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2003 - 02

The SKF Group – a worldwide corporation

SKF is an international industrial Group operating in more than 130 countries and is world leader in bearings.

The company was founded in 1907 following the invention of the self-aligning ball bearing by Sven Wingquist and, after only a few years, SKF began to expand all over the world.

Today, SKF has some 40 000 employees and around 80 manufacturing facilities spread throughout the world. An international sales network includes a large number of sales companies and some 7 000 distributors. Worldwide availability of SKF products is supported by a comprehensive technical advisory service.

The key to success has been a consistent emphasis on maintaining the highest quality of its products and services. Continuous investment in research and development has also played a vital role, resulting in many examples of epoch-making innovations.

The business of the Group consists of bearings, seals, special steels and a comprehensive range of other high-tech industrial components. The experience gained in these various fields provides SKF with the essential knowledge and expertise required in order to provide the customers with the most advanced engineering products and efficient service.



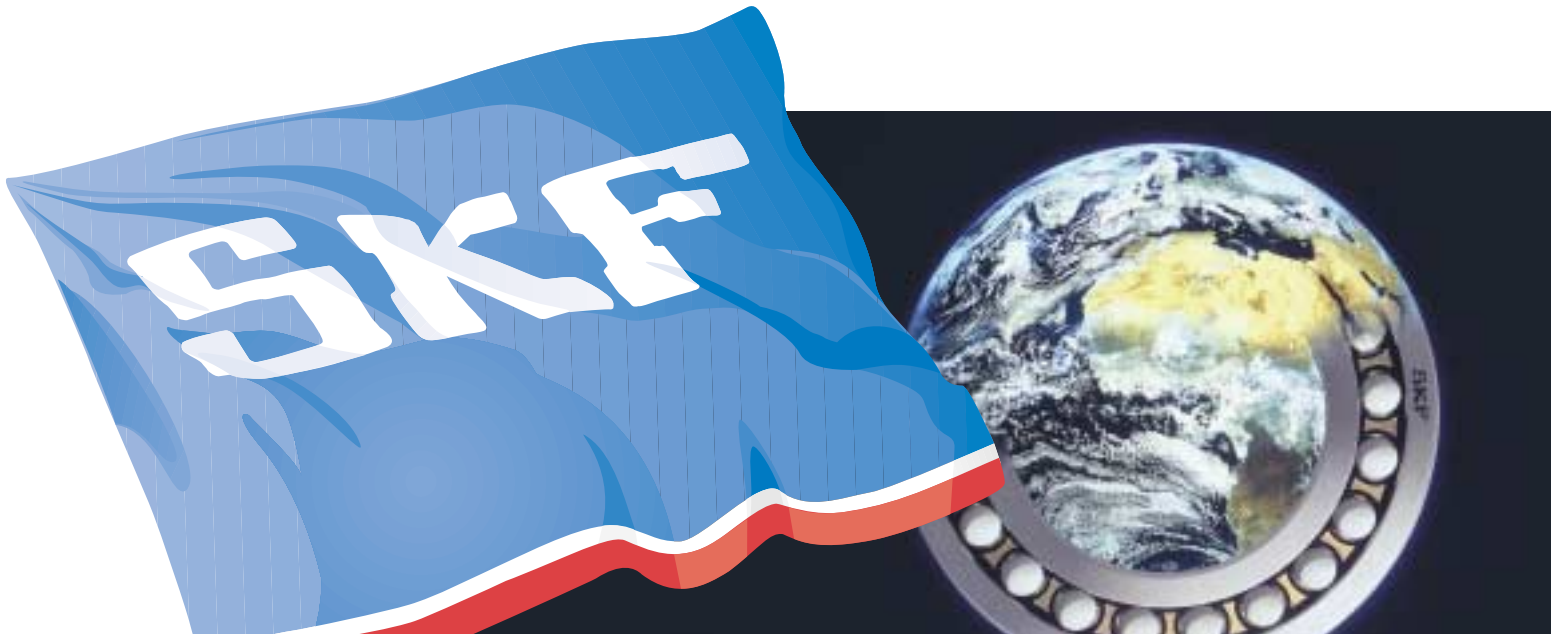
The SKF Group is the first major bearing manufacturer to have been granted approval according to ISO 14001, the international standard for environmental management systems. The certificate is the most comprehensive of its kind and covers more than 60 SKF production units in 17 countries.



