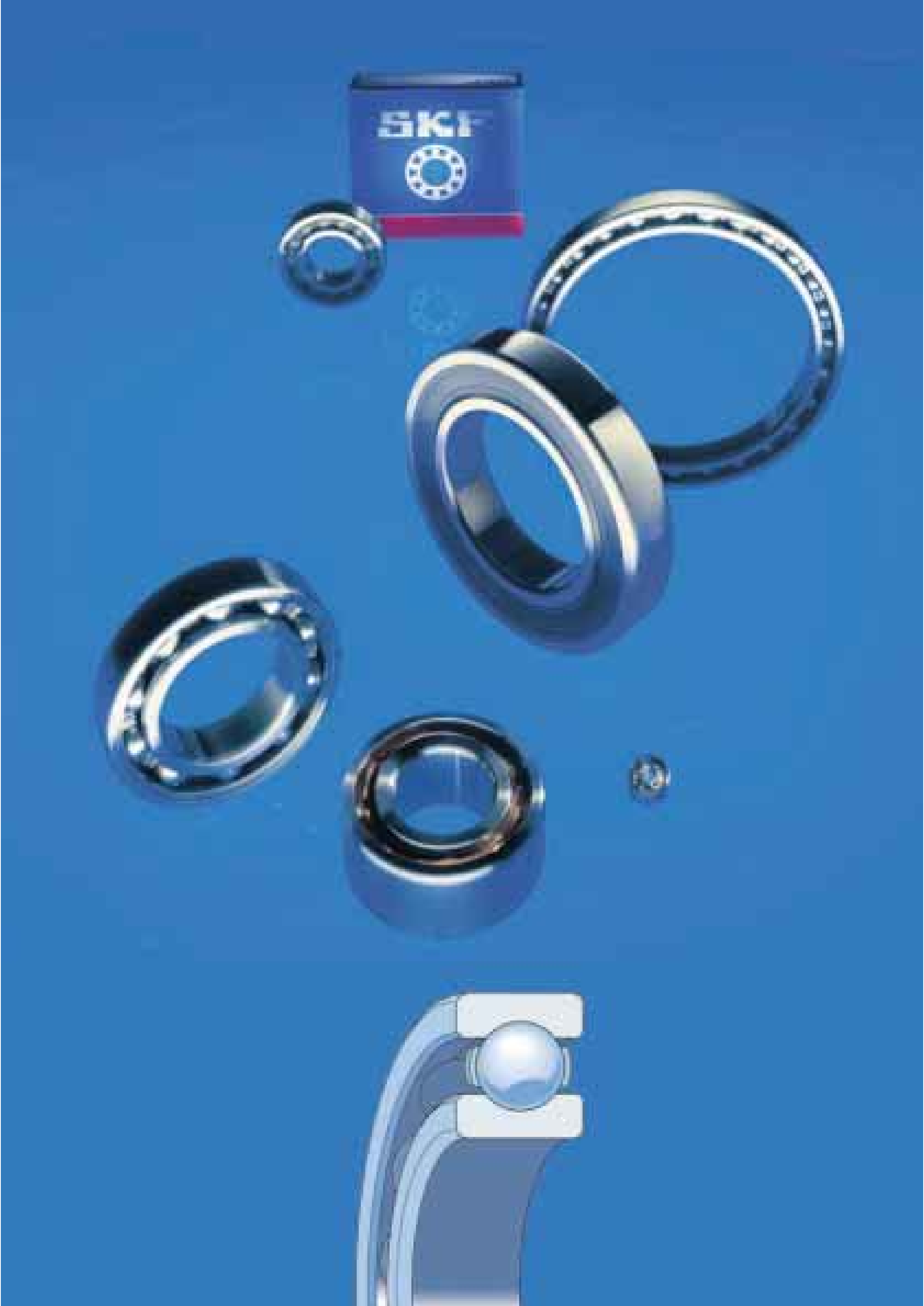

SKF
SKF

Product data


SKF

Deep groove ball bearings




SKF

Single row deep groove ball bearings

Single row deep groove ball bearings are particularly versatile. They are simple in design, non-separable, suitable for high and even very high speeds and are robust in operation, requiring little maintenance. Deep raceway grooves and the close conformity between the raceway grooves and the balls enable deep groove ball bearings to accommodate axial loads in both directions, in addition to radial loads, even at high speeds.

Single row deep groove ball bearings are the most widely used bearing type. Consequently, they are available from SKF in many executions and sizes:

- open basic design bearings
- sealed bearings
- ICOS™ oil sealed bearing units
- bearings with snap ring groove, with or without snap ring

Other deep groove ball bearings for special applications, shown in the sections “Engineering products” and “Mechatronics” include

- hybrid bearings (➔ **page 891**)
- insulated bearings (➔ **page 905**)
- high temperature bearings (➔ **page 917**)
- bearings with Solid Oil (➔ **page 945**)
- sensorized bearings (➔ **page 953**)

The SKF product range also includes inch-size bearings and bearings with a tapered bore. These variants are not included in this General Catalogue. Information will be provided on request.

Designs

Basic design bearings

Basic design SKF single row deep groove ball bearings (➔ **fig 1**) are open (unsealed). For manufacturing reasons, those sizes of open bearing that are also produced in sealed or shielded versions may have seal recesses in the outer ring.

Sealed bearings

The most popular sizes of deep groove ball bearings are also produced in sealed versions with shields or contact seals on one or both sides. Details regarding the suitability of the different seals for various operating conditions will be found in **table 1**. Sealed bearings in the wide 622, 623 and 630 series are particularly suitable for long maintenance-free service. In addition, ICOS bearing units with integrated radial shaft seals are available for higher sealing requirements.

The bearings with shields or seals on both sides are lubricated for life and are maintenance-free. They should not be washed or heated to temperatures above 80 °C. Depending on the series and size, deep groove ball bearings are supplied charged with one of three standard greases:

- LT10 grease for bearings in the 8 and 9 Diameter Series up to and including 30 mm outside diameter,

Fig 1

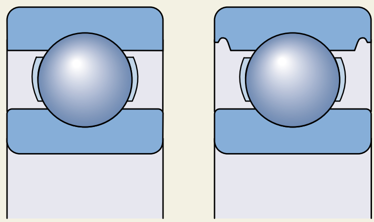


Table 1

Seal selection guidelines

Requirement	Shields Z	Low-friction seals RSL	RZ	Contact seals RSH	RS1
Low friction	+++	++	+++	o	o
High speeds	+++	+++	+++	o	o
Grease retention	o	+++	+	+++	++
Dust exclusion	o	++	+	+++	+++
Water exclusion					
static	–	o	–	+++	++
dynamic	–	o	–	+	+
high pressure	–	o	–	+++	o

Symbols: +++ excellent ++ very good + good o fair – not recommended

- MT47 grease for bearings in the 8 and 9 Diameter Series above 30 mm up to and including 62 mm outside diameter and for bearings in the 0, 1, 2 and 3 Diameter Series up to and including 62 mm outside diameter,
- MT33 grease for all bearings above 62 mm outside diameter.

Characteristics of the above standard greases are listed in **table 2**. The standard grease is not identified in the bearing designation. The quantity of grease fills some 25 to 35 % of the free space in the bearing. To special order, other grease filling grades are available. Also on request, special grease fills (➔ **table 2**) can be supplied

- high temperature grease GJN for bearings up to and including 62 mm outside diameter
- high temperature grease HT22 for bearings above 62 mm outside diameter
- low temperature grease LT20
- wide temperature range grease GWB
- wide temperature range and silent running grease LHT23

Table 2

SKF grease filling for sealed deep groove ball bearings

Technical specifications	Standard greases LT10	MT47	MT33	Special greases GJN	HT22	LT20	GWB	LHT23
Thickener	Lithium soap	Lithium soap	Lithium soap	Polyurea soap	Lithium complex soap	Lithium soap	Polyurea soap	Lithium soap
Base oil type	Diester oil	Mineral oil	Mineral oil	Mineral oil	Mineral oil	Diester oil	Ester oil	Ester oil
NLGI consistency class	2	2	3	2	3	2	2–3	2
Operating temperature, °C	–50 to +90	–30 to +110	–30 to +120	–30 to +150	–20 to +140	–55 to +110	–40 to +160	–50 to +140
Base oil viscosity, mm^2/s								
at 40 °C	12	70	74	115	110	15	70	26
at 100 °C	3,3	7,3	8,5	12,2	13	3,7	9,4	5,1

Bearings with shields

Bearings with shields, designation suffix Z or 2Z, are produced in one of two designs, depending on the bearing series and size (➔ **fig** 2). The shields are made of sheet steel and normally have a cylindrical extension in the shield bore to form a long sealing gap with the inner ring shoulder (**a**). Some shields do not have the extension (**b**).

Shielded bearings are primarily intended for applications where the inner ring rotates. If the outer ring rotates, there is a risk that the grease will leak from the bearing at high speeds.

Bearings with low-friction seals

SKF deep groove ball bearings with low-friction seals, designation suffixes RSL, 2RSL or RZ, 2RZ, are manufactured in three designs depending on bearing series and size (➔ **fig** 3):

- bearings in the 60, 62 and 63 series up to 25 mm outside diameter are equipped with RSL seals to design (**a**),
- bearings in the 60, 62 and 63 series from 25 mm and up to and including 52 mm outside diameter are equipped with RSL seals to design (**b**),
- other bearings have RZ seals (**c**).

The seals form an extremely narrow gap with the cylindrical surface of the inner ring shoulder or recess profile and are practically non-contacting. Because of this, bearings fitted with low-friction seals can be operated at the same high speeds as bearings with Z shields, but with improved seal performance.

The low-friction seals are made of oil and wear-resistant acrylonitrile butadiene rubber (NBR) with a sheet steel reinforcement. The permissible operating temperature range for these seals is –40 to +100 °C and up to +120 °C for brief periods.

Bearings with contact seals

Bearings with contact seals, designation suffixes RSH, 2RSH or RS1, 2RS1, are manufactured in four designs depending on bearing series and size (➔ **fig** 4):

- bearings in the 60, 62, and 63 series up to 25 mm outside diameter are equipped with RSH seals to design (**a**),
- bearings in the 60, 62 and 63 series from 25 mm and up to and including 52 mm outside diameter are equipped with RSH seals to design (**b**),
- other bearings have RS1 seals, which seal against the cylindrical surface of the inner ring shoulder (**c**) indicated by dimension d_1 in the product table or against a recess in the inner ring side face (**d**) indicated by dimension d_2 in the product table.

The seals are inserted in recesses in the outer ring and provide good sealing at this

Fig 2

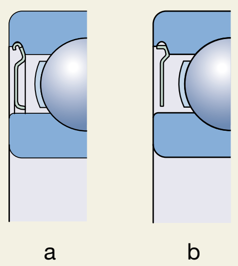


Fig 3

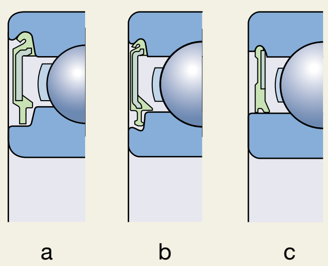
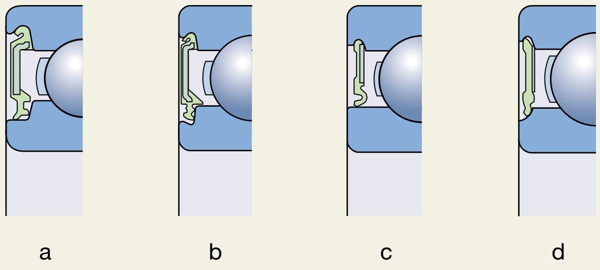


Fig 4



position without deforming the outer ring. Standard seals are made of acrylonitrile butadiene rubber (NBR) with a sheet steel reinforcement. The permissible operating temperature range for these seals is –40 to +100 °C and up to +120 °C for brief periods.

When sealed bearings are operated under certain extreme conditions, e.g. very high speeds or high temperatures, grease leakage may occur at the inner ring. For bearing arrangements where this would be detrimental, special design steps must be undertaken, please consult the SKF application engineering service.

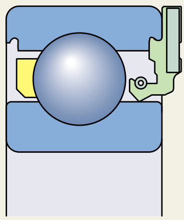
ICOS™ oil sealed bearing units

ICOS oil sealed bearing units have been developed by SKF. The new concept aims at applications where sealing requirements exceed the capabilities of standard sealed bearings. An ICOS unit consists of a 62 series deep groove ball bearing and an integral CR radial shaft seal (➔ **fig** 5). These units need less space than common two-component arrangements; they simplify mounting, and avoid expensive machining of the shaft because the inner ring shoulder serves as a perfect seal counterface.

The CR radial shaft seal is made of acrylonitrile butadiene rubber (NBR) and has a spring loaded Waveseal lip. The permissible operating temperature range for the seal is –40 to +100 °C and up to +120 °C for brief periods.

The speed limits quoted in the product table are based on the permissible circumferential speed for the CR seal, which in this case is 14 m/s.

Fig 5



Bearings with snap ring groove

Deep groove ball bearings with a snap ring groove can simplify arrangement design as the bearings can be axially located in the housing by a snap (or retaining) ring (➔ **fig** 6). This saves space. Appropriate snap rings are shown in the product table with designation and dimensions and may be supplied separately or already mounted on the bearing.

SKF deep groove ball bearings with a snap ring groove (➔ **fig** 7) are supplied as:

- open (unsealed) bearings, designation suffix N (**a**);
- open bearings with a snap ring, designation suffix NR (**b**);
- bearings with a Z shield at the opposite side and a snap ring, designation suffix ZNR (**c**);
- bearings with Z shields on both sides and a snap ring, designation suffix 2ZNR (**d**).

Matched bearing pairs

For bearing arrangements where the load carrying capacity of a single bearing is inadequate, or where the shaft has to be axially located in both directions with a given amount of axial clearance, SKF can supply matched pairs of single row deep groove ball bearings to order. Depending on the requirements the matched pairs can be supplied in tandem, back-to-back, or face-to-face arrangements (➔ **fig** 8). The bearings are matched in production so that, when mounted immediately adjacent to each other, the load will be evenly distributed between the bearings without having to use shims or similar devices.

Further information on matched bearing pairs can be found in the “SKF Interactive Engineering Catalogue” on CD-ROM or online at www.skf.com.

Fig 6

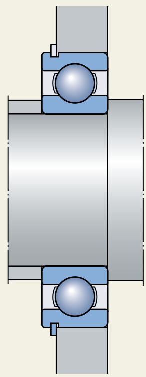
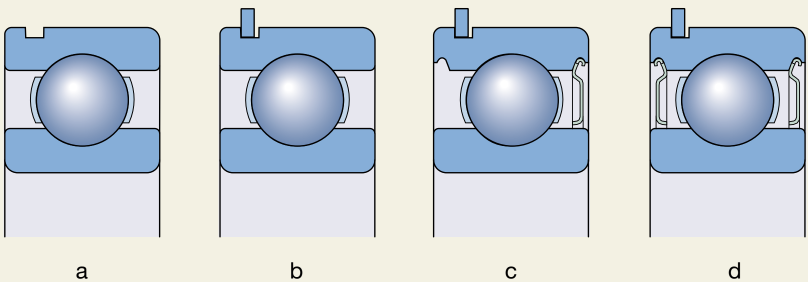


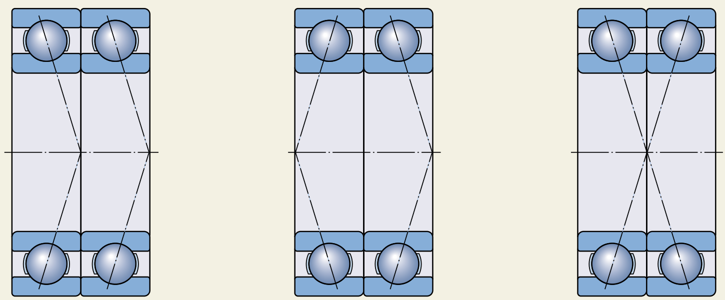
Fig 7



SKF Explorer class bearings

High performance SKF Explorer deep groove ball bearings are shown with an asterisk in the product tables. The higher performance of SKF Explorer deep groove ball bearings also includes quieter running. SKF Explorer bearings retain the designation of the earlier standard bearings. However, each bearing and its box are marked with the name “EXPLORER”.

Fig 8



Bearing data – general

Dimensions
The boundary dimensions of SKF single row deep groove ball bearings are in accordance with ISO 15:1998. Dimensions of the snap ring grooves and snap rings comply with ISO 464:1995.

Tolerances
SKF single row deep groove ball bearings are manufactured as standard to Normal tolerances.

SKF Explorer single row deep groove ball bearings are produced to higher precision than the ISO Normal tolerances. The dimensional accuracy corresponds to P6 tolerances, except the width tolerance, which is considerably tighter and reduced to

- 0/–60 µm for bearings with outside diameter up to 110 mm and
- 0/–100 µm for larger bearings.

The running accuracy depends on the bearing size and corresponds to

- P5 tolerances for bearings up to 52 mm outside diameter,
- P6 tolerances for bearings above 52 mm up to 110 mm outside diameter and
- Normal tolerances for larger bearings.

For bearing arrangements where accuracy is a key operational factor some SKF single row deep groove ball bearings are also available with accuracy completely to P6 or P5 tolerance class specifications. The availability of these bearings should always be checked before ordering.

The tolerances are in accordance with ISO 492:2002 and can be found in **tables** 3 to 5, starting on **page 125**.

Internal clearance
SKF single row deep groove ball bearings are manufactured with Normal radial internal clearance as standard. Most of the bearings are also available with C3 radial internal clearance. Some of the bearings can even be supplied with the appreciably greater C4 or the smaller C2 clearances. In addition, deep groove ball bearings are available with reduced or displaced internal clearance ranges. These special clearances may use reduced ranges of standard clearance classes or partitions of adjacent classes (➔ designation suffix CN on **page 300**). Bearings with internal clearance not to standard are supplied on request.

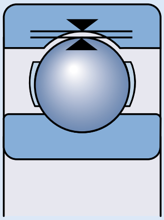
The values for radial internal clearance are given in **table** 3. They are in accordance with ISO 5753:1991 and are valid for unmounted bearings under zero measuring load.

Misalignment
Single row deep groove ball bearings have only limited ability to accommodate misalignment. The permissible angular misalignment between the inner and outer rings, which will not produce inadmissibly high additional stresses in the bearing, depends on

- the radial internal clearance of the bearing in operation,
- the bearing size,
- the internal design and
- the forces and moments acting on the bearing.

Because of the complex relationship between these factors, no generally applicable specific values can be given. However, depending on the various influences of the factors, the permissible angular misalignment lies between 2 and 10 minutes of arc. Any misalignment will result in increased bearing noise and reduced bearing service life.

Radial internal clearance of deep groove ball bearings



Bore diameter d		Radial internal clearance C2		Normal		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
mm		µm									
	6	0	7	2	13	8	23	–	–	–	–
6	**10**	0	7	2	13	8	23	14	29	20	37
10	**18**	0	9	3	18	11	25	18	33	25	45
18	**24**	0	10	5	20	13	28	20	36	28	48
24	**30**	1	11	5	20	13	28	23	41	30	53
30	**40**	1	11	6	20	15	33	28	46	40	64
40	**50**	1	11	6	23	18	36	30	51	45	73
50	**65**	1	15	8	28	23	43	38	61	55	90
65	**80**	1	15	10	30	25	51	46	71	65	105
80	**100**	1	18	12	36	30	58	53	84	75	120
100	**120**	2	20	15	41	36	66	61	97	90	140
120	**140**	2	23	18	48	41	81	71	114	105	160
140	**160**	2	23	18	53	46	91	81	130	120	180
160	**180**	2	25	20	61	53	102	91	147	135	200
180	**200**	2	30	25	71	63	117	107	163	150	230
200	**225**	4	32	28	82	73	132	120	187	175	255
225	**250**	4	36	31	92	87	152	140	217	205	290
250	**280**	4	39	36	97	97	162	152	237	255	320
280	**315**	8	45	42	110	110	180	175	260	260	360
315	**355**	8	50	50	120	120	200	200	290	290	405
355	**400**	8	60	60	140	140	230	230	330	330	460
400	**450**	10	70	70	160	160	260	260	370	370	520
450	**500**	10	80	80	180	180	290	290	410	410	570
500	**560**	20	90	90	200	200	320	320	460	460	630
560	**630**	20	100	100	220	220	350	350	510	510	700
630	**710**	30	120	120	250	250	390	390	560	560	780
710	**800**	30	130	130	280	280	440	440	620	620	860
800	**900**	30	150	150	310	310	490	490	690	690	960
900	**1 000**	40	160	160	340	340	540	540	760	760	1 040
1 000	**1 120**	40	170	170	370	370	590	590	840	840	1 120
1 120	**1 250**	40	180	180	400	400	640	640	910	910	1 220
1 250	**1 400**	60	210	210	440	440	700	700	1 000	1 000	1 340
1 400	**1 600**	60	230	230	480	480	770	770	1 100	1 100	1 470

Please refer to **page 137** for definition of radial internal clearance

Cages

Depending on the bearing series and size, SKF single row deep groove ball bearings are supplied with one of the following cages (➔ **fig 9**):

- ribbon-type cage of steel or brass sheet (**a**)
- riveted cage of steel or brass sheet (**b**)
- machined brass cage (**c**)
- snap-type cage of polyamide 6,6 (**d**)

Bearings having a pressed steel cage in standard execution may also be available with a machined brass or polyamide cage. For higher operating temperatures, polyamide 4,6 or PEEK cages may be advantageous. Before ordering, please check for availability.

Note:
Deep groove ball bearings with polyamide 6,6 cages can be operated at temperatures up to +120 °C. The lubricants generally used for rolling bearings do not have a detrimental effect on cage properties, with the exception of a few synthetic oils and greases with a synthetic oil base and lubricants containing a high proportion of EP additives when used at high temperatures.

For bearing arrangements, which are to be operated at continuously high temperatures or under arduous conditions, SKF recommends using bearings with a pressed steel or a machined brass cage.

For detailed information regarding the temperature resistance and the applicability of cages, please refer to the section "Cage materials", starting on **page 140**.

Minimum load

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

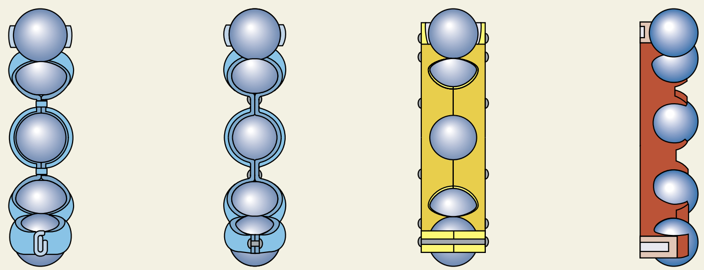
The requisite minimum radial load to be applied to deep groove ball bearings can be estimated using

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

where

F_{rm} = minimum radial load, kN
k_r = minimum load factor (➔ product tables)
ν = oil viscosity at operating temperature, mm^2/s
n = rotational speed, r/min
d_m = bearing mean diameter = 0,5 (d + D), mm

Fig 9



When starting up at low temperatures or when the lubricant is highly viscous, even greater minimum loads may be required. The weight of the components supported by the bearing, together with external forces, generally exceeds the requisite minimum load. If this is not the case, the deep groove ball bearing must be subjected to an additional radial load. For applications where deep groove ball bearings are used, an axial preload can be applied by adjusting the inner and outer rings against each other, or by using springs.