The SKF Engineering & Research Centre is situated just outside Utrecht in The Netherlands. In an area of 17 000 square metres (185 000 sq.ft) some 150 scientists, engineers and support staff are engaged in the further improvement of bearing performance. They are developing technologies aimed at achieving better materials, better designs, better lubricants and better seals – together leading to an even better understanding of the operation of a bearing in its application. This is also where the SKF Life Theory was evolved, enabling the design of bearings that are even more compact and offer even longer operational life.



SKF has developed the Channel concept in factories all over the world. This drastically reduces the lead-time from raw material to end product as well as work in progress and finished goods in stock. The concept enables faster and smoother information flow, eliminates bottlenecks and bypasses unnecessary steps in production. The Channel team members have the knowledge and commitment needed to share the responsibility for fulfilling objectives in areas such as quality, delivery time, production flow etc.



SKF manufactures ball bearings, roller bearings and plain bearings. The smallest are just a few millimetres (a fraction of an inch) in diameter, the largest several metres. SKF also manufactures bearing and oil seals that prevent dirt from entering and lubricant from leaking out. SKF's subsidiaries CR and RFT S.p.A. are among the world's largest producers of seals.

Wherever there is rotation, there is a need for some form of bearing. The function of a rolling bearing is to reduce or eliminate the friction between a fixed and a moving surface and to carry a load. The life of a rolling bearing must be compatible with the life of the application in which it is mounted.

The SKF standard product range comprises more than 22 000 variants, covering all the principal bearing types.

Made by SKF stands for excellence. It symbolises our consistent endeavour to achieve total quality in everything we do. For those who use our products, “Made by SKF” implies three main benefits.

Reliability – thanks to modern, efficient products, based on our worldwide application know-how, optimised materials, forward-looking designs and the most advanced production techniques.

Cost effectiveness – resulting from the favourable ratio between our product quality plus service facilities, and the purchase price of the product.

Market lead – which you can achieve by taking advantage of our products and services. Increased operating time and reduced downtime, as well as improved output and product quality are the key to a successful partnership.

Foreword

Machine tool applications require superior performance from the bearings used to support spindles and precision ball screws. They must exhibit this in terms of speed capability, system temperature stability, rigidity, accuracy and noise level; such characteristics are rarely met by bearings for general-purpose applications. Therefore, SKF produces special high-precision bearings designed to satisfy the most demanding requirements in the machine tool environment.

This catalogue presents the current range of SKF high-precision bearings and related products.

The data in this catalogue are based on current production. However, design refinement for continuous improvement in either manufacturing or performance may result in changes.

The basic load ratings have been calculated in accordance with the latest specifications. Earlier catalogues, in which the data varies from that given here, are rendered invalid. SKF reserves the right to make changes necessitated by technological developments.

In accordance with ISO (International Standards Organisation) Standard 1000: 1992, SI units (Système International d’Unités) are used in this catalogue.

Almost 100 years of worldwide experience

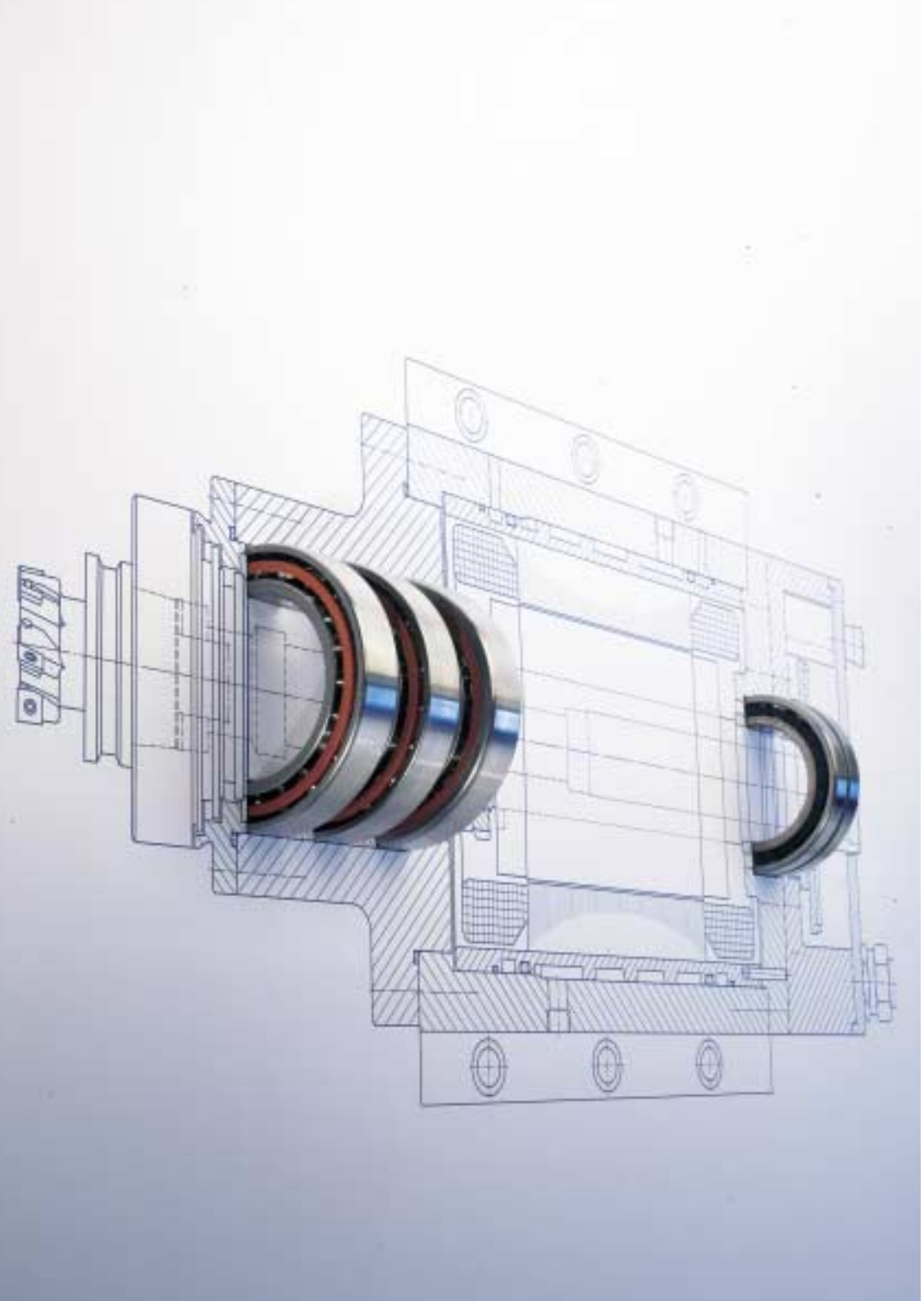
For almost 100 years SKF has held a leading position in all major industrial fields where rolling bearings are used. SKF not only supplies a wide range of bearings but also has broad experience in application engineering, and is at the forefront of systems design. This background comes from the partnerships developed over the years with leading firms in the machine tool industry. The more complex the problems, the more important it is to make use of SKF know-how in the manufacture and application of high-precision rolling bearings.

Where to find information

The bearing tables and technical data on the following pages provide information for bearing selection and application design. More details on bearing technology and on other related products can be found in specific publications, the SKF General Catalogue or the SKF Interactive Engineering Catalogue, available as a CD-ROM or online under **www.skf.com**.

This site also contains further information on the SKF Group, its products, services and contacts.





Principles of bearing selection and application

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Selection of bearing type

A wide range of high-precision bearings

It is not always possible to find an appropriate solution to application design and problems using standard bearings for general machinery. This is particularly true in machine tool applications where requirements are tougher.

SKF has developed a very comprehensive range of high-precision bearings for machine tool spindles, and other applications where high demands are placed on accuracy and speed capability. Each of these bearing types has specific features that make it particularly suitable for specific applications.

Internal design is state-of-the art, and differs in many ways from that of a standard bearing. The design has been optimised for outstanding speed capability in combination with the highest possible stiffness. When designing a bearing arrangement it is necessary to consider a number of factors, i.e.:

- Accuracy
- Available space
- Load
- Required system rigidity
- Accommodation of axial displacements
- Speed
- Heat generation.

Depending on the application, one or more of these factors will have a dominant influence. It is therefore, not possible to set down general rules for the selection of bearing type or bearing series. Where demands for high-precision and productivity are exceptionally high, SKF provides a comprehensive range of hybrid (i.e. bearings with steel rings and ceramic rolling elements) high-precision angular contact ball bearings and cylindrical roller bearings for machine tool spindles.