Axial load carrying capacity
If deep groove ball bearings are subjected to purely axial load, this axial load should generally not exceed the value of 0,5 C_0. Small bearings (bore diameter up to approx. 12 mm) and light series bearings (Diameter Series 8, 9, 0, and 1) should not be subjected to an axial load greater than 0,25 C_0. Excessive axial loads can lead to a considerable reduction in bearing service life.

Equivalent dynamic bearing load
For dynamically loaded single row deep groove ball bearings

$P = F_r$ when $F_a/F_r \leq e$
$P = XF_r + YF_a$ when $F_a/F_r > e$

The factors e, X and Y depend on the relationship $f_0\,F_a/C_0$, where f_0 is a calculation factor (➔ product tables), F_a the axial component of the load and C_0 the basic static load rating.

In addition, the factors are influenced by the magnitude of the radial internal clearance; increased clearance allows heavier axial loads to be supported. For bearings mounted with the usual fits (shaft tolerance j5 to n6 depending on the shaft diameter, and housing bore tolerance J7), the values for e, X and Y are listed in **table 4**. If a clearance greater than Normal is chosen because a reduction in clearance is expected in operation, the values given under “Normal clearance” should be used.

Equivalent static bearing load
For statically loaded single row deep groove ball bearings

$P_0 = 0,6\,F_r + 0,5\,F_a$

If $P_0 < F_r$, $P_0 = F_r$ should be used.

Table 4

Calculation factors for single row deep groove ball bearings

$f_0\,F_a/C_0$	Normal clearance e	X	Y	C3 clearance e	X	Y	C4 clearance e	X	Y
0,172	0,19	0,56	2,30	0,29	0,46	1,88	0,38	0,44	1,47
0,345	0,22	0,56	1,99	0,32	0,46	1,71	0,40	0,44	1,40
0,689	0,26	0,56	1,71	0,36	0,46	1,52	0,43	0,44	1,30
1,03	0,28	0,56	1,55	0,38	0,46	1,41	0,46	0,44	1,23
1,38	0,30	0,56	1,45	0,40	0,46	1,34	0,47	0,44	1,19
2,07	0,34	0,56	1,31	0,44	0,46	1,23	0,50	0,44	1,12
3,45	0,38	0,56	1,15	0,49	0,46	1,10	0,55	0,44	1,02
5,17	0,42	0,56	1,04	0,54	0,46	1,01	0,56	0,44	1,00
6,89	0,44	0,56	1,00	0,54	0,46	1,00	0,56	0,44	1,00

Intermediate values are obtained by linear interpolation

Supplementary designations
The designation suffixes used to identity certain features of SKF deep groove ball bearings are explained in the following.

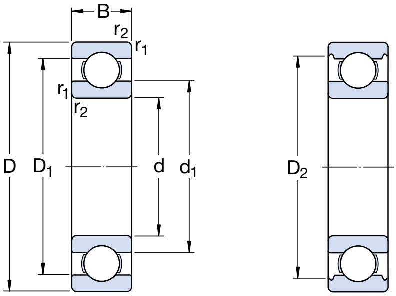
CN Normal radial clearance; generally only used in combination with one of the following letters that indicate reduced or displaced clearance range
- **H** reduced clearance range corresponding to the upper half of the actual clearance range
- **L** reduced clearance range corresponding to the lower half of the actual clearance range
- **P** displaced clearance range comprising the upper half of the actual clearance range plus the lower half of the next larger clearance range

The above letters are also used together with the following clearance classes: C2, C3, and C4

C2 Radial internal clearance less than Normal
C3 Radial internal clearance greater than Normal
C4 Radial internal clearance greater than C3
C5 Radial internal clearance greater than C4
DB Two single row deep groove ball bearings matched for paired mounting in a back-to-back arrangement
DF Two single row deep groove ball bearings matched for paired mounting in a face-to-face arrangement
DT Two single row deep groove ball bearings matched for paired mounting in a tandem arrangement
E Reinforced ball set
GJN Polyurea base grease of consistency 2 to the NLGI Scale for a temperature range –30 to +150 °C (normal fill grade)
HT Lithium base grease of consistency 3 to the NLGI Scale for a temperature range –20 to +140 °C (normal fill grade)
J Pressed steel cage
LHT23 Lithium base grease of consistency 2 to the NLGI Scale for a temperature range –50 to +140 °C (normal fill grade)
LT Lithium base grease of consistency 2 to the NLGI Scale for a temperature range –55 to +110 °C (normal fill grade)
LT10 Lithium base grease of consistency 2 to the NLGI Scale for a temperature range –50 to +90 °C (normal fill grade)
M Machined brass cage, ball centred. Different designs and material grades are identified by a figure following the M, e.g. M2
MA Machined brass cage, outer ring centred
MB Machined brass cage, inner ring centred
MT33 Lithium base grease of consistency 3 to the NLGI Scale for a temperature range –30 to +120 °C (normal fill grade)
MT47 Lithium base grease of consistency 2 to the NLGI Scale for a temperature range –30 to +110 °C (normal fill grade)
N Snap ring groove in the outer ring
NR Snap ring groove in the outer ring, with snap ring
N1 One notch in the outer ring side face (enabling stop to be used to prevent rotation of ring)
P5 Dimensional and running accuracy to ISO tolerance class 5
P6 Dimensional and running accuracy to ISO tolerance class 6
P52 P5 + C2
P62 P6 + C2
P63 P6 + C3
RS1 Acrylonitrile butadiene rubber (NBR) seal with sheet steel reinforcement on one side of the bearing
RSH Acrylonitrile butadiene rubber (NBR) seal with sheet steel reinforcement on one side of the bearing
RSL Low friction, acrylonitrile butadiene rubber (NBR) seal with sheet steel reinforcement on one side of the bearing

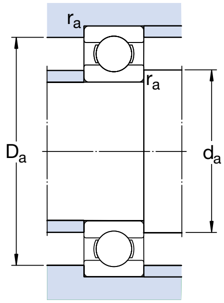
RZ	Low friction, acrylonitrile butadiene rubber (NBR) seal with sheet steel reinforcement on one side of the bearing
TH	Cage of fabric reinforced phenolic resin (snap-type)
TN	Injection moulded polyamide cage
TN9	Injection moulded glass fibre reinforced polyamide 6,6 cage
VL0241	Aluminium oxide coated outside surface of the outer ring for electrical resistance up to 1 000 V DC
VL2071	Aluminium oxide coated outside surface of the inner ring for electrical resistance up to 1 000 V DC
WT	Polyurea base grease of consistency 2–3 to the NLGI Scale for a temperature range –40 to +160 °C (normal fill grade)
Y	Pressed brass cage
Z	Pressed steel shield on one side of the bearing
2RS1	Acrylonitrile butadiene rubber (NBR) seal with sheet steel reinforcement on both sides of the bearing
2RSH	Acrylonitrile butadiene rubber (NBR) seal with sheet steel reinforcement on both sides of the bearing
2RSL	Low friction, acrylonitrile butadiene rubber (NBR) seal with sheet steel reinforcement on both sides of the bearing
2RZ	Low friction, acrylonitrile butadiene rubber (NBR) seal with sheet steel reinforcement on both sides of the bearing
2Z	Z shield on both sides of the bearing

Single row deep groove ball bearings
d **3 – 10** mm



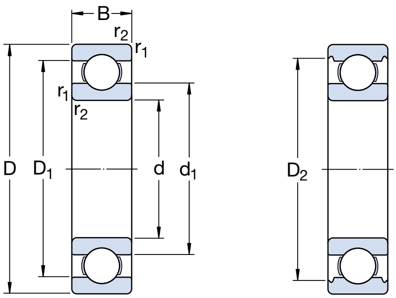
Principal dimensions d	D	B	Basic load ratings dynamic C	static C_0	Fatigue load limit P_u	Speed ratings Reference speed	Limiting speed	Mass	Designation
mm			kN		kN	r/min		kg	–
3	10	4	0,54	0,18	0,007	130 000	80 000	0,0015	**623**
4	9	2,5	0,54	0,18	0,007	140 000	85 000	0,0007	**618/4**
	11	4	0,715	0,232	0,010	130 000	80 000	0,0017	**619/4**
	12	4	0,806	0,28	0,012	120 000	75 000	0,0021	**604**
	13	5	0,936	0,29	0,012	110 000	67 000	0,0031	**624**
	16	5	1,11	0,38	0,016	95 000	60 000	0,0054	**634**
5	11	3	0,637	0,255	0,011	120 000	75 000	0,0012	**618/5**
	13	4	0,884	0,34	0,014	110 000	67 000	0,0025	**619/5**
	16	5	1,14	0,38	0,016	95 000	60 000	0,0050	* **625**
	19	6	2,34	0,95	0,04	80 000	50 000	0,0090	* **635**
6	13	3,5	0,884	0,345	0,015	110 000	67 000	0,0020	**618/6**
	15	5	1,24	0,475	0,02	100 000	63 000	0,0039	**619/6**
	19	6	2,34	0,95	0,04	80 000	50 000	0,0084	* **626**
7	14	3,5	0,956	0,4	0,017	100 000	63 000	0,0022	**618/7**
	17	5	1,48	0,56	0,024	90 000	56 000	0,0049	**619/7**
	19	6	2,34	0,95	0,04	85 000	53 000	0,0075	* **607**
	22	7	3,45	1,37	0,057	70 000	45 000	0,013	* **627**
8	16	4	1,33	0,57	0,024	90 000	56 000	0,0030	**618/8**
	19	6	1,9	0,735	0,031	80 000	50 000	0,0071	**619/8**
	22	7	3,45	1,37	0,057	75 000	48 000	0,012	* **608**
	24	8	3,9	1,66	0,071	63 000	40 000	0,017	* **628**
9	17	4	1,43	0,64	0,027	85 000	53 000	0,0034	**618/9**
	20	6	2,08	0,865	0,036	80 000	48 000	0,0076	**619/9**
	24	7	3,9	1,66	0,071	70 000	43 000	0,014	* **609**
	26	8	4,75	1,96	0,083	60 000	38 000	0,020	* **629**
10	19	5	1,38	0,585	0,025	80 000	48 000	0,0055	**61800**
	22	6	2,08	0,85	0,036	75 000	45 000	0,010	**61900**
	26	8	4,75	1,96	0,083	67 000	40 000	0,019	* **6000**
	28	8	4,62	1,96	0,083	63 000	40 000	0,022	**16100**
	30	9	5,4	2,36	0,1	56 000	34 000	0,032	* **6200**
	35	11	8,52	3,4	0,143	50 000	32 000	0,053	* **6300**

* SKF Explorer bearing



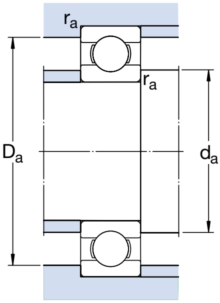
Dimensions					Abutment and fillet dimensions			Calculation factors	
d	d_1 ~	D_1 ~	D_2 ~	$r_{1,2}$ min	d_a min	D_a max	r_a max	k_r	f_0
mm					mm			–	
3	5,2	7,5	8,2	0,15	4,2	8,8	0,1	0,025	7,5
4	5,2	7,5	–	0,1	4,6	8,4	0,1	0,015	10
	5,9	9	9,8	0,15	4,8	10,2	0,1	0,02	9,9
	6,1	9	–	0,2	5,4	10,6	0,2	0,025	10
	6,7	10,3	11,2	0,2	5,8	11,2	0,2	0,025	10
	8,4	12	13,3	0,3	6,4	13,6	0,3	0,025	8,4
5	6,8	9,3	–	0,15	5,8	10,2	0,1	0,015	11
	7,6	10,8	11,4	0,2	6,4	11,6	0,2	0,02	11
	8,4	12	13,3	0,3	7,4	13,6	0,3	0,025	8,4
	10,7	15,3	16,5	0,3	7,4	16,6	0,3	0,03	13
6	7,9	11,2	–	0,15	6,8	12,2	0,1	0,015	11
	8,6	12,4	13,3	0,2	7,4	13,6	0,2	0,02	10
	11,1	15,2	16,5	0,3	8,4	16,6	0,3	0,025	13
7	8,9	12,2	–	0,15	7,8	13,2	0,1	0,015	11
	9,8	14,2	15,2	0,3	9	15	0,3	0,02	10
	11,1	15,2	16,5	0,3	9	17	0,3	0,025	13
	12,2	17,6	19,2	0,3	9,4	19,6	0,3	0,025	12
8	10,1	14	–	0,2	9,4	14,6	0,2	0,015	11
	11,1	16,1	19	0,3	10	17	0,3	0,02	10
	12,1	17,6	19,2	0,3	10	20	0,3	0,025	12
	14,5	19,8	20,6	0,3	10,4	21,6	0,3	0,025	13
9	11,1	15	–	0,2	10,4	15,6	0,2	0,015	11
	12	17	17,9	0,3	11	18	0,3	0,02	11
	14,4	19,8	21,2	0,3	11	22	0,3	0,025	13
	14,8	21,2	22,6	0,3	11,4	23,6	0,3	0,025	12
10	12,6	16,4	–	0,3	12	17	0,3	0,015	9,4
	13	18,1	19	0,3	12	20	0,3	0,02	9,3
	14,8	21,2	22,6	0,3	12	24	0,3	0,025	12
	16,7	23,4	24,8	0,6	14,2	23,8	0,3	0,025	13
	17	23,2	24,8	0,6	14,2	25,8	0,6	0,025	13
	17,5	26,9	28,7	0,6	14,2	30,8	0,6	0,03	11

Single row deep groove ball bearings
d **12 – 22** mm



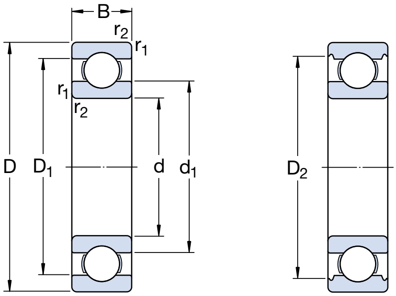
Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	B	C	C_0	P_u				
mm			kN		kN	r/min		kg	–
12	21	5	1,43	0,67	0,028	70 000	43 000	0,0063	**61801**
	24	6	2,25	0,98	0,043	67 000	40 000	0,011	**61901**
	28	8	5,4	2,36	0,10	60 000	38 000	0,022	* **6001**
	30	8	5,07	2,36	0,10	56 000	34 000	0,023	**16101**
	32	10	7,28	3,1	0,132	50 000	32 000	0,037	* **6201**
	37	12	10,1	4,15	0,176	45 000	28 000	0,060	* **6301**
15	24	5	1,56	0,8	0,034	60 000	38 000	0,0074	**61802**
	28	7	4,36	2,24	0,095	56 000	34 000	0,016	**61902**
	32	8	5,85	2,85	0,12	50 000	32 000	0,025	* **16002**
	32	9	5,85	2,85	0,12	50 000	32 000	0,030	* **6002**
	35	11	8,06	3,75	0,16	43 000	28 000	0,045	* **6202**
	42	13	11,9	5,4	0,228	38 000	24 000	0,082	* **6302**
17	26	5	1,68	0,93	0,039	56 000	34 000	0,0082	**61803**
	30	7	4,62	2,55	0,108	50 000	32 000	0,018	**61903**
	35	8	6,37	3,25	0,137	45 000	28 000	0,032	* **16003**
	35	10	6,37	3,25	0,137	45 000	28 000	0,039	* **6003**
	40	9	9,56	4,75	0,2	38 000	24 000	0,048	**98203**
	40	12	9,95	4,75	0,2	38 000	24 000	0,065	* **6203**
	40	12	11,4	5,4	0,228	38 000	24 000	0,064	**6203 ETN9**
	47	14	14,3	6,55	0,275	34 000	22 000	0,12	* **6303**
	62	17	22,9	10,8	0,455	28 000	18 000	0,27	**6403**
20	32	7	4,03	2,32	0,104	45 000	28 000	0,018	**61804**
	37	9	6,37	3,65	0,156	43 000	26 000	0,038	**61904**
	42	8	7,28	4,05	0,173	38 000	24 000	0,050	* **16004**
	42	9	7,93	4,5	0,19	38 000	24 000	0,051	**98204 Y**
	42	12	9,95	5	0,212	38 000	24 000	0,069	* **6004**
	47	14	13,5	6,55	0,28	32 000	20 000	0,11	* **6204**
	47	14	15,6	7,65	0,325	32 000	20 000	0,096	**6204 ETN9**
	52	15	16,8	7,8	0,335	30 000	19 000	0,14	* **6304**
	52	15	18,2	9	0,38	30 000	19 000	0,14	**6304 ETN9**
	72	19	30,7	15	0,64	24 000	15 000	0,40	**6404**
22	50	14	14	7,65	0,325	30 000	19 000	0,12	**62/22**
	56	16	18,6	9,3	0,39	28 000	18 000	0,18	**63/22**

* SKF Explorer bearing



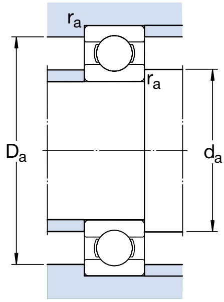
Dimensions					Abutment and fillet dimensions			Calculation factors	
d	d_1 ~	D_1 ~	D_2 ~	$r_{1,2}$ min	d_a min	D_a max	r_a max	k_r	f_0
mm					mm			–	
12	15	18,2	–	0,3	14	19	0,3	0,015	9,7
	15,5	20,6	21,4	0,3	14	22	0,3	0,02	9,7
	17	23,2	24,8	0,3	14	26	0,3	0,025	13
	16,7	23,4	24,8	0,3	14,4	27,6	0,3	0,025	13
	18,5	25,7	27,4	0,6	16,2	27,8	0,6	0,025	12
	19,5	29,5	31,5	1	17,6	31,4	1	0,03	11
15	17,9	21,1	–	0,3	17	22	0,3	0,015	10
	18,4	24,7	25,8	0,3	17	26	0,3	0,02	14
	20,2	27	28,2	0,3	17	30	0,3	0,02	14
	20,5	26,7	28,2	0,3	17	30	0,3	0,025	14
	21,7	29	30,4	0,6	19,2	30,8	0,6	0,025	13
	23,7	33,7	36,3	1	20,6	36,4	1	0,03	12
17	20,2	23,2	–	0,3	19	24	0,3	0,015	10
	20,4	26,7	27,8	0,3	19	28	0,3	0,02	15
	22,7	29,5	31,2	0,3	19	33	0,3	0,02	14
	23	29,2	31,4	0,3	19	33	0,3	0,025	14
	24,5	32,7	–	0,6	21,2	35,8	0,6	0,025	13
	24,5	32,7	35	0,6	21,2	35,8	0,6	0,025	13
	23,9	33,5	–	0,6	21,2	35,8	0,6	0,03	12
	26,5	37,4	39,7	1	22,6	41,4	1	0,03	12
	32,4	46,6	–	1,1	23,5	55,5	1	0,035	11
20	24	28,3	–	0,3	22	30	0,3	0,015	15
	25,6	31,4	32,8	0,3	22	35	0,3	0,02	15
	27,3	34,6	–	0,3	22	40	0,3	0,02	15
	27,4	36	36,2	0,6	23,2	38,8	0,6	0,025	14
	27,2	34,8	37,2	0,6	23,2	38,8	0,6	0,025	14
	28,8	38,5	40,6	1	25,6	41,4	1	0,025	13
	28,2	39,6	–	1	25,6	41,4	1	0,025	12
	30,4	41,6	44,8	1,1	27	45	1	0,03	12
	30,2	42,6	–	1,1	27	45	1	0,03	12
	37,1	54,8	–	1,1	29	63	1	0,035	11
22	32,2	42,1	44	1	27,6	44,4	1	0,025	14
	32,2	46,2	–	1,1	29	47	1	0,03	12

Single row deep groove ball bearings
d **25 – 35** mm



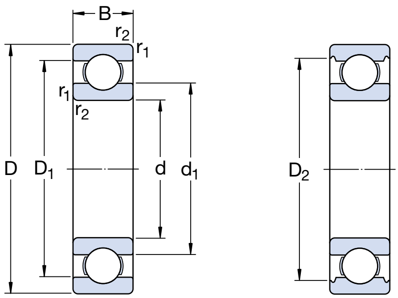
Principal dimensions d	D	B	Basic load ratings dynamic C	static C_0	Fatigue load limit P_u	Speed ratings Reference speed	Limiting speed	Mass	Designation
mm			kN		kN	r/min		kg	–
25	37	7	4,36	2,6	0,125	38 000	24 000	0,022	**61805**
	42	9	7,02	4,3	0,193	36 000	22 000	0,045	**61905**
	47	8	8,06	4,75	0,212	32 000	20 000	0,060	* **16005**
	47	12	11,9	6,55	0,275	32 000	20 000	0,080	* **6005**
	52	9	10,6	6,55	0,28	28 000	18 000	0,078	**98205**
	52	15	14,8	7,8	0,335	28 000	18 000	0,13	* **6205**
	52	15	17,8	9,8	0,40	28 000	18 000	0,12	**6205 ETN9**
	62	17	23,4	11,6	0,49	24 000	16 000	0,23	* **6305**
	62	17	26	13,4	0,57	24 000	16 000	0,21	**6305 ETN9**
	80	21	35,8	19,3	0,82	20 000	13 000	0,53	**6405**
28	58	16	16,8	9,5	0,405	26 000	16 000	0,18	**62/28**
	68	18	25,1	13,7	0,585	22 000	14 000	0,29	**63/28**
30	42	7	4,49	2,9	0,146	32 000	20 000	0,027	**61806**
	47	9	7,28	4,55	0,212	30 000	19 000	0,051	**61906**
	55	9	11,9	7,35	0,31	28 000	17 000	0,085	* **16006**
	55	13	13,8	8,3	0,355	28 000	17 000	0,12	* **6006**
	62	10	15,9	10,2	0,44	22 000	14 000	0,12	**98206**
	62	16	20,3	11,2	0,48	24 000	15 000	0,20	* **6206**
	62	16	23,4	12,9	0,54	24 000	15 000	0,19	**6206 ETN9**
	72	19	29,6	16	0,67	20 000	13 000	0,35	* **6306**
	72	19	32,5	17,3	0,74	22 000	14 000	0,33	**6306 ETN9**
	90	23	43,6	23,6	1,00	18 000	11 000	0,74	**6406**
35	47	7	4,75	3,2	0,17	28 000	18 000	0,030	**61807**
	55	10	9,56	6,8	0,29	26 000	16 000	0,080	**61907**
	62	9	13	8,15	0,38	24 000	15 000	0,11	* **16007**
	62	14	16,8	10,2	0,44	24 000	15 000	0,16	* **6007**
	72	17	27	15,3	0,66	20 000	13 000	0,29	* **6207**
	72	17	31,2	17,6	0,75	20 000	13 000	0,27	**6207 ETN9**
	80	21	35,1	19	0,82	19 000	12 000	0,46	* **6307**
	100	25	55,3	31	1,29	16 000	10 000	0,95	**6407**

* SKF Explorer bearing



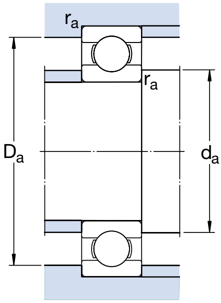
Dimensions					Abutment and fillet dimensions			Calculation factors	
d	d_1 ~	D_1 ~	D_2 ~	$r_{1,2}$ min	d_a min	D_a max	r_a max	k_r	f_0
mm					mm			–	
25	28,5	33,3	–	0,3	27	35	0,3	0,015	14
	30,2	36,8	37,8	0,3	27	40	0,3	0,02	15
	33,3	40,7	–	0,3	27	45	0,3	0,02	15
	32	40	42,2	0,6	28,2	43,8	0,6	0,025	14
	34,5	44	–	0,6	28,2	48,8	0,6	0,025	15
	34,4	44	46,3	1	30,6	46,4	1	0,025	14
	33,1	44,5	–	1	30,6	46,4	1	0,025	13
	36,6	50,4	52,7	1,1	32	55	1	0,03	12
	36,4	51,7	–	1,1	32	55	1	0,03	12
	45,4	62,9	–	1,5	34	71	1,5	0,035	12
28	37	49,2	–	1	33,6	52,4	1	0,025	14
	41,7	56	–	1,1	35	61	1	0,03	13
30	33,7	38,5	–	0,3	32	40	0,3	0,015	14
	35,2	41,8	42,8	0,3	32	45	0,3	0,02	14
	37,7	47,3	–	0,3	32	53	0,3	0,02	15
	38,2	46,8	49	1	34,6	50,4	1	0,025	15
	42,9	54,4	–	0,6	33,2	58,8	0,6	0,025	14
	40,4	51,6	54,1	1	35,6	56,4	1	0,025	14
	39,5	52,9	–	1	35,6	56,4	1	0,025	13
	44,6	59,1	61,9	1,1	37	65	1	0,03	13
	42,5	59,7	–	1,1	37	65	1	0,03	12
	50,3	69,7	–	1,5	41	79	1,5	0,035	12
35	38,7	43,5	–	0,3	37	45	0,3	0,015	14
	41,6	48,4	–	0,6	38,2	51,8	0,6	0,02	14
	44,1	53	–	0,3	37	60	0,3	0,02	14
	43,8	53,3	55,6	1	39,6	57,4	1	0,025	15
	46,9	60	62,7	1,1	42	65	1	0,025	14
	46,1	61,7	–	1,1	42	65	1	0,025	13
	49,6	65,4	69,2	1,5	44	71	1,5	0,03	13
	57,4	79,5	–	1,5	46	89	1,5	0,035	12

Single row deep groove ball bearings
d **40 – 60** mm



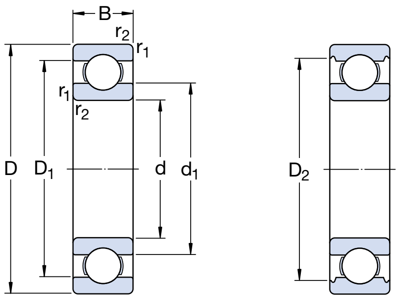
Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	B	C	C_0	P_u				
mm			kN		kN	r/min		kg	–
40	52	7	4,94	3,45	0,19	26 000	16 000	0,034	**61808**
	62	12	13,8	10	0,43	24 000	14 000	0,12	**61908**
	68	9	13,8	9,15	0,44	22 000	14 000	0,13	* **16008**
	68	15	17,8	11,6	0,49	22 000	14 000	0,19	* **6008**
	80	18	32,5	19	0,80	18 000	11 000	0,37	* **6208**
	80	18	35,8	20,8	0,88	18 000	11 000	0,34	**6208 ETN9**
	90	23	42,3	24	1,02	17 000	11 000	0,63	* **6308**
	110	27	63,7	36,5	1,53	14 000	9 000	1,25	**6408**
45	58	7	6,63	6,1	0,26	22 000	14 000	0,040	**61809**
	68	12	14	10,8	0,47	20 000	13 000	0,14	**61909**
	75	10	16,5	10,8	0,52	20 000	12 000	0,17	* **16009**
	75	16	22,1	14,6	0,64	20 000	12 000	0,25	* **6009**
	85	19	35,1	21,6	0,92	17 000	11 000	0,41	* **6209**
	100	25	55,3	31,5	1,34	15 000	9 500	0,83	* **6309**
	120	29	76,1	45	1,90	13 000	8 500	1,55	**6409**
50	65	7	6,76	6,8	0,285	20 000	13 000	0,052	**61810**
	72	12	14,6	11,8	0,50	19 000	12 000	0,14	**61910**
	80	10	16,8	11,4	0,56	18 000	11 000	0,18	* **16010**
	80	16	22,9	16	0,71	18 000	11 000	0,26	* **6010**
	90	20	37,1	23,2	0,98	15 000	10 000	0,46	* **6210**
	110	27	65	38	1,6	13 000	8 500	1,05	* **6310**
	130	31	87,1	52	2,2	12 000	7 500	1,9	**6410**
55	72	9	9,04	8,8	0,38	19 000	12 000	0,083	**61811**
	80	13	16,5	14	0,60	17 000	11 000	0,19	**61911**
	90	11	20,3	14	0,70	16 000	10 000	0,26	* **16011**
	90	18	29,6	21,2	0,90	16 000	10 000	0,39	* **6011**
	100	21	46,2	29	1,25	14 000	9 000	0,61	* **6211**
	120	29	74,1	45	1,90	12 000	8 000	1,35	* **6311**
	140	33	99,5	62	2,60	11 000	7 000	2,3	**6411**
60	78	10	11,9	11,4	0,49	17 000	11 000	0,11	**61812**
	85	13	16,5	14,3	0,60	16 000	10 000	0,20	**61912**
	95	11	20,8	15	0,74	15 000	9 500	0,28	* **16012**
	95	18	30,7	23,2	0,98	15 000	9 500	0,42	* **6012**
	110	22	55,3	36	1,53	13 000	8 000	0,78	* **6212**
	130	31	85,2	52	2,20	11 000	7 000	1,7	* **6312**
	150	35	108	69,5	2,90	10 000	6 300	2,75	**6412**

* SKF Explorer bearing



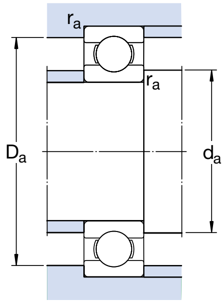
Dimensions					Abutment and fillet dimensions			Calculation factors	
d	d_1 ~	D_1 ~	D_2 ~	$r_{1,2}$ min	d_a min	D_a max	r_a max	k_r	f_0
mm					mm			–	
40	43,7	48,5	–	0,3	42	50	0,3	0,015	14
	46,9	55,1	–	0,6	43,2	58,8	0,6	0,02	16
	49,4	58,6	–	0,3	42	66	0,3	0,02	14
	49,3	58,8	61,1	1	44,6	63,4	1	0,025	15
	52,6	67,4	69,8	1,1	47	73	1	0,025	14
	52	68,8	–	1,1	47	73	1	0,025	13
	56,1	73,8	77,7	1,5	49	81	1,5	0,03	13
	62,8	87	–	2	53	97	2	0,035	12
45	49,1	53,9	–	0,3	47	56	0,3	0,015	17
	52,4	60,6	–	0,6	48,2	64,8	0,6	0,02	16
	55	65,4	–	0,6	48,2	71,8	0,6	0,02	14
	54,8	65,3	67,8	1	50,8	69,2	1	0,025	15
	57,6	72,4	75,2	1,1	52	78	1	0,025	14
	62,2	82,7	86,7	1,5	54	91	1,5	0,03	13
	68,9	95,8	–	2	58	107	2	0,035	12
50	55,1	59,9	–	0,3	52	63	0,3	0,015	17
	56,9	65,1	–	0,6	53,2	68,8	0,6	0,02	16
	60	70	–	0,6	53,2	76,8	0,6	0,02	14
	59,8	70,3	72,8	1	54,6	75,4	1	0,025	15
	62,5	77,4	81,6	1,1	57	83	1	0,025	14
	68,8	91,1	95,2	2	59	101	2	0,03	13
	75,5	104	–	2,1	64	116	2	0,035	12
55	60,6	66,4	–	0,3	57	70	0,3	0,015	17
	63,2	71,8	–	1	59,6	75,4	1	0,02	16
	67	78,1	–	0,6	58,2	86,8	0,6	0,02	15
	66,3	78,7	81,5	1,1	61	84	1	0,025	15
	69,1	85,8	89,4	1,5	64	91	1,5	0,025	14
	75,3	99,5	104	2	66	109	2	0,03	13
	81,6	113	–	2,1	69	126	2	0,035	12
60	65,6	72,4	–	0,3	62	76	0,3	0,015	17
	68,2	76,8	–	1	64,6	80,4	1	0,02	16
	72	83	–	0,6	63,2	91,8	0,6	0,02	14
	71,3	83,7	86,5	1,1	66	89	1	0,025	16
	75,5	94,6	98	1,5	69	101	1,5	0,025	14
	81,9	108	112	2,1	72	118	2	0,03	13
	88,1	122	–	2,1	74	136	2	0,035	12

Single row deep groove ball bearings
d **65 – 85** mm



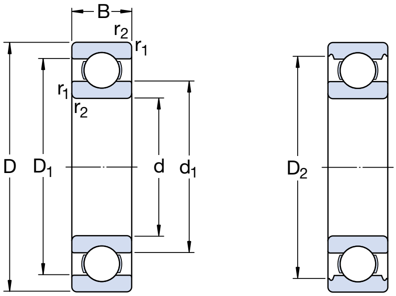
Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	B	C	C_0	P_u				
mm			kN		kN	r/min		kg	–
65	85	10	12,4	12,7	0,54	16 000	10 000	0,13	**61813**
	90	13	17,4	16	0,68	15 000	9 500	0,22	**61913**
	100	11	22,5	16,6	0,83	14 000	9 000	0,30	* **16013**
	100	18	31,9	25	1,06	14 000	9 000	0,44	* **6013**
	120	23	58,5	40,5	1,73	12 000	7 500	0,99	* **6213**
	140	33	97,5	60	2,5	10 000	6 700	2,10	* **6313**
	160	37	119	78	3,15	9 500	6 000	3,30	**6413**
70	90	10	12,4	13,2	0,56	15 000	9 000	0,14	**61814**
	100	16	23,8	21,2	0,9	14 000	8 500	0,35	**61914**
	110	13	29,1	25	1,06	13 000	8 000	0,43	* **16014**
	110	20	39,7	31	1,32	13 000	8 000	0,60	* **6014**
	125	24	63,7	45	1,9	11 000	7 000	1,05	* **6214**
	150	35	111	68	2,75	9 500	6 300	2,50	* **6314**
	180	42	143	104	3,9	8 500	5 300	4,85	**6414**
75	95	10	12,7	14,3	0,61	14 000	8 500	0,15	**61815**
	105	16	24,2	19,3	0,965	13 000	8 000	0,37	**61915**
	110	12	28,6	27	1,14	13 000	8 000	0,38	**16115**
	115	13	30,2	27	1,14	12 000	7 500	0,46	* **16015**
	115	20	41,6	33,5	1,43	12 000	7 500	0,64	* **6015**
	130	25	68,9	49	2,04	10 000	6 700	1,20	* **6215**
	160	37	119	76,5	3	9 000	5 600	3,00	* **6315**
	190	45	153	114	4,15	8 000	5 000	6,80	**6415**
80	100	10	13	15	0,64	13 000	8 000	0,15	**61816**
	110	16	25,1	20,4	1,02	12 000	7 500	0,40	**61916**
	125	14	35,1	31,5	1,32	11 000	7 000	0,60	* **16016**
	125	22	49,4	40	1,66	11 000	7 000	0,85	* **6016**
	140	26	72,8	55	2,2	9 500	6 000	1,40	* **6216**
	170	39	130	86,5	3,25	8 500	5 300	3,60	* **6316**
	200	48	163	125	4,5	7 500	4 800	8,00	**6416**
85	110	13	19,5	20,8	0,88	12 000	7 500	0,27	**61817**
	120	18	31,9	30	1,25	11 000	7 000	0,55	**61917**
	130	14	35,8	33,5	1,37	11 000	6 700	0,63	* **16017**
	130	22	52	43	1,76	11 000	6 700	0,89	* **6017**
	150	28	87,1	64	2,5	9 000	5 600	1,80	* **6217**
	180	41	140	96,5	3,55	8 000	5 000	4,25	* **6317**
	210	52	174	137	4,75	7 000	4 500	9,50	**6417**

* SKF Explorer bearing



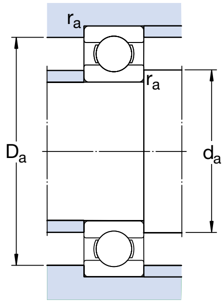
Dimensions					Abutment and fillet dimensions			Calculation factors	
d	d_1 ~	D_1 ~	D_2 ~	$r_{1,2}$ min	d_a min	D_a max	r_a max	k_r	f_0
mm					mm			–	
65	71,6	78,4	–	0,6	68,2	81,8	0,6	0,015	17
	73,2	81,8	–	1	69,6	85,4	1	0,02	17
	76,5	88,4	–	0,6	68,2	96,8	0,6	0,02	16
	76,3	88,7	91,5	1,1	71	94	1	0,025	16
	83,3	102	106	1,5	74	111	1,5	0,025	15
	88,4	116	121	2,1	77	128	2	0,03	13
	94	131	–	2,1	79	146	2	0,035	12
70	76,6	83,4	–	0,6	73,2	86,8	0,6	0,015	17
	79,7	90,3	–	1	74,6	95,4	1	0,02	16
	83,3	96,8	–	0,6	73,2	106	0,6	0,02	16
	82,9	97,2	99,9	1,1	76	104	1	0,025	16
	87,1	108	111	1,5	79	116	1,5	0,025	15
	95	125	130	2,1	82	138	2	0,03	13
	104	146	–	3	86	164	2,5	0,035	12
75	81,6	88,4	–	0,6	78,2	91,8	0,6	0,015	17
	84,7	95,3	–	1	79,6	100	1	0,02	14
	88,3	102	–	0,6	77	108	0,3	0,02	16
	88,3	102	–	0,6	78,2	111	0,6	0,02	16
	87,9	102	105	1,1	81	109	1	0,025	16
	92,1	113	117	1,5	84	121	1,5	0,025	15
	101	133	138	2,1	87	148	2	0,03	13
	110	154	–	3	91	174	2,5	0,035	12
80	86,6	93,4	–	0,6	83,2	96,8	0,6	0,015	17
	89,8	100	102	1	84,6	105	1	0,02	14
	95,3	110	–	0,6	83,2	121	0,6	0,02	16
	94,4	111	114	1,1	86	119	1	0,025	16
	101	122	127	2	91	129	2	0,025	15
	108	142	147	2,1	92	158	2	0,03	13
	117	163	–	3	96	184	2,5	0,035	12
85	93,2	102	–	1	89,6	105	1	0,015	17
	96,4	109	–	1,1	91	114	1	0,02	16
	100	115	–	0,6	88,2	126	0,6	0,02	16
	99,4	116	119	1,1	92	123	1	0,025	16
	106	130	134	2	94	141	2	0,025	15
	115	151	155	3	99	166	2,5	0,03	13
	123	171	–	4	105	190	3	0,035	12

Single row deep groove ball bearings
d **90 – 110** mm



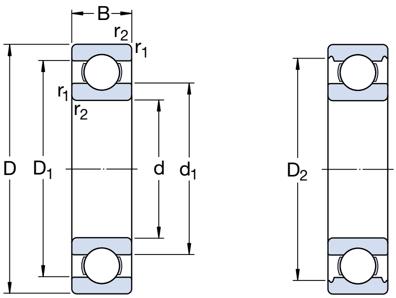
Principal dimensions			Basic load ratings		Fatigue load limit	Speed ratings		Mass	Designation
			dynamic	static		Reference speed	Limiting speed		
d	D	B	C	C_0	P_u				
mm			kN		kN	r/min		kg	–
90	115	13	19,5	22	0,915	11 000	7 000	0,28	**61818**
	125	18	33,2	31,5	1,23	11 000	6 700	0,59	**61918**
	140	16	43,6	39	1,56	10 000	6 300	0,85	* **16018**
	140	24	60,5	50	1,96	10 000	6 300	1,15	* **6018**
	160	30	101	73,5	2,8	8 500	5 300	2,15	* **6218**
	190	43	151	108	3,8	7 500	4 800	4,90	* **6318**
	225	54	186	150	5	6 700	4 300	11,5	**6418**
95	120	13	19,9	22,8	0,93	11 000	6 700	0,30	**61819**
	130	18	33,8	33,5	1,43	10 000	6 300	0,61	**61919**
	145	16	44,8	41,5	1,63	9 500	6 000	0,89	* **16019**
	145	24	63,7	54	2,08	9 500	6 000	1,20	* **6019**
	170	32	114	81,5	3	8 000	5 000	2,60	* **6219**
	200	45	159	118	4,15	7 000	4 500	5,65	* **6319**
100	125	13	19,9	24	0,95	10 000	6 300	0,31	**61820**
	140	20	42,3	41	1,63	9 500	6 000	0,83	**61920**
	150	16	46,2	44	1,73	9 500	5 600	0,91	* **16020**
	150	24	63,7	54	2,04	9 500	5 600	1,25	* **6020**
	180	34	127	93	3,35	7 500	4 800	3,15	* **6220**
	215	47	174	140	4,75	6 700	4 300	7,00	**6320**
105	130	13	20,8	19,6	1	10 000	6 300	0,32	**61821**
	145	20	44,2	44	1,7	9 500	5 600	0,87	**61921**
	160	18	54	51	1,86	8 500	5 300	1,20	* **16021**
	160	26	76,1	65,5	2,4	8 500	5 300	1,60	* **6021**
	190	36	140	104	3,65	7 000	4 500	3,70	* **6221**
	225	49	182	153	5,1	6 300	4 000	8,25	**6321**
110	140	16	28,1	26	1,25	9 500	5 600	0,60	**61822**
	150	20	43,6	45	1,66	9 000	5 600	0,90	**61922**
	170	19	60,2	57	2,04	8 000	5 000	1,45	* **16022**
	170	28	85,2	73,5	2,4	8 000	5 000	1,95	* **6022**
	200	38	151	118	4	6 700	4 300	4,35	* **6222**
	240	50	203	180	5,7	6 000	3 800	9,55	**6322**

* SKF Explorer bearing



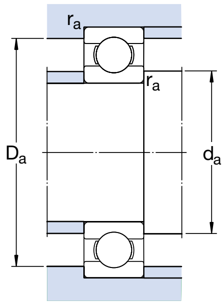
Dimensions					**Abutment and fillet dimensions**			**Calculation factors**	
d	d_1 ~	D_1 ~	D_2 ~	$r_{1,2}$ min	d_a min	D_a max	r_a max	k_r	f_0
mm					mm			–	
90	98,2	107	–	1	94,6	110	1	0,015	17
	101	114	117	1,1	96	119	1	0,02	16
	107	123	–	1	94,6	135	1	0,02	16
	106	124	128	1,5	97	133	1,5	0,025	16
	113	138	143	2	101	149	2	0,025	15
	121	159	164	3	104	176	2,5	0,03	13
	132	181	–	4	110	205	3	0,035	12
95	103	112	–	1	99,6	115	1	0,015	17
	106	119	122	1,1	101	124	1	0,02	17
	112	128	–	1	99,6	140	1	0,02	16
	111	129	133	1,5	102	138	1,5	0,025	16
	118	146	151	2,1	106	159	2	0,025	14
	128	167	172	3	109	186	2,5	0,03	13
100	108	117	–	1	105	120	1	0,015	17
	113	127	–	1,1	106	134	1	0,02	16
	116	134	–	1	105	145	1	0,02	17
	116	134	138	1,5	107	143	1,5	0,025	16
	125	155	160	2,1	111	169	2	0,025	14
	136	179	184	3	114	201	2,5	0,03	13
105	112	123	–	1	110	125	1	0,015	13
	118	132	–	1,1	111	139	1	0,02	17
	123	142	–	1	110	155	1	0,02	16
	123	143	147	2	116	149	2	0,025	16
	131	163	167	2,1	117	178	2	0,025	14
	142	188	–	3	119	211	2,5	0,03	13
110	119	131	–	1	115	135	1	0,015	14
	123	137	–	1,1	116	144	1	0,02	17
	130	150	–	1	115	165	1	0,02	16
	129	151	155	2	119	161	2	0,025	16
	138	172	177	2,1	122	188	2	0,025	14
	150	200	–	3	124	226	2,5	0,03	13

Single row deep groove ball bearings
d **120 – 170** mm



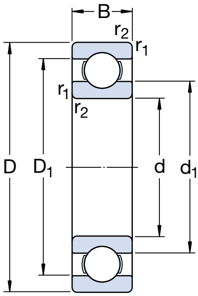
Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	B	C	C_0	P_u				
mm			kN		kN	r/min		kg	–
120	150	16	29,1	28	1,29	8 500	5 300	0,65	**61824**
	165	22	55,3	57	2,04	8 000	5 000	1,20	**61924**
	180	19	63,7	64	2,2	7 500	4 800	1,60	* **16024**
	180	28	88,4	80	2,75	7 500	4 800	2,05	* **6024**
	215	40	146	118	3,9	6 300	4 000	5,15	**6224**
	260	55	208	186	5,7	5 600	3 400	12,5	**6324**
130	165	18	37,7	43	1,6	8 000	4 800	0,93	**61826**
	180	24	65	67	2,28	7 500	4 500	1,85	**61926**
	200	22	83,2	81,5	2,7	7 000	4 300	2,35	* **16026**
	200	33	112	100	3,35	7 000	4 300	3,15	* **6026**
	230	40	156	132	4,15	5 600	3 600	5,80	**6226**
	280	58	229	216	6,3	5 000	3 200	17,5	**6326 M**
140	175	18	39	46,5	1,66	7 500	4 500	0,99	**61828**
	190	24	66,3	72	2,36	7 000	5 600	1,70	**61928 MA**
	210	22	80,6	86,5	2,8	6 700	4 000	2,50	**16028**
	210	33	111	108	3,45	6 700	4 000	3,35	**6028**
	250	42	165	150	4,55	5 300	3 400	7,45	**6228**
	300	62	251	245	7,1	4 800	4 300	22,0	**6328 M**
150	190	20	48,8	61	1,96	6 700	4 300	1,40	**61830**
	210	28	88,4	93	2,9	6 300	5 300	3,05	**61930 MA**
	225	24	92,2	98	3,05	6 000	3 800	3,15	**16030**
	225	35	125	125	3,9	6 000	3 800	4,80	**6030**
	270	45	174	166	4,9	5 000	3 200	9,40	**6230**
	320	65	276	285	7,8	4 300	4 000	26,0	**6330 M**
160	200	20	49,4	64	2	6 300	4 000	1,45	**61832**
	220	28	92,3	98	3,05	6 000	5 000	3,25	**61932 MA**
	240	25	99,5	108	3,25	5 600	3 600	3,70	**16032**
	240	38	143	143	4,3	5 600	3 600	5,90	**6032**
	290	48	186	186	5,3	4 500	3 000	14,5	**6232**
	340	68	276	285	7,65	4 000	3 800	29,0	**6332 M**
170	215	22	61,8	78	2,4	6 000	3 600	1,90	**61834**
	230	28	93,6	106	3,15	5 600	4 800	3,40	**61934 MA**
	260	28	119	129	3,75	5 300	3 200	5,00	**16034**
	260	42	168	173	5	5 300	4 300	7,90	**6034 M**
	310	52	212	224	6,1	4 300	3 800	17,5	**6234 M**
	360	72	312	340	8,8	3 800	3 400	34,5	**6334 M**