To make bearing selection easier, the properties of the different bearing types are described in the following.

Bearing materials

All-steel bearings

The performance and reliability of rolling bearings are determined to a large degree by the materials from which the bearing components are made.

Steels used for high-precision bearing rings and rolling elements are capable of being adequately hardened and have high fatigue strength and wear resistance. Moreover they have the structural and dimensional stability to satisfy the spindle operating temperatures.

SKF high-precision bearing rings and rolling elements are generally made of through-hardened carbon chromium steel containing approximately 1 % carbon and 1,5 % chromium.

Hybrid bearings

Machine tools, especially machining centres, can have higher machining efficiencies and higher machining accuracies when operated at higher speeds. For this reason, there have been ever-greater demands for higher speed performance of their spindle bearings. If the performance required is close to all-steel bearing limits, or if higher rigidity or longer life are needed, hybrid high-precision bearings can be used. Hybrid high-precision bearings consist of outer and inner rings made of bearing steel and ceramic (silicon nitride Si_3N_4) rolling elements, having density as light as 40 % of bearing steel but yet having a high rigidity (→ **fig 1**).

Comparison of performance and cost for different bearing design (material)

Fig 1

Bearing material	Performances	Cost	Typical applications
 Full steel			Machine tools. Precision equipments. High speed rolling mills. High speed electric motors, etc.
 Hybrids			Machine tools. Precision equipments. Turbochargers. High speed electric motors, etc.

Accuracy

Running accuracy

The running accuracy of a bearing arrangement is governed by the accuracy of all the component parts of the arrangement. Where bearings are concerned, this is primarily determined by the accuracy of form and position of the raceways on the bearing rings. When selecting the appropriate tolerance class for a particular bearing, the maximum radial runout of the inner ring (K_{ia}) is usually the determining factor for most applications.

To facilitate comparison, **Diagram 1** gives relative values of the radial runout for the different tolerance classes comparing bearings with bore diameters.

The different standards ABMA and ISO are also compared in **Table 1**.

Most SKF precision bearings are manufactured to tolerance class P4A, P4C, and SP. P4A is the standard tolerance class for angular contact ball bearings; SP the standard for cylindrical roller bearings and double direction angular contact thrust ball bearings, and P4C the standard class for single direction angular contact thrust ball bearings of the BTM series. For bearing arrangements where this standard precision is inadequate, angular contact ball bearings made to tolerance class PA9A, and cylindrical roller bearings and double direction angular contact thrust ball bearings made to tolerance class UP specifications can be supplied (→ **Table 2**).

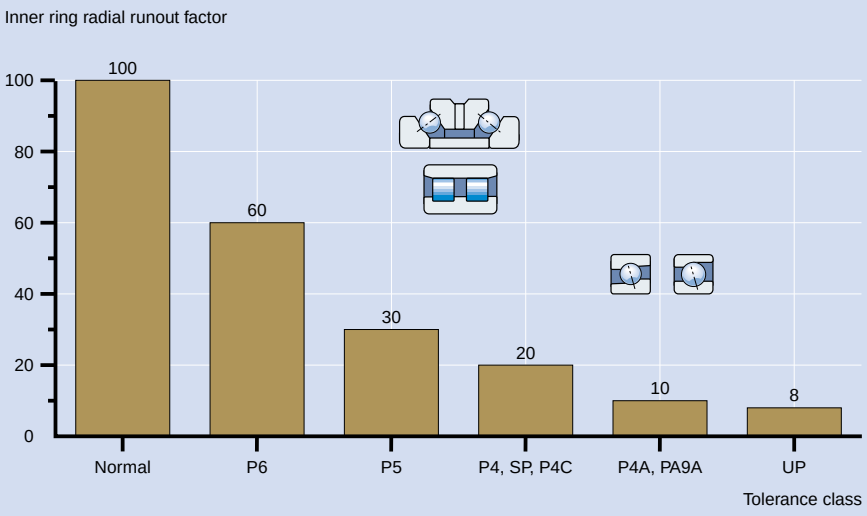
Normally, the maximum values of K_{ia} given in the table are much higher than the actual values. This means for example, that if bearings with class SP tolerances are used, depending on the bearing size and whether bearings are properly mounted, a running accuracy of less than 2 µm can be achieved.