* SKF Explorer bearing

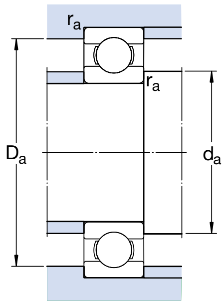


Dimensions					Abutment and fillet dimensions			Calculation factors	
d	d_1 ~	D_1 ~	D_2 ~	$r_{1,2}$ min	d_a min	D_a max	r_a max	k_r	f_0
mm					mm			–	
120	129	141	–	1	125	145	1	0,015	13
	134	151	–	1,1	126	159	1	0,02	17
	139	161	–	1	125	175	1	0,02	17
	139	161	165	2	129	171	2	0,025	16
	151	184	189	2,1	132	203	2	0,025	14
	165	215	–	3	134	246	2,5	0,03	14
130	140	155	–	1,1	136	159	1	0,015	16
	146	164	–	1,5	137	173	1,5	0,02	16
	154	176	–	1,1	136	192	1	0,02	16
	153	177	182	2	139	191	2	0,025	16
	161	198	–	3	144	216	2,5	0,025	15
	178	232	–	4	147	263	3	0,03	14
140	151	164	–	1,1	146	169	1	0,015	16
	156	175	–	1,5	147	183	1,5	0,02	17
	164	186	–	1,1	146	204	1	0,02	17
	163	187	192	2	149	201	2	0,025	16
	176	213	213	3	154	236	2,5	0,025	15
	191	248	248	4	157	283	3	0,03	14
150	163	177	–	1,1	156	184	1	0,015	17
	169	191	–	2	159	201	2	0,02	16
	175	199	–	1,1	156	219	1	0,02	16
	174	201	205	2,1	160	215	2	0,025	16
	191	227	–	3	164	256	2,5	0,025	15
	206	263	–	4	167	303	3	0,03	14
160	173	187	–	1,1	166	194	1	0,015	17
	179	201	–	2	169	211	2	0,02	16
	186	213	–	1,5	167	233	1,5	0,02	17
	186	214	–	2,1	169	231	2	0,025	16
	206	242	–	3	174	276	2,5	0,025	15
	219	281	–	4	177	323	3	0,03	14
170	184	201	–	1,1	176	209	1	0,015	17
	189	211	–	2	179	221	2	0,02	17
	200	229	–	1,5	177	253	1,5	0,02	16
	199	231	–	2,1	180	250	2	0,025	16
	219	259	–	4	187	293	3	0,025	15
	231	298	–	4	187	343	3	0,03	14

Single row deep groove ball bearings
d **180 – 260** mm

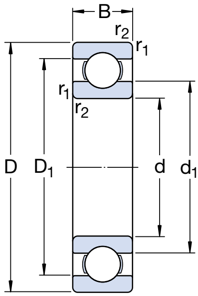


Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	B	C	C_0	P_u				
mm			kN		kN	r/min		kg	–
180	225	22	62,4	81,5	2,45	5 600	3 400	2,00	**61836**
	250	33	119	134	3,9	5 300	4 300	5,05	**61936 MA**
	280	31	138	146	4,15	4 800	4 000	6,60	**16036**
	280	46	190	200	5,6	4 800	4 000	10,5	**6036 M**
	320	52	229	240	6,4	4 000	3 600	18,5	**6236 M**
	380	75	351	405	10,4	3 600	3 200	42,5	**6336 M**
190	240	24	76,1	98	2,8	5 300	3 200	2,60	**61838**
	260	33	117	134	3,8	5 000	4 300	5,25	**61938 MA**
	290	31	148	166	4,55	4 800	3 000	7,90	**16038**
	290	46	195	216	5,85	4 800	3 800	11,0	**6038 M**
	340	55	255	280	7,35	3 800	3 400	23,0	**6238 M**
	400	78	371	430	10,8	3 400	3 000	49,0	**6338 M**
200	250	24	76,1	102	2,9	5 000	3 200	2,70	**61840**
	280	38	148	166	4,55	4 800	3 800	7,40	**61940 MA**
	310	34	168	190	5,1	4 300	2 800	8,85	**16040**
	310	51	216	245	6,4	4 300	3 600	14,0	**6040 M**
	360	58	270	310	7,8	3 600	3 200	28,0	**6240 M**
220	270	24	78	110	3	4 500	2 800	3,00	**61844**
	300	38	151	180	4,75	4 300	3 600	8,00	**61944 MA**
	340	37	174	204	5,2	4 000	2 400	11,5	**16044**
	340	56	247	290	7,35	4 000	3 200	18,5	**6044 M**
	400	65	296	365	8,8	3 200	3 000	37,0	**6244 M**
	460	88	410	520	12	3 000	2 600	72,5	**6344 M**
240	300	28	108	150	3,8	4 000	2 600	4,50	**61848**
	320	38	159	200	5,1	4 000	3 200	8,60	**61948 MA**
	360	37	178	220	5,3	3 600	3 000	14,5	**16048 MA**
	360	56	255	315	7,8	3 600	3 000	19,5	**6048 M**
	440	72	358	465	10,8	3 000	2 600	51,0	**6248 M**
	500	95	442	585	12,9	2 600	2 400	92,5	**6348 M**
260	320	28	111	163	4	3 800	2 400	4,80	**61852**
	360	46	212	270	6,55	3 600	3 000	14,5	**61952 MA**
	400	44	238	310	7,2	3 200	2 800	21,5	**16052 MA**
	400	65	291	375	8,8	3 200	2 800	29,5	**6052 M**
	480	80	390	530	11,8	2 600	2 400	65,5	**6252 M**

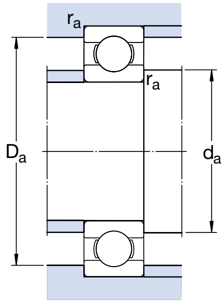


Dimensions				Abutment and fillet dimensions			Calculation factors	
d	d_1 ~	D_1 ~	$r_{1,2}$ min	d_a min	D_a max	r_a max	k_r	f_0
mm				mm			–	
180	194	211	1,1	186	219	1	0,015	17
	203	227	2	189	241	2	0,02	16
	214	246	2	189	271	2	0,02	16
	212	248	2,1	190	270	2	0,025	16
	227	273	4	197	303	3	0,025	15
	245	314	4	197	363	3	0,03	14
190	206	224	1,5	197	233	1,5	0,015	17
	213	237	2	199	251	2	0,02	17
	224	255	2	199	281	2	0,02	16
	222	258	2,1	200	280	2	0,025	16
	240	290	4	207	323	3	0,025	15
	259	331	5	210	380	4	0,03	14
200	216	234	1,5	207	243	1,5	0,015	17
	226	254	2,1	210	270	2	0,02	16
	237	272	2	209	301	2	0,02	16
	235	275	2,1	210	300	2	0,025	16
	255	302	4	217	343	3	0,025	15
220	236	254	1,5	227	263	1,5	0,015	17
	246	274	2,1	230	290	2	0,02	17
	262	298	2,1	230	330	2	0,02	16
	258	302	3	233	327	2,5	0,025	16
	283	335	4	237	383	3	0,025	15
	300	381	5	240	440	4	0,03	14
240	259	281	2	249	291	2	0,015	17
	266	294	2,1	250	310	2	0,02	17
	280	320	2,1	250	350	2	0,02	17
	278	322	3	253	347	2,5	0,025	16
	308	373	4	257	423	3	0,025	15
	330	411	5	260	480	4	0,03	15
260	279	301	2	269	311	2	0,015	17
	292	328	2,1	270	350	2	0,02	16
	307	352	3	273	387	2,5	0,02	16
	305	355	4	277	383	3	0,025	16
	336	405	5	280	460	4	0,025	15

Single row deep groove ball bearings
d **280 – 420** mm

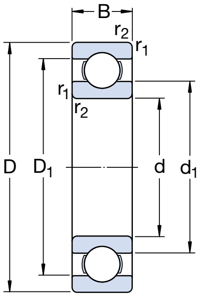


Principal dimensions			Basic load ratings		Fatigue load limit	Speed ratings		Mass	Designation
			dynamic	static		Reference speed	Limiting speed		
d	D	B	C	C_0	P_u				
mm			kN		kN	r/min		kg	–
280	350	33	138	200	4,75	3 400	2 200	7,40	**61856**
	380	46	216	285	6,7	3 200	2 800	15,0	**61956 MA**
	420	44	242	335	7,5	3 000	2 600	23,0	**16056 MA**
	420	65	302	405	9,3	3 000	2 600	31,0	**6056 M**
	500	80	423	600	12,9	2 600	2 200	71,0	**6256 M**
300	380	38	172	245	5,6	3 200	2 600	10,5	**61860 MA**
	420	56	270	375	8,3	3 000	2 400	24,5	**61960 MA**
	460	50	286	405	8,8	2 800	2 400	32,0	**16060 MA**
	460	74	358	500	10,8	2 800	2 400	44,0	**6060 M**
	540	85	462	670	13,7	2 400	2 000	88,5	**6260 M**
320	400	38	172	255	5,7	3 000	2 400	11,0	**61864 MA**
	440	56	276	400	8,65	2 800	2 400	25,5	**61964 MA**
	480	50	281	405	8,65	2 600	2 200	34,0	**16064 MA**
	480	74	371	540	11,4	2 600	2 200	46,0	**6064 M**
340	420	38	178	275	6	2 800	2 400	11,5	**61868 MA**
	460	56	281	425	9	2 600	2 200	26,5	**61968 MA**
	520	57	345	520	10,6	2 400	2 000	45,0	**16068 MA**
	520	82	423	640	13,2	2 400	2 000	62,0	**6068 M**
360	440	38	182	285	6,1	2 600	2 200	12,0	**61872 MA**
	480	56	291	450	9,15	2 600	2 000	28,0	**61972 MA**
	540	57	351	550	11	2 400	1 900	49,0	**16072 MA**
	540	82	462	735	15	2 400	1 900	64,5	**6072 M**
380	480	46	242	390	8	2 400	2 000	20,0	**61876 MA**
	520	65	338	540	10,8	2 400	1 900	40,0	**61976 MA**
	560	57	377	620	12,2	2 200	1 800	51,0	**16076 MA**
	560	82	462	750	14,6	2 200	1 800	67,5	**6076 M**
400	500	46	247	405	8,15	2 400	1 900	20,5	**61880 MA**
	540	65	345	570	11,2	2 200	1 800	41,5	**61980 MA**
	600	90	520	865	16,3	2 000	1 700	87,5	**6080 M**
420	520	46	251	425	8,3	2 200	1 800	21,5	**61884 MA**
	560	65	351	600	11,4	2 200	1 800	43,0	**61984 MA**
	620	90	507	880	16,3	2 000	1 600	91,5	**6084 M**

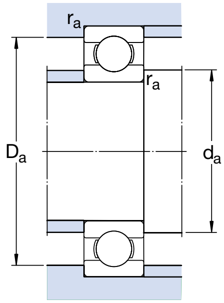


Dimensions				Abutment and fillet dimensions			Calculation factors	
d	d_1 ~	D_1 ~	$r_{1,2}$ min	d_a min	D_a max	r_a max	k_r	f_0
mm				mm			–	
280	302	327	2	289	341	2	0,015	17
	312	348	2,1	291	369	2	0,02	17
	326	374	3	293	407	2,5	0,02	17
	325	375	4	296	404	3	0,025	16
	353	427	5	300	480	4	0,025	15
300	326	354	2,1	309	371	2	0,015	17
	338	382	3	313	407	2,5	0,02	16
	352	408	4	315	445	3	0,02	16
	350	410	4	315	445	3	0,025	16
	381	459	5	320	520	4	0,025	15
320	346	374	2,1	332	388	2	0,015	17
	358	402	3	333	427	2,5	0,02	16
	372	428	4	335	465	3	0,02	17
	370	431	4	335	465	3	0,025	16
340	366	394	2,1	352	408	2	0,015	17
	378	423	3	353	447	2,5	0,02	17
	398	462	4	355	505	3	0,02	16
	396	462	5	360	500	4	0,025	16
360	385	416	2,1	372	428	2	0,015	17
	398	442	3	373	467	2,5	0,02	17
	418	482	4	375	525	3	0,02	16
	416	485	5	378	522	4	0,025	16
380	412	449	2,1	392	468	2	0,015	17
	425	475	4	395	505	3	0,02	17
	438	502	4	395	545	3	0,02	17
	436	502	5	398	542	4	0,025	16
400	432	471	2,1	412	488	2	0,015	17
	445	495	4	415	525	3	0,02	17
	462	536	5	418	582	4	0,025	16
420	452	491	2,1	432	508	2	0,015	17
	465	515	4	435	545	3	0,02	17
	482	558	5	438	602	4	0,025	16

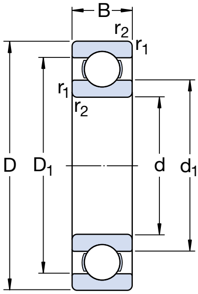
Single row deep groove ball bearings
d **440 – 710** mm



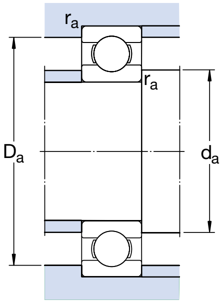
Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	B	C	C_0	P_u				
mm			kN		kN	r/min		kg	–
440	540	46	255	440	8,5	2 200	1 800	22,5	**61888 MA**
	600	74	410	720	13,2	2 000	1 600	60,5	**61988 MA**
	650	94	553	965	17,6	1 900	1 500	105	**6088 M**
460	580	56	319	570	10,6	2 000	1 600	35,0	**61892 MA**
	620	72	423	750	13,7	1 900	1 600	62,5	**61992 MA**
	680	100	582	1 060	19	1 800	1 500	120	**6092 MB**
480	600	56	325	600	10,8	1 900	1 600	36,5	**61896 MA**
	650	78	449	815	14,6	1 800	1 500	74,0	**61996 MA**
	700	100	618	1 140	20	1 700	1 400	125	**6096 MB**
500	620	56	332	620	11,2	1 800	1 500	40,5	**618/500 MA**
	670	78	462	865	15	1 700	1 400	77,0	**619/500 MA**
	720	100	605	1 140	19,6	1 600	1 300	135	**60/500 N1MAS**
530	650	56	332	655	11,2	1 700	1 400	39,5	**618/530 MA**
	710	82	488	930	15,6	1 600	1 300	90,5	**619/530 MA**
	780	112	650	1 270	20,8	1 500	1 200	185	**60/530 N1MAS**
560	680	56	345	695	11,8	1 600	1 300	42,0	**618/560 MA**
	750	85	494	980	16,3	1 500	1 200	105	**619/560 MA**
	820	115	663	1 470	22	1 400	1 200	210	**60/560 N1MAS**
600	730	60	364	765	12,5	1 500	1 200	52,0	**618/600 MA**
	800	90	585	1 220	19,6	1 400	1 100	125	**619/600 MA**
630	780	69	442	965	15,3	1 400	1 100	73,0	**618/630 MA**
	850	100	624	1 340	21,2	1 300	1 100	160	**619/630 N1MA**
	920	128	819	1 760	27	1 200	1 000	285	**60/630 N1MBS**
670	820	69	442	1 000	15,6	1 300	1 100	83,5	**618/670 MA**
	900	103	676	1 500	22,4	1 200	1 000	185	**619/670 MA**
	980	136	904	2 040	30	1 100	900	345	**60/670 N1MAS**
710	870	74	475	1 100	16,6	1 200	1 000	93,5	**618/710 MA**
	950	106	663	1 500	22	1 100	900	220	**619/710 MA**
	1 030	140	956	2 200	31,5	1 000	850	375	**60/710 MA**



Dimensions				Abutment and fillet dimensions			Calculation factors	
d	d_1 ~	D_1 ~	$r_{1,2}$ min	d_a min	D_a max	r_a max	k_r	f_0
mm				mm			–	
440	472	510	2,1	452	528	2	0,015	17
	492	548	4	455	585	3	0,02	17
	505	586	6	463	627	5	0,025	16
460	498	542	3	473	567	2,5	0,015	17
	512	568	4	476	604	3	0,02	17
	528	614	6	483	657	5	0,025	16
480	518	564	3	493	587	2,5	0,015	17
	535	595	5	498	632	4	0,02	17
	548	630	6	503	677	5	0,025	16
500	538	582	3	513	607	2,5	0,015	17
	555	615	5	518	652	4	0,02	17
	568	650	6	523	697	5	0,025	16
530	568	614	3	543	637	2,5	0,015	17
	587	653	5	548	692	4	0,02	17
	613	697	6	553	757	5	0,025	16
560	598	644	3	573	667	2,5	0,015	17
	622	688	5	578	732	4	0,02	17
	648	732	6	583	797	5	0,025	16
600	642	688	3	613	717	2,5	0,015	17
	664	736	5	618	782	4	0,02	17
630	678	732	4	645	765	3	0,015	17
	702	778	6	653	827	5	0,02	17
	725	825	7,5	658	892	6	0,025	16
670	718	772	4	685	805	3	0,015	17
	745	825	6	693	877	5	0,02	17
	772	878	7,5	698	952	6	0,025	16
710	671	819	4	725	855	3	0,015	17
	790	870	6	733	927	5	0,02	17
	813	927	7,5	738	1002	6	0,025	16

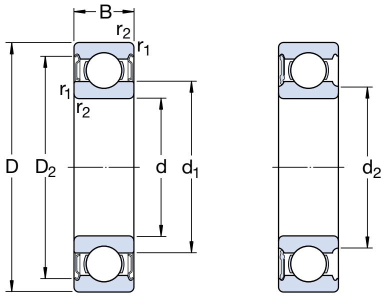


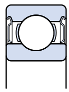
Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	B	C	C_0	P_u				
mm			kN		kN	r/min		kg	–
750	920	78	527	1 250	18,3	1 100	900	110	**618/750 MA**
	1 000	112	761	1 800	25,5	1 000	850	255	**619/750 MA**
800	980	82	559	1 370	19,3	1 000	850	130	**618/800 MA**
	1 060	115	832	2 040	28,5	950	800	275	**619/800 MA**
	1 150	155	1 010	2 550	34,5	900	750	535	**60/800 N1MAS**
850	1 030	82	559	1 430	19,6	950	750	140	**618/850 MA**
900	1 090	85	618	1 600	21,6	850	700	160	**618/900 MA**
1 000	1 220	100	637	1 800	22,8	750	600	245	**618/1000 MA**
1 060	1 280	100	728	2 120	26,5	670	560	260	**618/1060 MA**
1 120	1 360	106	741	2 200	26,5	630	530	315	**618/1120 MA**
1 180	1 420	106	761	2 360	27,5	560	480	330	**618/1180 MB**
1 500	1 820	140	1 210	4 400	46,5	380	240	690	**618/1500 TN**

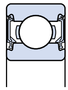


Dimensions				Abutment and fillet dimensions			Calculation factors	
d	d_1 ~	D_1 ~	$r_{1,2}$ min	d_a min	D_a max	r_a max	k_r	f_0
mm				mm			–	
750	804	866	5	768	902	4	0,015	17
	835	915	6	773	977	5	0,02	17
800	857	923	5	818	962	4	0,015	17
	884	976	6	823	1 037	5	0,02	17
	918	1 032	7,5	828	1 122	6	0,025	16
850	907	973	5	868	1 012	4	0,015	17
900	961	1 030	5	918	1 072	4	0,015	17
1 000	1 076	1 145	6	1 023	1 197	5	0,015	17
1 060	1 132	1 209	6	1 083	1 257	5	0,015	17
1 120	1 202	1 278	6	1 143	1 337	5	0,015	17
1 180	1 262	1 339	6	1 203	1 397	5	0,015	17
1 500	1 607	1 714	7,5	1 528	1 792	6	0,015	17

Sealed single row deep groove ball bearings
d **3 – 7** mm




2Z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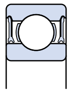
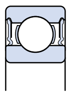
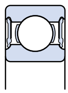
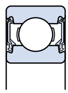

2RZ


2RS1

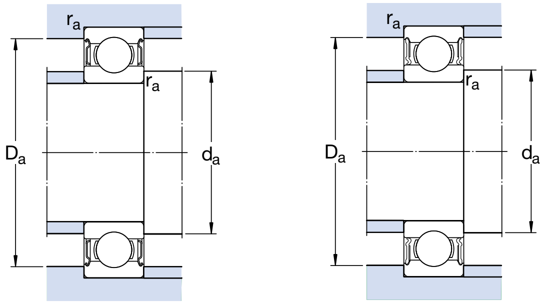

2RS1


2RSH

Principal dimensions			**Basic load ratings** dynamic	static	**Fatigue load limit**	**Speed ratings** Reference speed	Limiting[1] speed	**Mass**	**Designations** sealed both sides	sealed one side
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
3	10	4	0,54	0,18	0,007	130 000	60 000	0,0015	**623-2Z**	**623-Z**
	10	4	0,54	0,18	0,007	–	40 000	0,0015	**623-2RS1**	**623-RS1**
4	9	3,5	0,54	0,18	0,007	140 000	70 000	0,0010	**628/4-2Z**	–
	9	4	0,54	0,18	0,007	140 000	70 000	0,0013	**638/4-2Z**	–
	11	4	0,72	0,23	0,010	130 000	63 000	0,0017	**619/4-2Z**	–
	12	4	0,81	0,28	0,012	120 000	60 000	0,0021	**604-2Z**	**604-Z**
	13	5	0,94	0,29	0,012	110 000	53 000	0,0031	**624-2Z**	**624-Z**
	16	5	1,11	0,38	0,016	95 000	48 000	0,0054	**634-2Z**	**634-Z**
	16	5	1,11	0,38	0,016	95 000	48 000	0,0054	**634-2RZ**	**634-RZ**
	16	5	1,11	0,38	0,016	–	28 000	0,0054	**634-2RS1**	**634-RS1**
5	11	4	0,64	0,26	0,011	120 000	60 000	0,0014	**628/5-2Z**	–
	11	5	0,64	0,26	0,011	120 000	60 000	0,0016	**638/5-2Z**	–
	13	4	0,88	0,34	0,014	110 000	53 000	0,0025	**619/5-2Z**	–
	16	5	1,14	0,38	0,016	95 000	48 000	0,005	*** 625-2Z**	*** 625-Z**
	19	6	2,34	0,95	0,04	80 000	40 000	0,009	*** 635-2Z**	*** 635-Z**
	19	6	2,34	0,95	0,04	80 000	40 000	0,009	*** 635-2RZ**	*** 635-RZ**
	19	6	2,34	0,95	0,04	–	24 000	0,009	*** 635-2RS1**	*** 635-RS1**
6	13	5	0,88	0,35	0,015	110 000	53 000	0,0026	**628/6-2Z**	–
	15	5	1,24	0,48	0,02	100 000	50 000	0,0039	**619/6-2Z**	–
	19	6	2,34	0,95	0,04	80 000	40 000	0,0084	*** 626-2Z**	*** 626-Z**
	19	6	2,34	0,95	0,04	80 000	40 000	0,0084	*** 626-2RSL**	*** 626-RSL**
	19	6	2,34	0,95	0,04	–	24 000	0,0084	*** 626-2RSH**	*** 626-RSH**
7	14	5	0,956	0,4	0,017	100 000	50 000	0,0031	**628/7-2Z**	–
	17	5	1,48	0,56	0,024	90 000	45 000	0,0049	**619/7-2Z**	–
	19	6	2,34	0,95	0,04	85 000	43 000	0,0075	*** 607-2Z**	*** 607-Z**
	19	6	2,34	0,95	0,04	85 000	43 000	0,0075	*** 607-2RSL**	*** 607-RSL**
	19	6	2,34	0,95	0,04	–	24 000	0,0075	*** 607-2RSH**	*** 607-RSH**
	22	7	3,45	1,37	0,057	70 000	36 000	0,013	*** 627-2Z**	*** 627-Z**
	22	7	3,45	1,37	0,057	70 000	36 000	0,012	*** 627-2RSL**	*** 627-RSL**
	22	7	3,45	1,37	0,057	–	22 000	0,012	*** 627-2RSH**	*** 627-RSH**

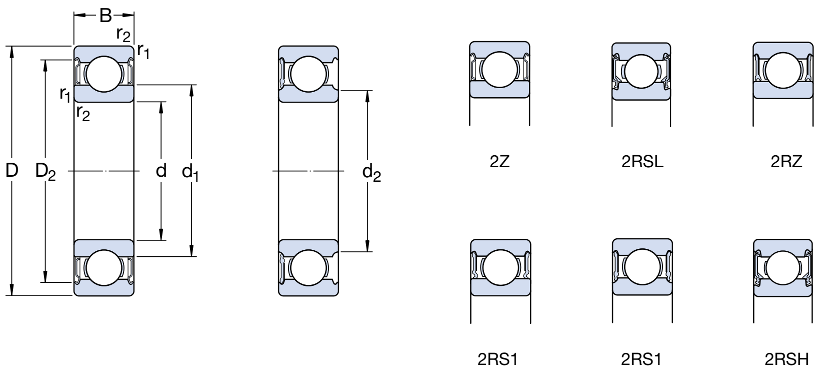
* SKF Explorer bearing
[1] For bearings with only one shield or low-friction seal (Z, RZ, RSL), the limiting speeds for open bearings are valid

SKF



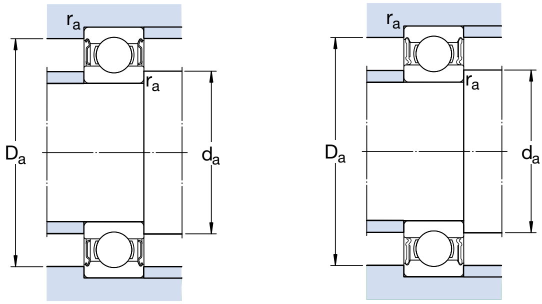
Dimensions					Abutment and fillet dimensions				Calculation factors	
d	d_1 ~	d_2 ~	D_2 ~	$r_{1,2}$ min	d_a min	d_a max	D_a max	r_a max	k_r	f_0
mm					mm				–	
3	5,2	–	8,2	0,15	4,2	–	8,8	0,1	0,025	7,5
	5,2	–	8,2	0,15	4,2	–	8,8	0,1	0,025	7,5
4	5,2	–	7,8	0,1	4,6	–	8,4	0,1	0,015	10
	5,2	–	7,8	0,1	4,6	–	8,4	0,1	0,015	10
	5,9	–	9,8	0,15	4,8	–	10,2	0,1	0,02	9,9
	6,1	–	9,8	0,2	5,4	–	10,6	0,2	0,025	10
	6,7	–	11,2	0,2	5,8	–	11,2	0,2	0,025	7,3
	8,4	–	13,3	0,3	6,4	–	13,6	0,3	0,03	8,4
	8,4	–	13,3	0,3	6,4	–	13,6	0,3	0,03	8,4
	8,4	–	13,3	0,3	6,4	–	13,6	0,3	0,03	8,4
5	6,8	–	9,7	0,15	5,8	–	10,2	0,1	0,015	11
	6,8	–	9,7	0,15	5,8	–	10,2	0,1	0,015	11
	7,6	–	11,4	0,2	6,4	–	11,6	0,2	0,02	11
	8,4	–	13,3	0,3	7,4	–	13,6	0,3	0,025	8,4
	10,7	–	16,5	0,3	7,4	–	16,6	0,3	0,03	13
	10,7	–	16,5	0,3	7,4	–	16,6	0,3	0,03	13
	10,7	–	16,5	0,3	7,4	–	16,6	0,3	0,03	13
6	7,9	–	11,7	0,15	6,8	–	12,2	0,1	0,015	11
	8,6	–	13,3	0,2	7,4	–	13,6	0,2	0,02	10
	11,1	–	16,5	0,3	8,4	–	16,6	0,3	0,025	13
	–	9,5	16,5	0,3	8,4	9,4	16,6	0,3	0,025	13
	–	9,5	16,5	0,3	8,4	9,4	16,6	0,3	0,025	13
7	8,9	–	12,6	0,15	7,8	–	13,2	0,1	0,015	11
	9,8	–	15,2	0,3	9	–	15	0,3	0,02	10
	11,1	–	16,5	0,3	9	–	17	0,3	0,025	13
	–	9,5	16,5	0,3	9	9,4	17	0,3	0,025	13
	–	9,5	16,5	0,3	9	9,4	17	0,3	0,025	13
	12,2	–	19,2	0,3	9,4	–	19,6	0,3	0,025	12
	–	10,6	19,2	0,3	9,4	10,5	19,6	0,3	0,025	12
	–	10,6	19,2	0,3	9,4	10,5	19,6	0,3	0,025	12

Sealed single row deep groove ball bearings
d **8 – 9** mm



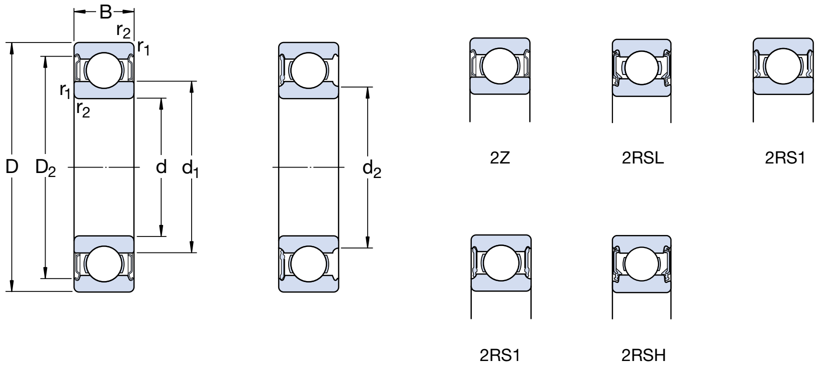
Principal dimensions d	 D	 B	Basic load ratings dynamic C	 static C_0	Fatigue load limit P_u	Speed ratings Reference speed	 Limiting[1)] speed	Mass	Designations sealed both sides	 sealed one side
mm			kN		kN	r/min		kg	–	
8	16	5	1,33	0,57	0,024	90 000	45 000	0,0036	**628/8-2Z**	**–**
	16	5	1,33	0,57	0,024	–	26 000	0,0036	**628/8-2RS1**	**–**
	16	6	1,33	0,57	0,024	90 000	45 000	0,0043	**638/8-2Z**	**–**
	19	6	1,9	0,74	0,031	80 000	40 000	0,0071	**619/8-2Z**	**–**
	19	6	1,9	0,74	0,031	–	24 000	0,0071	**619/8-2RS1**	**–**
	19	6	2,21	0,95	0,04	85 000	43 000	0,0072	**607/8-2Z**	**607/8-Z**
	22	7	3,45	1,37	0,057	75 000	38 000	0,012	* **608-2Z**	* **608-Z**
	22	7	3,45	1,37	0,057	75 000	38 000	0,012	* **608-2RSL**	* **608-RSL**
	22	7	3,45	1,37	0,057	–	22 000	0,012	* **608-2RSH**	* **608-RSH**
	22	11	3,45	1,37	0,057	–	22 000	0,016	**630/8-2RS1**	**–**
	24	8	3,9	1,66	0,071	63 000	32 000	0,017	* **628-2Z**	* **628-Z**
	24	8	3,9	1,66	0,071	63 000	32 000	0,017	* **628-2RZ**	* **628-RZ**
	24	8	3,9	1,66	0,071	–	19 000	0,017	* **628-2RS1**	* **628-RS1**
	28	9	4,62	1,96	0,083	60 000	30 000	0,030	**638-2RZ**	**638-RZ**
9	17	5	1,43	0,64	0,027	85 000	43 000	0,0043	**628/9-2Z**	**628/9-Z**
	17	5	1,43	0,64	0,027	–	24 000	0,0043	**628/9-2RS1**	**–**
	20	6	2,08	0,87	0,036	80 000	38 000	0,0076	**619/9-2Z**	**–**
	24	7	3,9	1,66	0,071	70 000	34 000	0,014	* **609-2Z**	* **609-Z**
	24	7	3,9	1,66	0,071	70 000	34 000	0,014	* **609-2RSL**	* **609-RSL**
	24	7	3,9	1,66	0,071	–	19 000	0,014	* **609-2RSH**	* **609-RSH**
	26	8	4,75	1,96	0,083	60 000	30 000	0,020	* **629-2Z**	* **629-Z**
	26	8	4,75	1,96	0,083	60 000	30 000	0,020	* **629-2RSL**	* **629-RSL**
	26	8	4,75	1,96	0,083	–	19 000	0,020	* **629-2RSH**	* **629-RSH**

* SKF Explorer bearing
1) For bearings with only one shield or low-friction seal (Z, RZ, RSL), the limiting speeds for open bearings are valid



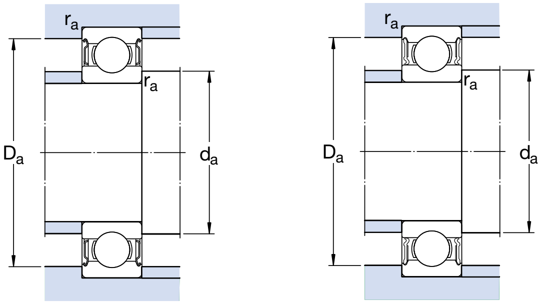
Dimensions					Abutment and fillet dimensions				Calculation factors	
d	d_1 ~	d_2 ~	D_2 ~	$r_{1,2}$ min	d_a min	d_a max	D_a max	r_a max	k_r	f_0
mm					mm				–	
8	10,1	–	14,5	0,2	9,4	–	14,6	0,2	0,015	11
	–	9,5	14,5	0,2	9,4	9,4	14,6	0,2	0,015	11
	10,1	–	14,5	0,2	9,4	–	14,6	0,2	0,015	11
	11,1	–	17	0,3	10	–	17	0,3	0,02	10
	–	10,4	17	0,3	10	10	17	0,3	0,02	10
	11,1	–	16,5	0,3	10	–	17	0,3	0,025	13
	12,1	–	19,2	0,3	10	–	20	0,3	0,025	12
	–	10,6	19,2	0,3	10	10,5	20	0,3	0,025	12
	–	10,6	19,2	0,3	10	10,5	20	0,3	0,025	12
	11,8	–	19	0,3	10	–	20	0,3	0,025	12
	14,5	–	20,6	0,3	10,4	–	21,6	0,3	0,025	13
	14,5	–	20,6	0,3	10,4	–	21,6	0,3	0,025	13
	14,5	–	20,6	0,3	10,4	–	21,6	0,3	0,025	13
	14,8	–	22,6	0,3	10,4	–	25,6	0,3	0,025	12
9	11,1	–	15,5	0,2	10,4	–	15,6	0,2	0,015	11
	–	10,6	15,5	0,2	10,4	10,5	15,6	0,2	0,015	11
	12	–	17,9	0,3	11	–	18	0,3	0,02	11
	14,4	–	21,2	0,3	11	–	22	0,3	0,025	13
	–	12,8	21,2	0,3	11	12,5	22	0,3	0,025	13
	–	12,8	21,2	0,3	11	12,5	22	0,3	0,025	13
	14,8	–	22,6	0,3	11,4	–	23,6	0,3	0,025	12
	–	13	22,6	0,3	11,4	12,5	23,6	0,3	0,025	12
	–	13	22,6	0,3	11,4	12,5	23,6	0,3	0,025	12

Sealed single row deep groove ball bearings
d **10 – 12** mm



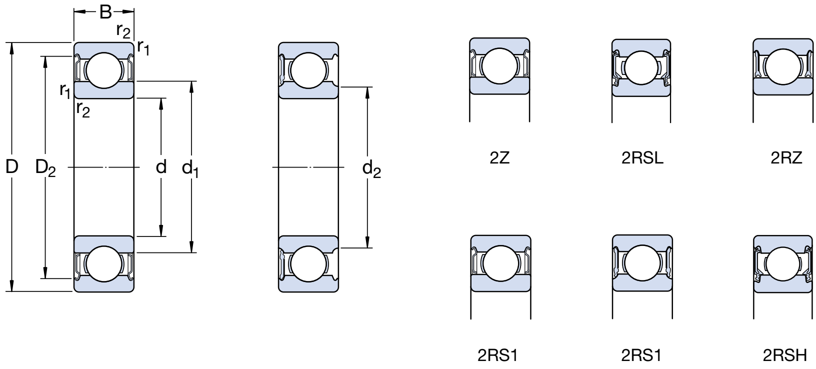
Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting[1)] speed	Mass	Designations sealed both sides	sealed one side
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
10	19	5	1,38	0,59	0,025	80 000	38 000	0,0055	**61800-2Z**	–
	19	5	1,38	0,59	0,025	–	22 000	0,0055	**61800-2RS1**	–
	22	6	2,08	0,85	0,036	75 000	36 000	0,010	**61900-2Z**	–
	22	6	2,08	0,85	0,036	–	20 000	0,010	**61900-2RS1**	–
	26	8	4,75	1,96	0,083	67 000	34 000	0,019	* **6000-2Z**	* **6000-Z**
	26	8	4,75	1,96	0,083	67 000	34 000	0,019	* **6000-2RSL**	* **6000-RSL**
	26	8	4,75	1,96	0,083	–	19 000	0,019	* **6000-2RSH**	* **6000-RSH**
	26	12	4,62	1,96	0,083	–	19 000	0,025	**63000-2RS1**	–
	28	8	4,62	1,96	0,083	63 000	32 000	0,022	**16100-2Z**	–
	30	9	5,4	2,36	0,1	56 000	28 000	0,032	* **6200-2Z**	* **6200-Z**
	30	9	5,4	2,36	0,1	56 000	28 000	0,032	* **6200-2RSL**	* **6200-RSL**
	30	9	5,4	2,36	0,1	–	17 000	0,032	* **6200-2RSH**	* **6200-RSH**
	30	14	5,07	2,36	0,1	–	17 000	0,04	**62200-2RS1**	–
	35	11	8,52	3,4	0,143	50 000	26 000	0,053	* **6300-2Z**	* **6300-Z**
	35	11	8,52	3,4	0,143	50 000	26 000	0,053	* **6300-2RSL**	* **6300-RSL**
	35	11	8,52	3,4	0,143	–	15 000	0,053	* **6300-2RSH**	* **6300-RSH**
	35	17	8,06	3,4	0,143	–	15 000	0,06	**62300-2RS1**	–
12	21	5	1,43	0,67	0,028	70 000	36 000	0,0063	**61801-2Z**	–
	21	5	1,43	0,67	0,028	–	20 000	0,0063	**61801-2RS1**	–
	24	6	2,25	0,98	0,043	67 000	32 000	0,011	**61901-2Z**	–
	24	6	2,25	0,98	0,043	–	19 000	0,011	**61901-2RS1**	–
	28	8	5,4	2,36	0,1	60 000	30 000	0,022	* **6001-2Z**	* **6001-Z**
	28	8	5,4	2,36	0,1	60 000	30 000	0,022	* **6001-2RSL**	* **6001-RSL**
	28	8	5,4	2,36	0,1	–	17 000	0,022	* **6001-2RSH**	* **6001-RSH**
	28	12	5,07	2,36	0,1	–	17 000	0,029	**63001-2RS1**	–
	30	8	5,07	2,36	0,1	56 000	28 000	0,023	**16101-2Z**	–
	30	8	5,07	2,36	0,1	–	16 000	0,023	**16101-2RS1**	–
	32	10	7,28	3,1	0,132	50 000	26 000	0,037	* **6201-2Z**	* **6201-Z**
	32	10	7,28	3,1	0,132	50 000	26 000	0,037	* **6201-2RSL**	* **6201-RSL**
	32	10	7,28	3,1	0,132	–	15 000	0,037	* **6201-2RSH**	* **6201-RSH**
	32	14	6,89	3,1	0,132	–	15 000	0,045	**62201-2RS1**	–
	37	12	10,1	4,15	0,176	45 000	22 000	0,060	* **6301-2Z**	* **6301-Z**
	37	12	10,1	4,15	0,176	45 000	22 000	0,060	* **6301-2RSL**	* **6301-RSL**
	37	12	10,1	4,15	0,176	–	14 000	0,060	* **6301-2RSH**	* **6301-RSH**
	37	17	9,75	4,15	0,176	–	14 000	0,070	**62301-2RS1**	–

* SKF Explorer bearing
1) For bearings with only one shield or low-friction seal (Z, RZ, RSL), the limiting speeds for open bearings are valid



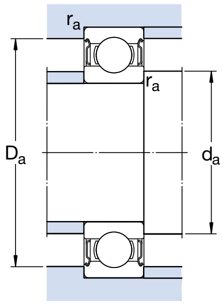
Dimensions					Abutment and fillet dimensions				Calculation factors	
d	d_1 ~	d_2 ~	D_2 ~	$r_{1,2}$ min	d_a min	d_a max	D_a max	r_a max	k_r	f_0
mm					mm				–	
10	12,6	–	17,3	0,3	12	–	17	0,3	0,015	9,4
	–	11,8	17,3	0,3	11,8	11,8	17	0,3	0,015	9,4
	13	–	19	0,3	12	–	20	0,3	0,02	9,3
	–	12	19	0,3	12	12	20	0,3	0,02	9,3
	14,8	–	22,6	0,3	12	–	24	0,3	0,025	12
	–	13	22,6	0,3	12	12,5	24	0,3	0,025	12
	–	13	22,6	0,3	12	12,5	24	0,3	0,025	12
	14,8	–	22,6	0,3	12	–	24	0,3	0,025	12
	16,7	–	24,8	0,6	14,2	–	23,8	0,3	0,025	13
	17	–	24,8	0,6	14,2	–	25,8	0,6	0,025	13
	–	15,2	24,8	0,6	14,2	15	25,8	0,6	0,025	13
	–	15,2	24,8	0,6	14,2	15	25,8	0,6	0,025	13
	17	–	24,8	0,6	14,2	–	25,8	0,6	0,025	13
	17,5	–	28,7	0,6	14,2	–	30,8	0,6	0,03	11
	–	15,7	28,7	0,6	14,2	15,5	30,8	0,6	0,03	11
	–	15,7	28,7	0,6	14,2	15,5	30,8	0,6	0,03	11
	17,5	–	28,7	0,6	14,2	–	30,8	0,6	0,03	11
12	15	–	19,1	0,3	14	–	19	0,3	0,015	9,7
	–	14,1	19,1	0,3	14	14	19	0,3	0,015	9,7
	15,5	–	21,4	0,3	14	–	22	0,3	0,02	9,7
	15,5	–	21,4	0,3	14	–	22	0,3	0,02	9,7
	17	–	24,8	0,3	14	–	26	0,3	0,025	13
	–	15,2	24,8	0,3	14	15	26	0,3	0,025	13
	–	15,2	24,8	0,3	14	15	26	0,3	0,025	13
	17	–	24,8	0,3	14	–	26	0,3	0,025	13
	16,7	–	24,8	0,3	14,4		27,6	0,3	0,025	13
	16,7	–	24,8	0,3	14,4		27,6	0,3	0,025	13
	18,5	–	27,4	0,6	16,2	–	27,8	0,6	0,025	12
	–	16,6	27,4	0,6	16,2	16,5	27,8	0,6	0,025	12
	–	16,6	27,4	0,6	16,2	16,5	27,8	0,6	0,025	12
	18,5	–	27,4	0,6	16,2	–	27,8	0,6	0,025	12
	19,5	–	31,5	1	17,6	–	31,4	1	0,03	11
	–	17,7	31,5	1	17,6	17,6	31,4	1	0,03	11
	–	17,7	31,5	1	17,6	17,6	31,4	1	0,03	11
	19,5	–	31,5	1	17,6	–	31,4	1	0,03	11

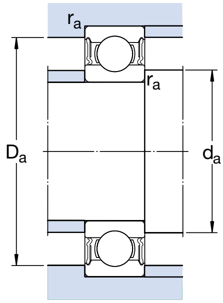
Sealed single row deep groove ball bearings
d **15 – 17** mm



Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting[1] speed	Mass	Designations sealed both sides	sealed one side
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
15	24	5	1,56	0,8	0,034	60 000	30 000	0,0074	**61802-2Z**	–
	24	5	1,56	0,8	0,034	–	17 000	0,0074	**61802-2RS1**	–
	28	7	4,36	2,24	0,095	56 000	28 000	0,016	**61902-2Z**	–
	28	7	4,36	2,24	0,095	56 000	28 000	0,016	**61902-2RZ**	–
	28	7	4,36	2,24	0,095	–	16 000	0,016	**61902-2RS1**	–
	32	8	5,85	2,85	0,12	50 000	26 000	0,025	* **16002-2Z**	* **16002-Z**
	32	9	5,85	2,85	0,12	50 000	26 000	0,030	* **6002-2Z**	* **6002-Z**
	32	9	5,85	2,85	0,12	50 000	26 000	0,030	* **6002-2RSL**	* **6002-RSL**
	32	9	5,85	2,85	0,12	–	14 000	0,030	* **6002-2RSH**	* **6002-RSH**
	32	13	5,59	2,85	0,12	–	14 000	0,039	**63002-2RS1**	–
	35	11	8,06	3,75	0,16	43 000	22 000	0,045	* **6202-2Z**	* **6202-Z**
	35	11	8,06	3,75	0,16	43 000	22 000	0,045	* **6202-2RSL**	* **6202-RSL**
	35	11	8,06	3,75	0,16	–	13 000	0,045	* **6202-2RSH**	* **6202-RSH**
	35	14	7,8	3,75	0,16	–	13 000	0,054	**62202-2RS1**	–
	42	13	11,9	5,4	0,228	38 000	19 000	0,082	* **6302-2Z**	* **6302-Z**
	42	13	11,9	5,4	0,228	38 000	19 000	0,082	* **6302-2RSL**	* **6302-RSL**
	42	13	11,9	5,4	0,228	–	12 000	0,082	* **6302-2RSH**	* **6302–RSH**
	42	17	11,4	5,4	0,228	–	12 000	0,11	**62302-2RS1**	–
17	26	5	1,68	0,93	0,039	56 000	28 000	0,0082	**61803-2Z**	–
	26	5	1,68	0,93	0,039	56 000	28 000	0,0082	**61803-2RZ**	–
	26	5	1,68	0,93	0,039	–	16 000	0,0082	**61803-2RS1**	–
	30	7	4,62	2,55	0,108	50 000	26 000	0,018	**61903-2Z**	–
	30	7	4,62	2,55	0,108	50 000	26 000	0,018	**61903-2RZ**	–
	30	7	4,62	2,55	0,108	–	14 000	0,018	**61903-2RS1**	–
	35	8	6,37	3,25	0,137	45 000	22 000	0,032	* **16003-2Z**	–
	35	10	6,37	3,25	0,137	45 000	22 000	0,039	* **6003-2Z**	* **6003-Z**
	35	10	6,37	3,25	0,137	45 000	22 000	0,039	* **6003-2RSL**	* **6003-RSL**
	35	10	6,37	3,25	0,137	–	13 000	0,039	* **6003-2RSH**	* **6003-RSH**
	35	14	6,05	3,25	0,137	–	13 000	0,052	**63003-2RS1**	–
	40	12	9,95	4,75	0,2	38 000	19 000	0,065	* **6203-2Z**	* **6203-Z**
	40	12	9,95	4,75	0,2	38 000	19 000	0,065	* **6203-2RSL**	* **6203-RSL**
	40	12	9,95	4,75	0,2	–	12 000	0,065	* **6203-2RSH**	* **6203-RSH**
	40	16	9,56	4,75	0,2	–	12 000	0,083	**62203-2RS1**	–
	47	14	14,3	6,55	0,275	34 000	17 000	0,12	* **6303-2Z**	* **6303-Z**
	47	14	14,3	6,55	0,275	34 000	17 000	0,12	* **6303-2RSL**	* **6303-RSL**
	47	14	14,3	6,55	0,275	–	11 000	0,12	* **6303-2RSH**	* **6303-RSH**
	47	19	13,5	6,55	0,275	–	11 000	0,15	**62303-2RS1**	–

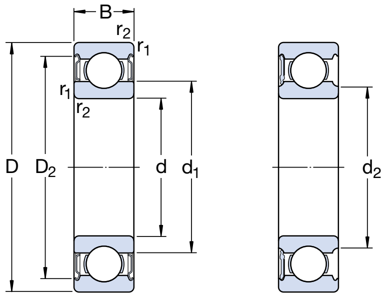
* SKF Explorer bearing
1) For bearings with only one shield or low-friction seal (Z, RZ, RSL), the limiting speeds for open bearings are valid

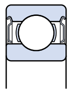


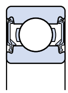


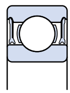
Dimensions					Abutment and fillet dimensions				Calculation factors	
d	d_1 ~	d_2 ~	D_2 ~	$r_{1,2}$ min	d_a min	d_a max	D_a max	r_a max	k_r	f_0
mm					mm				–	
15	17,9	–	22,1	0,3	17	–	22	0,3	0,015	10
	17,9	–	22,1	0,3	17	–	22	0,3	0,015	10
	18,4	–	25,8	0,3	17	–	26	0,3	0,02	14
	18,4	–	25,8	0,3	17	–	26	0,3	0,02	14
	–	17,4	25,8	0,3	17	17,3	26	0,3	0,02	14
	20,2	–	28,2	0,3	17	–	30	0,3	0,02	14
	20,5	–	28,2	0,3	17	–	30	0,3	0,025	14
	–	18,7	28,2	0,3	17	18,5	30	0,3	0,025	14
	–	18,7	28,2	0,3	17	18,5	30	0,3	0,025	14
	20,5	–	28,2	0,3	17	–	30	0,3	0,025	14
	21,7	–	30,4	0,6	19,2	–	30,8	0,6	0,025	13
	–	19,4	30,4	0,6	19,2	19,4	30,8	0,6	0,025	13
	–	19,4	30,4	0,6	19,2	19,4	30,8	0,6	0,025	13
	21,7	–	30,4	0,6	19,2	–	30,8	0,6	0,025	13
	23,7	–	36,3	1	20,6	–	36,4	1	0,03	12
	–	21,1	36,3	1	20,6	21	36,4	1	0,03	12
	–	21,1	36,3	1	20,6	21	36,4	1	0,03	12
	23,7	–	36,3	1	20,6	–	36,4	1	0,03	12
17	20,2	–	24,1	0,3	19	–	24	0,3	0,015	10
	20,2	–	24,1	0,3	19	–	24	0,3	0,015	10
	–	19,3	24,1	0,3	19	19,2	24	0,3	0,015	10
	20,4	–	27,8	0,3	19	–	28	0,3	0,02	15
	20,4	–	27,8	0,3	19	–	28	0,3	0,02	15
	–	19,4	27,8	0,3	19	19,3	28	0,3	0,02	15
	22,7	–	31,2	0,3	19	–	33	0,3	0,02	14
	23	–	31,4	0,3	19	–	33	0,3	0,025	14
	–	20,7	31,4	0,3	19	20,5	33	0,3	0,025	14
	–	20,7	31,4	0,3	19	20,5	33	0,3	0,025	14
	23	–	31,4	0,3	19	–	33	0,3	0,025	14
	24,5	–	35	0,6	21,2	–	35,8	0,6	0,025	13
	–	22,2	35	0,6	21,2	22	35,8	0,6	0,025	13
	–	22,2	35	0,6	21,2	22	35,8	0,6	0,025	13
	24,5	–	35	0,6	21,2	–	35,8	0,6	0,025	13
	26,5	–	39,7	1	22,6	–	41,4	1	0,03	12
	–	24	39,7	1	22,6	23,5	41,4	1	0,03	12
	–	24	39,7	1	22,6	23,5	41,4	1	0,03	12
	26,5	–	39,7	1	22,6	–	41,4	1	0,03	12

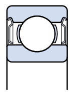
Sealed single row deep groove ball bearings
d **20 – 25** mm

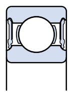


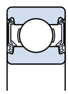

2Z


2RSL


2RZ

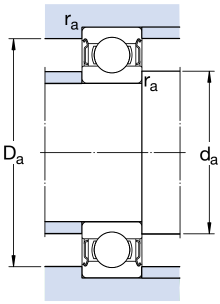

2RS1

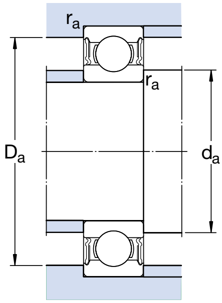

2RS1


2RSH

Principal dimensions			**Basic load ratings** dynamic	static	**Fatigue load limit**	**Speed ratings** Reference speed	Limiting[1] speed	**Mass**	**Designations** sealed both sides	sealed one side
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
20	32	7	4,03	2,32	0,104	45 000	22 000	0,018	**61804-2RZ**	–
	32	7	4,03	2,32	0,104	–	13 000	0,018	**61804-2RS1**	–
	37	9	6,37	3,65	0,156	43 000	20 000	0,038	**61904-2RZ**	–
	37	9	6,37	3,65	0,156	–	12 000	0,038	**61904-2RS1**	–
	42	12	9,95	5	0,212	38 000	19 000	0,069	* **6004-2Z**	* **6004-Z**
	42	12	9,95	5	0,212	38 000	19 000	0,069	* **6004-2RSL**	* **6004-RSL**
	42	12	9,95	5	0,212	–	11 000	0,069	* **6004-2RSH**	* **6004-RSH**
	42	16	9,36	5	0,212	–	11 000	0,086	**63004-2RS1**	–
	47	14	13,5	6,55	0,28	32 000	17 000	0,11	* **6204-2Z**	* **6204-Z**
	47	14	13,5	6,55	0,28	32 000	17 000	0,11	* **6204-2RSL**	* **6204-RSL**
	47	14	13,5	6,55	0,28	–	10 000	0,11	* **6204-2RSH**	* **6204-RSH**
	47	18	12,7	6,55	0,28	–	10 000	0,13	**62204-2RS1**	–
	52	15	16,8	7,8	0,335	30 000	15 000	0,14	* **6304-2Z**	* **6304-Z**
	52	15	16,8	7,8	0,335	30 000	15 000	0,14	* **6304-2RSL**	* **6304-RSL**
	52	15	16,8	7,8	0,335	–	9 500	0,14	* **6304-2RSH**	* **6304-RSH**
	52	21	15,9	7,8	0,335	–	9 500	0,20	**62304-2RS1**	–
22	50	14	14	7,65	0,325	–	9 000	0,12	**62/22-2RS1**	–
25	37	7	4,36	2,6	0,125	38 000	19 000	0,022	**61805-2RZ**	–
	37	7	4,36	2,6	0,125	–	11 000	0,022	**61805-2RS1**	–
	42	9	7,02	4,3	0,193	36 000	18 000	0,045	**61905-2RZ**	–
	42	9	7,02	4,3	0,193	–	10 000	0,045	**61905-2RS1**	–
	47	12	11,9	6,55	0,275	32 000	16 000	0,08	* **6005-2Z**	* **6005-Z**
	47	12	11,9	6,55	0,275	32 000	16 000	0,08	* **6005-2RSL**	* **6005-RSL**
	47	12	11,9	6,55	0,275	–	9 500	0,08	* **6005-2RSH**	* **6005-RSH**
	47	16	11,2	6,55	0,275	–	9 500	0,10	**63005-2RS1**	–
	52	15	14,8	7,8	0,335	28 000	14 000	0,13	* **6205-2Z**	* **6205-Z**
	52	15	14,8	7,8	0,335	28 000	14 000	0,13	* **6205-2RSL**	* **6205-RSL**
	52	15	14,8	7,8	0,335	–	8 500	0,13	* **6205-2RSH**	* **6205-RSH**
	52	18	14	7,8	0,335	–	8 500	0,15	**62205-2RS1**	–
	62	17	23,4	11,6	0,49	24 000	13 000	0,23	* **6305-2Z**	* **6305-Z**
	62	17	23,4	11,6	0,49	24 000	13 000	0,23	* **6305-2RZ**	* **6305-RZ**
	62	17	23,4	11,6	0,49	–	7 500	0,23	* **6305-2RS1**	* **6305-RS1**
	62	24	22,5	11,6	0,49	–	7 500	0,32	**62305-2RS1**	–

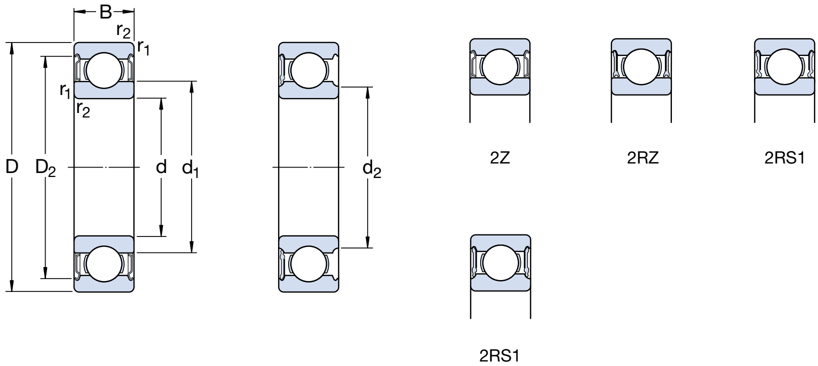
* SKF Explorer bearing
[1] For bearings with only one shield or low-friction seal (Z, RZ, RSL), the limiting speeds for open bearings are valid





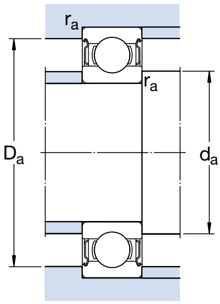
Dimensions					Abutment and fillet dimensions				Calculation factors	
d	d_1 ~	d_2 ~	D_2 ~	$r_{1,2}$ min	d_a min	d_a max	D_a max	r_a max	k_r	f_0
mm					mm				–	
20	24	–	29,5	0,3	22	–	30	0,3	0,015	15
	–	22,6	29,5	0,3	22	22,5	30	0,3	0,015	15
	25,6	–	32,8	0,3	22	–	35	0,3	0,02	15
	–	24,2	32,8	0,3	22	24	35	0,3	0,02	15
	27,2	–	37,2	0,6	23,2	–	38,8	0,6	0,025	14
	–	24,9	37,2	0,6	23,2	24,5	38,8	0,6	0,025	14
	–	24,9	37,2	0,6	23,2	24,5	38,8	0,6	0,025	14
	27,2	–	37,2	0,6	23,2	–	38,8	0,6	0,025	14
	28,8	–	40,6	1	25,6	–	41,4	1	0,025	13
	–	26,3	40,6	1	25,6	26	41,4	1	0,025	13
	–	26,3	40,6	1	25,6	26	41,4	1	0,025	13
	28,8	–	40,6	1	25,6	–	41,4	1	0,025	13
	30,4	–	44,8	1,1	27	–	45	1	0,03	12
	–	27,2	44,8	1,1	27	27	45	1	0,03	12
	–	27,2	44,8	1,1	27	27	45	1	0,03	12
	30,4	–	44,8	1,1	27	–	45	1	0,03	12
22	–	32,2	44	1	27,6	32	44,4	1	0,025	14
25	28,5	–	34,3	0,3	27	–	35	0,3	0,015	14
	–	27,4	34,3	0,3	27	27,3	35	0,3	0,015	14
	30,2	–	37,8	0,3	27	–	40	0,3	0,02	15
	–	29,2	37,8	0,3	27	29	40	0,3	0,02	15
	32	–	42,2	0,6	28,2	–	43,8	0,6	0,025	14
	–	29,7	42,2	0,6	28,2	29,5	43,8	0,6	0,025	14
	–	29,7	42,2	0,6	28,2	29,5	43,8	0,6	0,025	14
	32	–	42,2	0,6	29,2	–	43,8	0,6	0,025	14
	34,4	–	46,3	1	30,6	–	46,4	1	0,025	14
	–	31,8	46,3	1	30,6	31,5	46,4	1	0,025	14
	–	31,8	46,3	1	30,6	31,5	46,4	1	0,025	14
	34,4	–	46,3	1	30,6	–	46,4	1	0,025	14
	36,6	–	52,7	1,1	32	–	55	1	0,03	12
	36,6	–	52,7	1,1	32	–	55	1	0,03	12
	36,6	–	52,7	1,1	32	–	55	1	0,03	12
	36,6	–	52,7	1,1	32	–	55	1	0,03	12

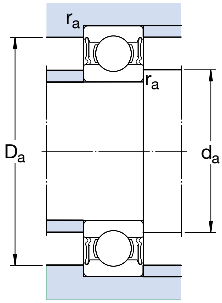
Sealed single row deep groove ball bearings
d **30 – 35** mm



Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting[1)] speed	Mass	Designations sealed both sides	sealed one side
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
30	42	7	4,49	2,9	0,146	32 000	16 000	0,027	**61806-2RZ**	**–**
	42	7	4,49	2,9	0,146	–	9 500	0,027	**61806-2RS1**	**–**
	47	9	7,28	4,55	0,212	30 000	15 000	0,051	**61906-2RZ**	**–**
	47	9	7,28	4,55	0,212	–	8 500	0,051	**61906-2RS1**	**–**
	55	13	13,8	8,3	0,355	28 000	14 000	0,12	* **6006-2Z**	* **6006-Z**
	55	13	13,8	8,3	0,355	28 000	14 000	0,12	* **6006-2RZ**	* **6006-RZ**
	55	13	13,8	8,3	0,355	–	8 000	0,12	* **6006-2RS1**	* **6006-RS1**
	55	19	13,3	8,3	0,355	–	8 000	0,16	**63006-2RS1**	**–**
	62	16	20,3	11,2	0,475	24 000	12 000	0,20	* **6206-2Z**	* **6206-Z**
	62	16	20,3	11,2	0,475	24 000	12 000	0,20	* **6206-2RZ**	* **6206-RZ**
	62	16	20,3	11,2	0,475	–	7 500	0,20	* **6206-2RS1**	* **6206-RS1**
	62	20	19,5	11,2	0,475	–	7 500	0,24	**62206-2RS1**	**–**
	72	19	29,6	16	0,67	20 000	11 000	0,35	* **6306-2Z**	* **6306-Z**
	72	19	29,6	16	0,67	20 000	11 000	0,35	* **6306-2RZ**	* **6306-RZ**
	72	19	29,6	16	0,67	–	6 300	0,35	* **6306-2RS1**	* **6306-RS1**
	72	27	28,1	16	0,67	–	6 300	0,48	**62306-2RS1**	**–**
35	47	7	4,75	3,2	0,166	28 000	14 000	0,03	**61807-2RZ**	**–**
	47	7	4,75	3,2	0,166	–	8 000	0,03	**61807-2RS1**	**–**
	55	10	9,56	6,8	0,29	26 000	13 000	0,08	**61907-2RZ**	**–**
	55	10	9,56	6,8	0,29	–	7 500	0,08	**61907-2RS1**	**–**
	62	14	16,8	10,2	0,44	24 000	12 000	0,16	* **6007-2Z**	* **6007-Z**
	62	14	16,8	10,2	0,44	24 000	12 000	0,16	* **6007-2RZ**	* **6007-RZ**
	62	14	16,8	10,2	0,44	–	7 000	0,16	* **6007-2RS1**	* **6007-RS1**
	62	20	15,9	10,2	0,44	–	7 000	0,21	**63007-2RS1**	**–**
	72	17	27	15,3	0,655	20 000	10 000	0,29	* **6207-2Z**	* **6207-Z**
	72	17	27	15,3	0,655	–	6 300	0,29	* **6207-2RS1**	* **6207-RS1**
	72	23	25,5	15,3	0,655	–	6 300	0,37	**62207-2RS1**	**–**
	80	21	35,1	19	0,815	19 000	9 500	0,46	* **6307-2Z**	* **6307-Z**
	80	21	35,1	19	0,815	–	6 000	0,46	* **6307-2RS1**	* **6307-RS1**
	80	31	33,2	19	0,815	–	6 000	0,66	**62307-2RS1**	**–**

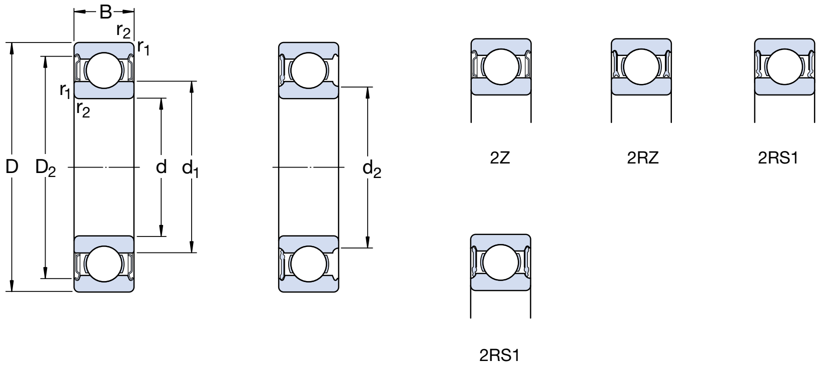
* SKF Explorer bearing
1) For bearings with only one shield or low-friction seal (Z, RZ, RSL), the limiting speeds for open bearings are valid





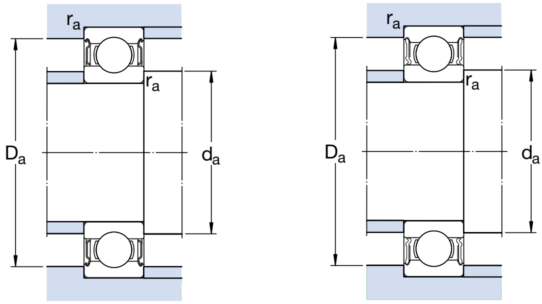
Dimensions					Abutment and fillet dimensions				Calculation factors	
d	d_1 ~	d_2 ~	D_2 ~	$r_{1,2}$ min	d_a min	d_a max	D_a max	r_a max	k_r	f_0
mm					mm				–	
30	33,7	–	39,5	0,3	32	–	40	0,3	0,015	14
	–	32,6	39,5	0,3	32	32,5	40	0,3	0,015	14
	35,2	–	42,8	0,3	32	–	45	0,3	0,02	14
	–	34,2	42,8	0,3	32	34	45	0,3	0,02	14
	38,2	–	49	1	34,6	–	50,4	1	0,025	15
	38,2	–	49	1	34,6	–	50,4	1	0,025	15
	38,2	–	49	1	34,6	–	50,4	1	0,025	15
	38,2	–	49	1	34,6	–	50,4	1	0,025	15
	40,4	–	54,1	1	35,6	–	56,4	1	0,025	14
	40,4	–	54,1	1	35,6	–	56,4	1	0,025	14
	40,4	–	54,1	1	35,6	–	56,4	1	0,025	14
	40,4	–	54,1	1	35,6	–	56,4	1	0,025	14
	44,6	–	61,9	1,1	37	–	65	1	0,03	13
	44,6	–	61,9	1,1	37	–	65	1	0,03	13
	44,6	–	61,9	1,1	37	–	65	1	0,03	13
	44,6	–	61,9	1,1	37	–	65	1	0,03	13
35	38,7	–	44,4	0,3	37	–	45	0,3	0,015	14
	–	37,6	44,4	0,3	37	37,5	45	0,3	0,015	14
	41,6	–	50,5	0,6	38,2	–	51,8	0,6	0,02	14
	41,6	–	50,5	0,6	38,2	–	51,8	0,6	0,02	14
	43,8	–	55,6	1	39,6	–	57,4	1	0,025	15
	43,8	–	55,6	1	39,6	–	57,4	1	0,025	15
	43,8	–	55,6	1	39,6	–	57,4	1	0,025	15
	43,8	–	55,6	1	39,6	–	57,4	1	0,025	15
	46,9	–	62,7	1,1	42	–	65	1	0,025	14
	46,9	–	62,7	1,1	42	–	65	1	0,025	14
	46,9	–	62,7	1,1	42	–	65	1	0,025	14
	49,6	–	69,2	1,5	44	–	71	1,5	0,03	13
	49,6	–	69,2	1,5	44	–	71	1,5	0,03	13
	49,6	–	69,2	1,5	44	–	71	1,5	0,03	13

Sealed single row deep groove ball bearings
d **40 – 45** mm



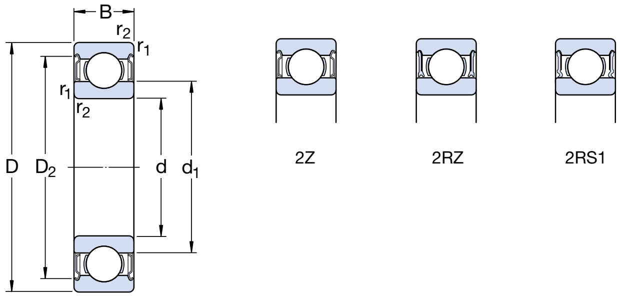
Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting[1)] speed	Mass	Designations sealed both sides	sealed one side
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
40	52	7	4,94	3,45	0,186	26 000	13 000	0,034	**61808-2RZ**	–
	52	7	4,94	3,45	0,186	–	7 500	0,034	**61808-2RS1**	–
	62	12	13,8	10	0,425	24 000	12 000	0,12	**61908-2RZ**	–
	62	12	13,8	10	0,425	–	6 700	0,12	**61908-2RS1**	–
	68	15	17,8	11,6	0,49	22 000	11 000	0,19	* **6008-2Z**	* **6008-Z**
	68	15	17,8	11,6	0,49	22 000	11 000	0,19	* **6008-2RZ**	* **6008-RZ**
	68	15	17,8	11,6	0,49	–	6 300	0,19	* **6008-2RS1**	* **6008-RS1**
	68	21	16,8	11,6	0,49	–	6 300	0,26	**63008-2RS1**	–
	80	18	32,5	19	0,8	18 000	9 000	0,37	* **6208-2Z**	* **6208-Z**
	80	18	32,5	19	0,8	18 000	9 000	0,37	* **6208-2RZ**	* **6208-RZ**
	80	18	32,5	19	0,8	–	5 600	0,37	* **6208-2RS1**	* **6208-RS1**
	80	23	30,7	19	0,8	–	5 600	0,44	**62208-2RS1**	–
	90	23	42,3	24	1,02	17 000	8 500	0,63	* **6308-2Z**	* **6308-Z**
	90	23	42,3	24	1,02	17 000	8 500	0,63	* **6308-2RZ**	* **6308-RZ**
	90	23	42,3	24	1,02	–	5 000	0,63	* **6308-2RS1**	* **6308-RS1**
	90	33	41	24	1,02	–	5 000	0,89	**62308-2RS1**	–
45	58	7	6,63	6,1	0,26	22 000	11 000	0,04	**61809-2RZ**	–
	58	7	6,63	6,1	0,26	–	6 700	0,04	**61809-2RS1**	–
	68	12	14	10,8	0,465	20 00	10 000	0,14	**61909-2RZ**	–
	68	12	14	10,8	0,465	–	6 000	0,14	**61909-2RS1**	–
	75	16	22,1	14,6	0,64	20 000	10 000	0,25	* **6009-2Z**	* **6009-Z**
	75	16	22,1	14,6	0,64	–	5 600	0,25	* **6009-2RS1**	* **6009-RS1**
	75	23	20,8	14,6	0,64	–	5 600	0,34	**63009-2RS1**	–
	85	19	35,1	21,6	0,915	17 000	8 500	0,41	* **6209-2Z**	* **6209-Z**
	85	19	35,1	21,6	0,915	–	5 000	0,41	* **6209-2RS1**	* **6209-RS1**
	85	23	33,2	21,6	0,915	–	5 000	0,48	**62209-2RS1**	–
	100	25	55,3	31,5	1,34	15 000	7 500	0,83	* **6309-2Z**	* **6309-Z**
	100	25	55,3	31,5	1,34	–	4 500	0,83	* **6309-2RS1**	* **6309-RS1**
	100	36	52,7	31,5	1,34	–	4 500	1,15	**62309-2RS1**	–

* SKF Explorer bearing
1) For bearings with only one shield or low-friction seal (Z, RZ, RSL), the limiting speeds for open bearings are valid



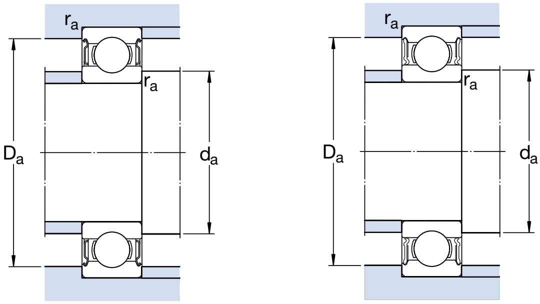
Dimensions					Abutment and fillet dimensions				Calculation factors	
d	d_1 ~	d_2 ~	D_2 ~	$r_{1,2}$ min	d_a min	d_a max	D_a max	r_a max	k_r	f_0
mm					mm				–	
40	43,7	–	49,6	0,3	42	–	50	0,3	0,015	14
	–	42,6	49,6	0,3	42	42,5	50	0,3	0,015	14
	46,9	–	57,3	0,6	43,2	–	58,8	0,6	0,02	16
	46,9	–	57,3	0,6	43,2	–	58,8	0,6	0,02	16
	49,3	–	61,1	1	44,6	–	63,4	1	0,025	15
	49,3	–	61,1	1	44,6	–	63,4	1	0,025	15
	49,3	–	61,1	1	44,6	–	63,4	1	0,025	15
	49,3	–	61,1	1	44,6	–	63,4	1	0,025	15
	52,6	–	69,8	1,1	47	–	73	1	0,025	14
	52,6	–	69,8	1,1	47	–	73	1	0,025	14
	52,6	–	69,8	1,1	47	–	73	1	0,025	14
	52,6	–	69,8	1,1	47	–	73	1	0,025	14
	56,1	–	77,7	1,5	49	–	81	1,5	0,03	13
	56,1	–	77,7	1,5	49	–	81	1,5	0,03	13
	56,1	–	77,7	1,5	49	–	81	1,5	0,03	13
	56,1	–	77,7	1,5	49	–	81	1,5	0,03	13
45	49,1	–	55,4	0,3	47	–	56	0,3	0,015	17
	49,1	–	55,4	0,3	47	–	56	0,3	0,015	17
	52,4	–	62,8	0,6	48,2	–	64,8	0,6	0,02	16
	52,4	–	62,8	0,6	48,2	–	64,8	0,6	0,02	16
	54,8	–	67,8	1	50,8	–	69,2	1	0,025	15
	54,8	–	67,8	1	50,8	–	69,2	1	0,025	15
	54,8	–	67,8	1	50,8	–	69,2	1	0,025	15
	57,6	–	75,2	1,1	52	–	78	1	0,025	14
	57,6	–	75,2	1,1	52	–	78	1	0,025	14
	57,6	–	75,2	1,1	52	–	78	1	0,025	14
	62,2	–	86,7	1,5	54	–	91	1,5	0,03	13
	62,2	–	86,7	1,5	54	–	91	1,5	0,03	13
	62,2	–	86,7	1,5	54	–	91	1,5	0,03	13

Sealed single row deep groove ball bearings
d **50 – 55** mm



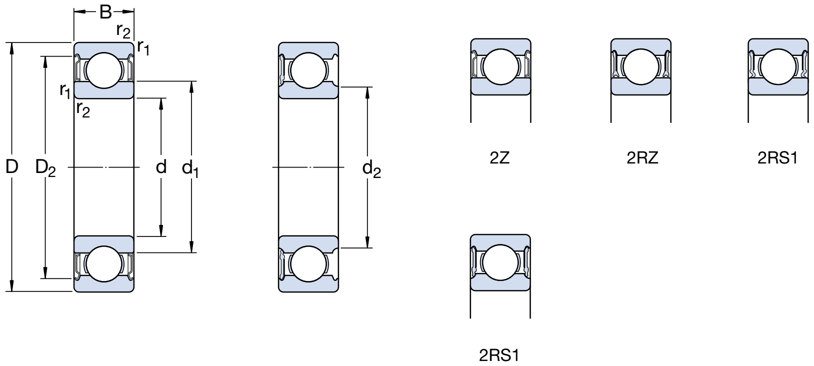
Principal dimensions d	D	B	Basic load ratings dynamic C	static C_0	Fatigue load limit P_u	Speed ratings Reference speed	Limiting[1)] speed	Mass	Designations sealed both sides	sealed one side
mm			kN		kN	r/min		kg	–	
50	65	7	6,76	6,8	0,285	20 000	10 000	0,052	**61810-2RZ**	–
	65	7	6,76	6,8	0,285	–	6 000	0,052	**61810-2RS1**	–
	72	12	14,6	11,8	0,5	19 000	9 500	0,14	**61910-2RZ**	–
	72	12	14,6	11,8	0,5	–	5 600	0,14	**61910-2RS1**	–
	80	16	22,9	16	0,71	18 000	9 000	0,26	*** 6010-2Z**	*** 6010-Z**
	80	16	22,9	16	0,71	18 000	9 000	0,26	*** 6010-2RZ**	*** 6010-RZ**
	80	16	22,9	16	0,71	–	5 000	0,26	*** 6010-2RS1**	*** 6010-RS1**
	80	23	21,6	16	0,71	–	5 000	0,37	**63010-2RS1**	–
	90	20	37,1	23,2	0,98	15 000	8 000	0,46	*** 6210-2Z**	*** 6210-Z**
	90	20	37,1	23,2	0,98	15 000	8 000	0,46	*** 6210-2RZ**	*** 6210-RZ**
	90	20	37,1	23,2	0,98	–	4 800	0,46	*** 6210-2RS1**	*** 6210-RS1**
	90	23	35,1	23,2	0,98	–	4 800	0,52	**62210-2RS1**	–
	110	27	65	38	1,6	13 000	6 700	1,05	*** 6310-2Z**	*** 6310-Z**
	110	27	65	38	1,6	–	4 300	1,05	*** 6310-2RS1**	*** 6310-RS1**
	110	40	61,8	38	1,6	–	4 300	1,55	**62310-2RS1**	–
55	72	9	9,04	8,8	0,375	19 000	9 500	0,083	**61811-2RZ**	–
	72	9	9,04	8,8	0,375	–	5 300	0,083	**61811-2RS1**	–
	80	13	16,5	14	0,6	17 000	8 500	0,19	**61911-2RZ**	–
	80	13	16,5	14	0,6	–	5 000	0,19	**61911-2RS1**	–
	90	18	29,6	21,2	0,9	16 000	8 000	0,39	*** 6011-2Z**	*** 6011-Z**
	90	18	29,6	21,2	0,9	–	4 500	0,39	*** 6011-2RS1**	*** 6011-RS1**
	100	21	46,2	29	1,25	14 000	7 000	0,61	*** 6211-2Z**	*** 6211-Z**
	100	21	46,2	29	1,25	–	4 300	0,61	*** 6211-2RS1**	*** 6211-RS1**
	100	25	43,6	29	1,25	–	4 300	0,70	**62211-2RS1**	–
	120	29	74,1	45	1,9	12 000	6 300	1,35	*** 6311-2Z**	*** 6311-Z**
	120	29	74,1	45	1,9	–	3 800	1,35	*** 6311-2RS1**	*** 6311-RS1**
	120	43	71,5	45	1,9	–	3 800	1,95	**62311-2RS1**	–

* SKF Explorer bearing
1) For bearings with only one shield or low-friction seal (Z, RZ, RSL), the limiting speeds for open bearings are valid



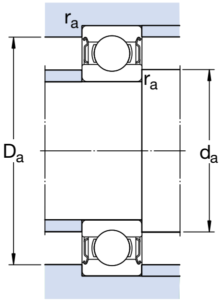
Dimensions				Abutment and fillet dimensions			Calculation factors	
d	d_1 ~	D_2 ~	$r_{1,2}$ min	d_a min	D_a max	r_a max	k_r	f_0
mm				mm			–	
50	55,1	61,8	0,3	52	63	0,3	0,015	17
	55,1	61,8	0,3	52	63	0,3	0,015	17
	56,9	67,3	0,6	53,2	68,8	0,6	0,02	16
	56,9	67,3	0,6	53,2	68,8	0,6	0,02	16
	59,8	72,8	1	54,6	75,4	1	0,025	15
	59,8	72,8	1	54,6	75,4	1	0,025	15
	59,8	72,8	1	54,6	75,4	1	0,025	15
	59,8	72,8	1	54,6	75,4	1	0,025	15
	62,5	81,6	1,1	57	83	1	0,025	14
	62,5	81,6	1,1	57	83	1	0,025	14
	62,5	81,6	1,1	57	83	1	0,025	14
	62,5	81,6	1,1	57	83	1	0,025	14
	68,8	95,2	2	59	101	2	0,03	13
	68,8	95,2	2	59	101	2	0,03	13
	68,8	95,2	2	59	101	2	0,03	13
55	60,6	68,6	0,3	57	70	0,3	0,015	17
	60,6	68,6	0,3	57	70	0,3	0,015	17
	63,2	74,2	1	59,6	75,4	1	0,02	16
	63,2	74,2	1	59,6	75,4	1	0,02	16
	66,3	81,5	1,1	61	84	1	0,025	15
	66,3	81,5	1,1	61	84	1	0,025	15
	69,1	89,4	1,5	64	91	1,5	0,025	14
	69,1	89,4	1,5	64	91	1,5	0,025	14
	69,1	89,4	1,5	64	91	1,5	0,025	14
	75,3	104	2	66	109	2	0,03	13
	75,3	104	2	66	109	2	0,03	13
	75,3	104	2	66	109	2	0,03	13

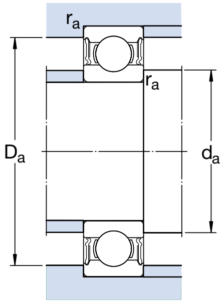
Sealed single row deep groove ball bearings
d **60 – 65** mm



Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting[1)] speed	Mass	Designations sealed both sides	sealed one side
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
60	78	10	11,9	11,4	0,49	17 000	8 500	0,11	**61812-2RZ**	**–**
	78	10	11,9	11,4	0,49	–	4 800	0,11	**61812-2RS1**	**–**
	85	13	16,5	14,3	0,6	16 000	8 000	0,20	**61912-2RZ**	**–**
	85	13	16,5	14,3	0,6	–	4 500	0,20	**61912-2RS1**	**–**
	95	18	30,7	23,2	0,98	15 000	7 500	0,42	*** 6012-2Z**	*** 6012-Z**
	95	18	30,7	23,2	0,98	15 000	7 500	0,42	*** 6012-2RZ**	*** 6012-RZ**
	95	18	30,7	23,2	0,98	–	4 300	0,42	*** 6012-2RS1**	*** 6012-RS1**
	110	22	55,3	36	1,53	13 000	6 300	0,78	*** 6212-2Z**	*** 6212-Z**
	110	22	55,3	36	1,53	–	4 000	0,78	*** 6212-2RS1**	*** 6212-RS1**
	110	28	52,7	36	1,53	–	4 000	0,97	**62212-2RS1**	**–**
	130	31	85,2	52	2,2	11 000	5 600	1,70	*** 6312-2Z**	*** 6312-Z**
	130	31	85,2	52	2,2	–	3 400	1,70	*** 6312-2RS1**	*** 6312-RS1**
	130	46	81,9	52	2,2	–	3 400	2,50	**62312-2RS1**	**–**
65	85	10	12,4	12,7	0,54	16 000	8 000	0,13	**61813-2RZ**	**–**
	85	10	12,4	12,7	0,54	–	4 500	0,13	**61813-2RS1**	**–**
	90	13	17,4	16	0,68	15 000	7 500	0,22	**61913-2RZ**	**–**
	90	13	17,4	16	0,68	–	4 300	0,22	**61913-2RS1**	**–**
	100	18	31,9	25	1,06	14 000	7 000	0,44	*** 6013-2Z**	*** 6013-Z**
	100	18	31,9	25	1,06	–	4 000	0,44	*** 6013-2RS1**	*** 6013-RS1**
	120	23	58,5	40,5	1,73	12 000	6 000	0,99	*** 6213-2Z**	*** 6213-Z**
	120	23	58,5	40,5	1,73	–	3 600	0,99	*** 6213-2RS1**	*** 6213-RS1**
	120	31	55,9	40,5	1,73	–	3 600	1,25	**62213-2RS1**	**–**
	140	33	97,5	60	2,5	10 000	5 300	2,10	*** 6313-2Z**	*** 6313-Z**
	140	33	97,5	60	2,5	–	3 200	2,10	*** 6313-2RS1**	*** 6313-RS1**
	140	48	92,3	60	2,5	–	3 200	3,00	**62313-2RS1**	**–**

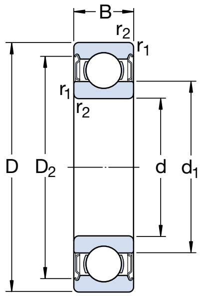
* SKF Explorer bearing
1) For bearings with only one shield or low-friction seal (Z, RZ, RSL), the limiting speeds for open bearings are valid

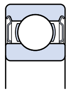


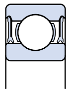


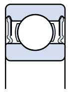
Dimensions					Abutment and fillet dimensions				Calculation factors	
d	d_1 ~	d_2 ~	D_2 ~	$r_{1,2}$ min	d_a min	d_a max	D_a max	r_a max	k_r	f_0
mm					mm				–	
60	65,6	–	74,5	0,3	62	–	76	0,3	0,015	17
	65,6	–	74,5	0,3	62	–	76	0,3	0,015	17
	68,2	–	79,2	1	64,6	–	80,4	1	0,02	16
	68,2	–	79,2	1	64,6	–	80,4	1	0,02	16
	71,3	–	86,5	1,1	66	–	89	1	0,025	16
	71,3	–	86,5	1,1	66	–	89	1	0,025	16
	71,3	–	86,5	1,1	66	–	89	1	0,025	16
	75,5	–	98	1,5	69	–	101	1,5	0,025	14
	75,5	–	98	1,5	69	–	101	1,5	0,025	14
	75,5	–	98	1,5	69	–	101	1,5	0,025	14
	81,9	–	112	2,1	72	–	118	2	0,03	13
	81,9	–	112	2,1	72	–	118	2	0,03	13
	81,9	–	112	2,1	72	–	118	2	0,03	13
65	71,6	–	80,5	0,6	68,2	–	81,8	0,6	0,015	17
	71,6	–	80,5	0,6	68,2	–	81,8	0,6	0,015	17
	73,2	–	84,2	1	69,6	–	85,4	1	0,02	17
	–	73,2	84,2	1	69,6	73	85,4	1	0,02	17
	76,3	–	91,5	1,1	71	–	94	1	0,025	16
	76,3	–	91,5	1,1	71	–	94	1	0,025	16
	83,3	–	106	1,5	74	–	111	1,5	0,025	15
	83,3	–	106	1,5	74	–	111	1,5	0,025	15
	83,3	–	106	1,5	74	–	111	1,5	0,025	15
	88,4	–	121	2,1	77	–	128	2	0,03	13
	88,4	–	121	2,1	77	–	128	2	0,03	13
	88,4	–	121	2,1	77	–	128	2	0,03	13

Sealed single row deep groove ball bearings
d **70 – 80** mm




2Z

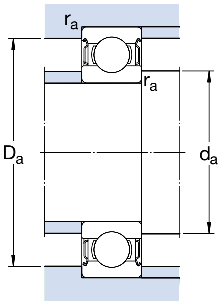

2RZ

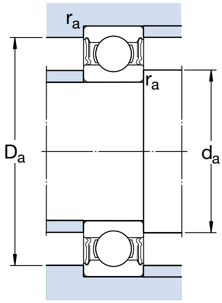

2RS1

Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting[1)] speed	Mass	Designations sealed both sides	sealed one side
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
70	90	10	12,4	13,2	0,56	15 000	7 500	0,14	**61814-2RZ**	–
	90	10	12,4	13,2	0,56	–	4 300	0,14	**61814-2RS1**	–
	100	16	23,8	21,2	0,9	14 000	7 000	0,35	**61914-2RZ**	–
	100	16	23,8	21,2	0,9	–	4 000	0,35	**61914-2RS1**	–
	110	20	39,7	31	1,32	13 000	6 300	0,60	* **6014-2Z**	* **6014-Z**
	110	20	39,7	31	1,32	–	3 600	0,60	* **6014-2RS1**	* **6014-RS1**
	125	24	63,7	45	1,9	11 000	5 600	1,10	* **6214-2Z**	* **6214-Z**
	125	24	63,7	45	1,9	–	3 400	1,10	* **6214-2RS1**	* **6214-RS1**
	125	31	60,5	45	1,9	–	3 400	1,30	**62214-2RS1**	–
	150	35	111	68	2,75	9 500	5 000	2,50	* **6314-2Z**	* **6314-Z**
	150	35	111	68	2,75	–	3 000	2,50	* **6314-2RS1**	* **6314-RS1**
	150	51	104	68	2,75	–	3 000	3,55	**62314-2RS1**	–
75	95	10	12,7	14,3	0,61	14 000	7 000	0,15	**61815-2RZ**	–
	95	10	12,7	14,3	0,61	–	4 000	0,15	**61815-2RS1**	–
	105	16	24,2	19,3	0,965	13 000	6 300	0,37	**61915-2RZ**	–
	105	16	24,2	19,3	0,965	–	3 600	0,37	**61915-2RS1**	–
	115	20	41,6	33,5	1,43	12 000	6 000	0,64	* **6015-2Z**	* **6015-Z**
	115	20	41,6	33,5	1,43	12 000	6 000	0,64	* **6015-2RZ**	* **6015-RZ**
	115	20	41,6	33,5	1,43	–	3 400	0,64	* **6015-2RS1**	* **6015-RS1**
	130	25	68,9	49	2,04	10 000	5 300	1,20	* **6215-2Z**	* **6215-Z**
	130	25	68,9	49	2,04	–	3 200	1,20	* **6215-2RS1**	* **6215-RS1**
	160	37	119	76,5	3	9 000	4 500	3,00	* **6315-2Z**	* **6315-Z**
	160	37	119	76,5	3	–	2 800	3,00	* **6315-2RS1**	* **6315-RS1**
80	100	10	13	15	0,64	13 000	6 300	0,15	**61816-2RZ**	–
	100	10	13	15	0,64	–	3 600	0,15	**61816-2RS1**	–
	110	16	25,1	20,4	1,02	12 000	6 000	0,40	**61916-2RZ**	–
	110	16	25,1	20,4	1,02	–	3 400	0,40	**61916-2RS1**	–
	125	22	49,4	40	1,66	11 000	5 600	0,85	* **6016-2Z**	* **6016-Z**
	125	22	49,4	40	1,66	–	3 200	0,85	* **6016-2RS1**	* **6016-RS1**
	140	26	72,8	55	2,2	9 500	4 800	1,40	* **6216-2Z**	* **6216-Z**
	140	26	72,8	55	2,2	–	3 000	1,40	* **6216-2RS1**	* **6216-RS1**
	170	39	130	86,5	3,25	8 500	4 300	3,60	* **6316-2Z**	* **6316-Z**
	170	39	130	86,5	3,25	–	2 600	3,60	* **6316-2RS1**	* **6316-RS1**

* SKF Explorer bearing
1) For bearings with only one shield or low-friction seal (Z, RZ, RSL), the limiting speeds for open bearings are valid

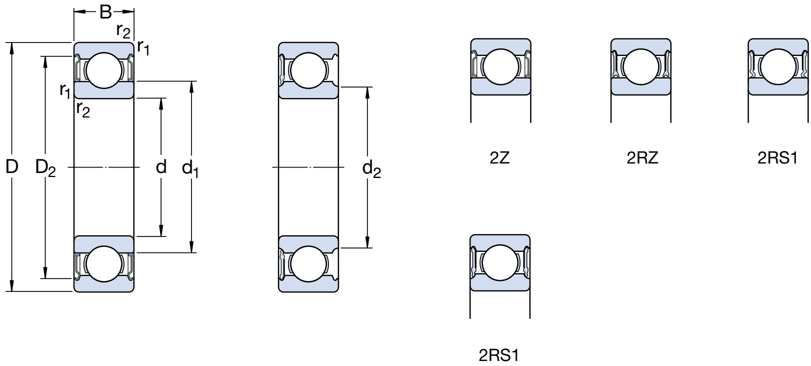
SKF





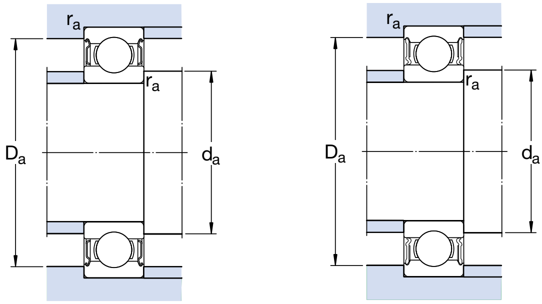
Dimensions				**Abutment and fillet dimensions**			**Calculation factors**	
d	d_1 ~	D_2 ~	$r_{1,2}$ min	d_a min	D_a max	r_a max	k_r	f_0
mm				mm			–	
70	76,6	85,5	0,6	73,2	86,8	0,6	0,015	17
	76,6	85,5	0,6	73,2	86,8	0,6	0,015	17
	79,7	93,3	1	74,6	95,4	1	0,02	16
	79,7	93,3	1	74,6	95,4	1	0,02	16
	82,9	99,9	1,1	76	104	1	0,025	16
	82,9	99,9	1,1	76	104	1	0,025	16
	87,1	111	1,5	79	116	1,5	0,025	15
	87,1	111	1,5	79	116	1,5	0,025	15
	87,1	111	1,5	79	116	1,5	0,025	15
	95	130	2,1	82	138	2	0,03	13
	95	130	2,1	82	138	2	0,03	13
	95	130	2,1	82	138	2	0,03	13
75	81,6	90,5	0,6	78,2	91,8	0,6	0,015	17
	81,6	90,5	0,6	78,2	91,8	0,6	0,015	17
	84,7	98,3	1	79,6	100	1	0,02	14
	84,7	98,3	1	79,6	100	1	0,02	14
	87,9	105	1,1	81	109	1	0,025	16
	87,9	105	1,1	81	109	1	0,025	16
	87,	105	1,1	81	109	1	0,025	16
	92,1	117	1,5	84	121	1,5	0,025	15
	92,1	117	1,5	84	121	1,5	0,025	15
	101	138	2,1	87	148	2	0,03	13
	101	138	2,1	87	148	2	0,03	13
80	86,6	95,5	0,6	83,2	96,8	0,6	0,015	17
	86,6	95,5	0,6	83,2	96,8	0,6	0,015	17
	89,8	102	1	84,6	105	1	0,02	14
	89,8	102	1	84,6	105	1	0,02	14
	94,4	114	1,1	86	119	1	0,025	16
	94,4	114	1,1	86	119	1	0,025	16
	101	127	2	91	129	2	0,025	15
	101	127	2	91	129	2	0,025	15
	108	147	2,1	92	158	2	0,03	13
	108	147	2,1	92	158	2	0,03	13

Sealed single row deep groove ball bearings
d **85 – 100** mm



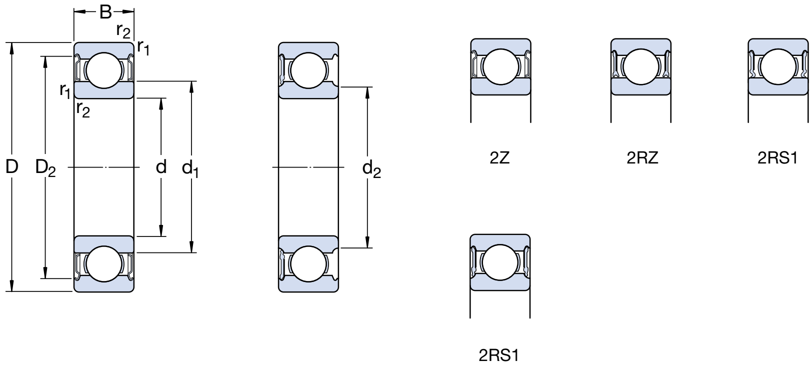
Principal dimensions			**Basic load ratings** dynamic	static	**Fatigue load limit**	**Speed ratings** Reference speed	Limiting[1)] speed	**Mass**	**Designations** sealed both sides	sealed one side
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
85	110	13	19,5	20,8	0,88	12 000	6 000	0,27	**61817-2RZ**	**–**
	110	13	19,5	20,8	0,88	–	3 400	0,27	**61817-2RS1**	**–**
	130	22	52	43	1,76	11 000	5 300	0,89	*** 6017-2Z**	*** 6017-Z**
	130	22	52	43	1,76	–	3 000	0,89	*** 6017-2RS1**	*** 6017-RS1**
	150	28	87,1	64	2,5	9 000	4 500	1,80	*** 6217-2Z**	*** 6217-Z**
	150	28	87,1	64	2,5	–	2 800	1,80	*** 6217-2RS1**	*** 6217-RS1**
	180	41	140	96,5	3,55	8 000	4 000	4,25	*** 6317-2Z**	*** 6317-Z**
	180	41	140	96,5	3,55	–	2 400	4,25	*** 6317-2RS1**	*** 6317-RS1**
90	115	13	19,5	22	0,915	11 000	5 600	0,28	**61818-2RZ**	**–**
	115	13	19,5	22	0,915	–	3 200	0,28	**61818-2RS1**	**–**
	140	24	60,5	50	1,96	10 000	5 000	1,15	*** 6018-2Z**	*** 6018-Z**
	140	24	60,5	50	1,96	–	2 800	1,15	*** 6018-2RS1**	*** 6018-RS1**
	160	30	101	73,5	2,8	8 500	4 300	2,15	*** 6218-2Z**	*** 6218-Z**
	160	30	101	73,5	2,8	–	2 600	2,15	*** 6218-2RS1**	*** 6218-RS1**
	190	43	151	108	3,8	7 500	3 800	4,90	*** 6318-2Z**	*** 6318-Z**
	190	43	151	108	3,8	–	2 400	4,90	*** 6318-2RS1**	*** 6318-RS1**
95	120	13	19,9	22,8	0,93	11 000	5 300	0,30	**61819-2RZ**	**–**
	120	13	19,9	22,8	0,93	–	3 000	0,30	**61819-2RS1**	**–**
	130	18	33,8	33,5	1,43	–	3 000	0,61	**61919-2RS1**	**–**
	145	24	63,7	54	2,08	9 500	4 800	1,20	*** 6019-2Z**	*** 6019-Z**
	145	24	63,7	54	2,08	–	2 800	1,20	*** 6019-2RS1**	*** 6019-RS1**
	170	32	114	81,5	3	8 000	4 000	2,60	*** 6219-2Z**	*** 6219-Z**
	170	32	114	81,5	3	–	2 400	2,60	*** 6219-2RS1**	*** 6219-RS1**
	200	45	159	118	4,15	7 000	3 600	5,65	*** 6319-2Z**	*** 6319-Z**
	200	45	159	118	4,15	–	2 200	5,65	*** 6319-2RS1**	*** 6319-RS1**
100	125	13	19,9	24	0,95	10 000	5 300	0,31	**61820-2RZ**	**–**
	125	13	19,9	24	0,95	–	3 000	0,31	**61820-2RS1**	**–**
	150	24	63,7	54	2,04	9 500	4 500	1,25	*** 6020-2Z**	*** 6020-Z**
	150	24	63,7	54	2,04	–	2 600	1,25	*** 6020-2RS1**	*** 6020-RS1**
	180	34	127	93	3,35	7 500	3 800	3,15	*** 6220-2Z**	*** 6220-Z**
	180	34	127	93	3,35	–	2 400	3,15	*** 6220-2RS1**	*** 6220-RS1**
	215	47	174	140	4,75	6 700	3 400	7,00	**6320-2Z**	**6320-Z**

* SKF Explorer bearing
1) For bearings with only one shield or low-friction seal (Z, RZ, RSL), the limiting speeds for open bearings are valid



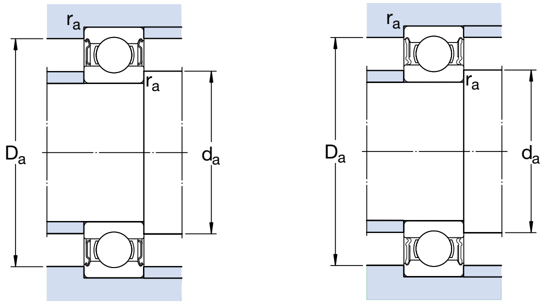
Dimensions					Abutment and fillet dimensions				Calculation factors	
d	d_1 ~	d_2 ~	D_2 ~	$r_{1,2}$ min	d_a min	d_a max	D_a max	r_a max	k_r	f_0
mm					mm				–	
85	93,2	–	104	1	89,6	–	105	1	0,015	17
	93,2	–	104	1	89,6	–	105	1	0,015	17
	99,4	–	119	1,1	92	–	123	1	0,025	16
	99,4	–	119	1,1	92	–	123	1	0,025	16
	106	–	134	2	94	–	141	2	0,025	15
	106	–	134	2	94	–	141	2	0,025	15
	115	–	155	3	99	–	166	2,5	0,03	13
	115	–	155	3	99	–	166	2,5	0,03	13
90	98,2	–	109	1	94,6	–	110	1	0,015	17
	98,2	–	109	1	94,6	–	110	1	0,015	17
	106	–	128	1,5	97	–	133	1,5	0,025	16
	106	–	128	1,5	97	–	133	1,5	0,025	16
	113	–	143	2	101	–	149	2	0,025	15
	–	106	143	2	101	105	149	2	0,025	15
	121	–	164	3	104	–	176	2,5	0,03	13
	121	–	164	3	104	–	176	2,5	0,03	13
95	103	–	114	1	99,6	–	115	1	0,015	17
	103	–	114	1	99,6	–	115	1	0,015	17
	106	–	122	1,1	101	–	124	1	0,02	17
	111	–	133	1,5	102	–	138	1,5	0,025	16
	110	–	133	1,5	102	–	138	1,5	0,025	16
	118	–	151	2,1	106	–	159	2	0,025	14
	–	112	151	2,1	106	111	159	2	0,025	14
	128	–	172	3	109	–	186	2,5	0,03	13
	–	121	172	3	109	120	186	2,5	0,03	13
100	108	–	119	1	105	–	120	1	0,015	17
	108	–	119	1	105	–	120	1	0,015	17
	116	–	138	1,5	107	–	143	1,5	0,025	16
	–	110	138	1,5	107	109	143	1,5	0,025	16
	125	–	160	2,1	111	–	169	2	0,025	14
	–	118	160	2,1	111	117	169	2	0,025	14
	136	–	184	3	114	–	201	2,5	0,03	13

Sealed single row deep groove ball bearings
d **105 – 160** mm



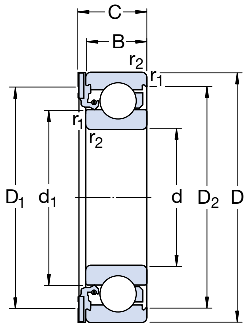
Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed[1]	Mass	Designations sealed both sides	sealed one side
d	D	B	C	C_0	P_u					
mm			kN		kN	r/min		kg	–	
105	130	13	20,8	19,6	1	10 000	5 000	0,32	**61821-2RZ**	**–**
	130	13	20,8	19,6	1	–	2 800	0,32	**61821-2RS1**	**–**
	160	26	76,1	65,5	2,4	8 500	4 300	1,60	* **6021-2Z**	* **6021-Z**
	160	26	76,1	65,5	2,4	–	2 400	1,60	* **6021-2RS1**	* **6021-RS1**
	190	36	140	104	3,65	7 000	3 600	3,70	* **6221-2Z**	* **6221-Z**
	190	36	140	104	3,65	–	2 200	3,70	* **6221-2RS1**	* **6221-RS1**
	225	49	182	153	5,1	6 300	3 200	8,25	**6321-2Z**	**6321-Z**
110	140	16	28,1	26	1,25	9 500	4 500	0,60	**61822-2RZ**	**–**
	140	16	28,1	26	1,25	–	2 600	0,60	**61822-2RS1**	**–**
	170	28	85,2	73,5	2,4	8 000	4 000	1,95	* **6022-2Z**	* **6022-Z**
	170	28	85,2	73,5	2,4	–	2 400	1,95	* **6022-2RS1**	* **6022-RS1**
	200	38	151	118	4	6 700	3 400	4,35	* **6222-2Z**	* **6222-Z**
	240	50	203	180	5,7	6 000	3 000	9,55	**6322-2ZTN9**	**6322-ZTN9**
120	150	16	29,1	28	1,29	8 500	4 300	0,65	**61824-2RZ**	**–**
	150	16	29,1	28	1,29	–	2 400	0,65	**61824-2RS1**	**–**
	180	28	88,4	80	2,75	7 500	3 800	2,05	* **6024-2Z**	* **6024-Z**
	180	28	88,4	80	2,75	–	2 200	2,05	* **6024-2RS1**	* **6024-RS1**
	215	40	146	118	3,9	6 300	3 200	5,15	**6224-2Z**	**6224-Z**
130	165	18	37,7	43	1,6	8 000	3 800	0,93	**61826-2RZ**	**–**
	165	18	37,7	43	1,6	–	2 200	0,93	**61826-2RS1**	**–**
	200	33	112	100	3,35	7 000	3 400	3,15	* **6026-2Z**	* **6026-Z**
	200	33	112	100	3,35	–	2 000	3,15	* **6026-2RS1**	* **6026-RS1**
	230	40	156	132	4,15	5 600	3 000	5,80	**6226-2Z**	**6226-Z**
140	175	18	39	46,5	1,66	7 500	3 600	0,99	**61828-2RZ**	**–**
	175	18	39	46,5	1,66	–	2 000	0,99	**61828-2RS1**	**–**
	210	33	111	108	3,45	6 700	3 200	3,35	**6028-2Z**	**6028-Z**
	210	33	111	108	3,45	–	1 800	3,35	**6028-2RS1**	**6028-RS1**
150	225	35	125	125	3,9	6 000	3 000	4,80	**6030-2Z**	**6030-Z**
	225	35	125	125	3,9	–	1 700	4,80	**6030-2RS1**	**6030-RS1**
160	240	38	143	143	4,3	5 600	2 800	5,90	**6032-2Z**	**6032-Z**
	240	38	143	143	4,3	–	1 600	5,90	**6032-2RS1**	**6032-RS1**

* SKF Explorer bearing
[1] For bearings with only one shield or low-friction seal (Z, RZ, RSL), the limiting speeds for open bearings are valid



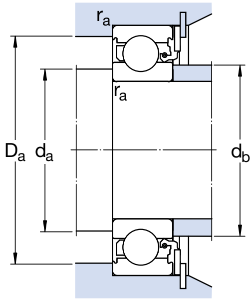
Dimensions					Abutment and fillet dimensions				Calculation factors	
d	d_1 ~	d_2 ~	D_2 ~	$r_{1,2}$ min	d_a min	d_a max	D_a max	r_a max	k_r	f_0
mm					mm				–	
105	112	–	124	1	110	–	125	1	0,015	13
	–	111	124	1	110	110	125	1	0,015	13
	123	–	147	2	116	–	149	2	0,025	16
	–	117	147	2	116	116	149	2	0,025	16
	131	–	167	2,1	117	–	178	2	0,025	14
	–	125	167	2,1	117	124	178	2	0,025	14
	141	–	193	3	119	–	211	2,5	0,03	13
110	119	–	134	1	115	–	135	1	0,015	14
	–	115	134	1	115	115	135	1	0,015	14
	129	–	155	2	119	–	161	2	0,025	16
	129	–	155	2	119	–	161	2	0,025	16
	138	–	177	2,1	122	–	188	2	0,025	14
	149	–	205	3	124	–	226	2,5	0,03	13
120	129	–	144	1	125	–	145	1	0,015	13
	–	125	144	1	125	125	145	1	0,015	13
	139	–	165	2	129	–	171	2	0,025	16
	–	133	165	2	129	132	171	2	0,025	16
	151	–	189	2,1	132	–	203	2	0,025	14
130	140	–	158	1,1	136	–	159	1	0,015	16
	–	137	158	1,1	136	136	159	1	0,015	16
	153	–	182	2	139	–	191	2	0,025	16
	153	–	182	2	139	–	191	2	0,025	16
	161	–	203	3	144	–	216	2,5	0,025	15
140	151	–	167	1,1	146	–	169	1	0,015	16
	–	148	167	1,1	146	147	169	1	0,015	16
	163	–	192	2	149	–	201	2	0,025	16
	–	156	192	2	149	155	201	2	0,025	16
150	174	–	205	2,1	160	–	215	2	0,025	16
	174	–	205	2,1	160	–	215	2	0,025	16
160	186	–	219	2,1	169	–	231	2	0,025	16
	–	179	219	2,1	169	178	231	2	0,025	16

ICOS™ oil sealed bearing units
d **12 – 30** mm



Principal dimensions				**Basic load ratings** dynamic	static	**Fatigue load limit**	**Limiting speed**	**Mass**	**Designation**
d	D	B	C	C	C_0	P_u			
mm				kN		kN	r/min	kg	–
12	32	10	12,6	7,28	3,1	0,132	14 000	0,041	* **ICOS-D1B01-TN9**
15	35	11	13,2	8,06	3,75	0,16	12 000	0,048	* **ICOS-D1B02-TN9**
17	40	12	14,2	9,95	4,75	0,2	11 000	0,071	* **ICOS-D1B03-TN9**
20	47	14	16,2	13,5	6,55	0,28	9 300	0,11	* **ICOS-D1B04-TN9**
25	52	15	17,2	14,8	7,8	0,335	7 700	0,14	* **ICOS-D1B05-TN9**
30	62	16	19,4	20,3	11,2	0,475	6 500	0,22	* **ICOS-D1B06-TN9**

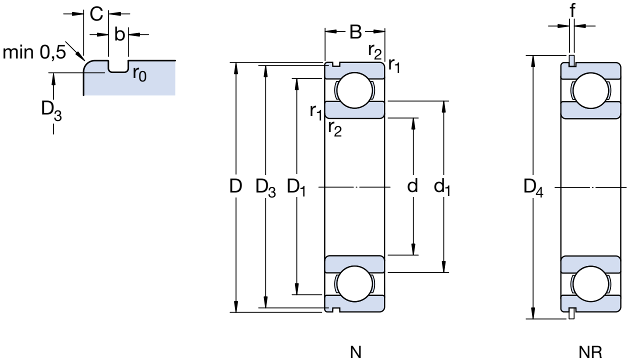
* SKF Explorer bearing



Dimensions					Abutment and fillet dimensions				Calculation factors	
d	d_1 ~	D_1 ~	D_2 ~	$r_{1,2}$ min	d_a min	d_b max	D_a max	r_a max	k_r	f_0
mm					mm				–	
12	18,4	–[1)]	27,4	0,6	16,2	18	27,8	0,6	0,025	12
15	21,7	30,8	30,4	0,6	19,2	21,5	30,8	0,6	0,025	13
17	24,5	35,6	35	0,6	21,2	24	35,8	0,6	0,025	13
20	28,8	42	40,6	1	25,6	28,5	41,4	1	0,025	13
25	34,3	47	46,3	1	30,6	34	46,4	1	0,025	14
30	40,3	55,6	54,1	1	35,6	40	56,4	1	0,025	14

1) Full rubber cross section

Single row deep groove ball bearings with snap ring groove
d **10 – 45** mm



Principal dimensions d	D	B	Basic load ratings dynamic C	static C_0	Fatigue load limit P_u	Speed ratings Reference speed	Limiting speed	Mass	Designations Bearing with snap ring groove	snap ring groove and snap ring	Snap ring
mm			kN		kN	r/min		kg	–		
10	30	9	5,4	2,36	0,1	56 000	34 000	0,032	* **6200 N**	* **6200 NR**	SP 30
12	32	10	7,28	3,1	0,132	50 000	32 000	0,037	* **6201 N**	* **6201 NR**	SP 32
15	35	11	8,06	3,75	0,16	43 000	28 000	0,045	* **6202 N**	* **6202 NR**	SP 35
17	40	12	9,95	4,75	0,2	38 000	24 000	0,065	* **6203 N**	* **6203 NR**	SP 40
	47	14	14,3	6,55	0,275	34 000	22 000	0,12	* **6303 N**	* **6303 NR**	SP 47
20	42	12	9,5	5	0,212	38 000	24 000	0,069	* **6004 N**	* **6004 NR**	SP 42
	47	14	13,5	6,55	0,28	32 000	20 000	0,11	* **6204 N**	* **6204 NR**	SP 47
	52	15	16,8	7,8	0,335	30 000	19 000	0,14	* **6304 N**	* **6304 NR**	SP 52
25	47	12	11,9	6,55	0,275	32 000	20 000	0,08	* **6005 N**	* **6005 NR**	SP 47
	52	15	14,8	7,8	0,335	28 000	18 000	0,13	* **6205 N**	* **6205 NR**	SP 52
	62	17	23,4	11,6	0,49	24 000	16 000	0,23	* **6305 N**	* **6305 NR**	SP 62
30	55	13	13,8	8,3	0,355	28 000	17 000	0,12	* **6006 N**	* **6006 NR**	SP 55
	62	16	20,3	11,2	0,475	24 000	15 000	0,20	* **6206 N**	* **6206 NR**	SP 62
	72	19	29,6	16	0,67	20 000	13 000	0,35	* **6306 N**	* **6306 NR**	SP 72
35	62	14	16,8	10,2	0,44	24 000	15 000	0,16	* **6007 N**	* **6007 NR**	SP 62
	72	17	27	15,3	0,655	20 000	13 000	0,29	* **6207 N**	* **6207 NR**	SP 72
	80	21	35,1	19	0,815	19 000	12 000	0,46	* **6307 N**	* **6307 NR**	SP 80
	100	25	55,3	31	1,29	16 000	10 000	0,95	**6407 N**	**6407 NR**	SP 100
40	68	15	17,8	11,6	0,49	22 000	14 000	0,19	* **6008 N**	* **6008 NR**	SP 68
	80	18	32,5	19	0,8	18 000	11 000	0,37	* **6208 N**	* **6208 NR**	SP 80
	90	23	42,3	24	1,02	17 000	11 000	0,63	* **6308 N**	* **6308 NR**	SP 90
	110	27	63,7	36,5	1,53	14 000	9 000	1,25	**6408 N**	**6408 NR**	SP 110
45	75	16	22,1	14,6	0,64	20 000	12 000	0,25	* **6009 N**	* **6009 NR**	SP 75
	85	19	35,1	21,6	0,915	17 000	11 000	0,41	* **6209 N**	* **6209 NR**	SP 85
	100	25	55,3	31,5	1,34	15 000	9 500	0,83	* **6309 N**	* **6309 NR**	SP 100
	120	29	76,1	45	1,9	13 000	8 500	1,55	**6409 N**	**6409 NR**	SP 120

* SKF Explorer bearing