Dimensional accuracy

The dimensional accuracy of a bearing is important with respect to the fit between bearing ring and shaft or housing. As the fit influences the clearance or preload of mounted bearings, the tolerances for the bearing and its seating lie within narrow limits. Where cylindrical roller bearings with a tapered bore are concerned, slightly larger dimensional deviations are permissible than for example, angular contact ball bearings with similar running accuracy. This is because the clearance or preload of the bearing is determined by driving up the inner ring on its tapered seating.

Comparison of accuracy of different tolerance classes

Diagram 1



Comparison of different standards

Table 1

ABMA Tolerance class	ISO Tolerance class	SKF Standard
ABEC 9	2	PA9A
ABEC 7	4	P4
ABEC 5	5	P5

SKF tolerance classes for high-precision bearings

Table 2

SKF Tolerance Class	Boundary dimensions ISO, ABMA	Running accuracy ISO, ABMA
SP	ISO 5, ABEC 5	ISO 4, ABEC 7
UP	ISO 4, ABEC 7	ISO 2, ABEC 9
P4A	ISO 4, ABEC 7	ISO 2, ABEC 9[1)]
P4C	ISO 4, ABEC 7	ISO 4, ABEC 7
PA9A	ISO 2, ABEC 9	ISO 2, ABEC 9

[1)] Up to 120 mm bore diameter, for larger sizes, ABEC 7 or better

Rigidity

The rigidity of a spindle/bearing system is of particular importance in machine tool applications as the magnitude of the deflection under load determines the productivity and machining accuracy of a machine tool. Bearing stiffness influences the stiffness of a bearing arrangement and thus the stiffness of the complete spindle/bearing system. However, bearings alone cannot be the only reason for high or low rigidity: when checking the behaviour of a complete system, bending of the spindle itself, position and number of support bearings and tool overhang may also have a great influence.

The stiffness of a bearing depends on its type and size, the most important criteria being:

- type of rolling elements (rollers or balls)
- number and size of the rolling elements, and
- contact angle.

Because of the much larger contact area between rolling elements and raceways in a roller bearing than in a ball bearing, roller bearing stiffness is much higher than ball bearing stiffness. The number of rolling elements has a greater influence on bearing stiffness, than the size of the rolling elements. Because of this, high-precision bearings almost always have the dimensions of the light Diameter Series 0 or 9. Where high radial stiffness is required, bearings having the smallest possible contact angle should be used. Conversely, where high axial stiffness is called for, the contact angle should be as large as possible.

Speed

There is a limit to the speed at which rolling bearings can be operated. Generally, it is the permitted operating temperature with respect to the lubricant being used, or to the material of the bearing components that sets the limit. The speed at which this limiting bearing 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/preload, all play a part in determining speed capability.

Available space

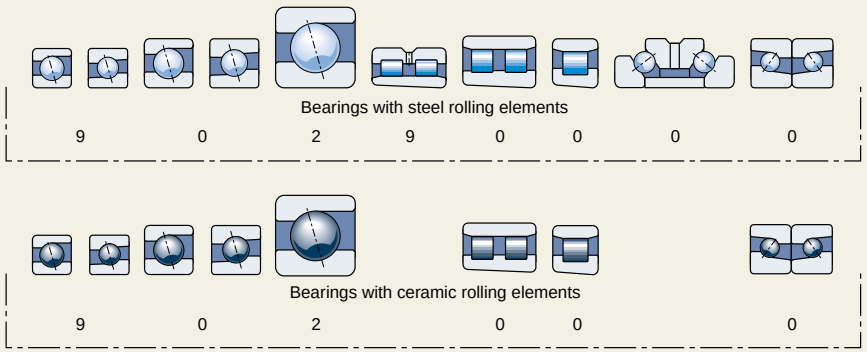
High-precision bearing arrangements generally call for bearings with a low cross-section because of the space available, together with the high requirements in respect of stiffness and running accuracy of the arrangement. These bearings generally have a large number of rolling elements and consequently a high stiffness. They also enable relatively large diameter spindles to be used for a given housing bore diameter, and therefore exhibit all the advantages important for the stiffness and running accuracy of a bearing arrangement, e.g. a spindle bearing arrangement. Almost all of the angular contact ball bearings, cylindrical roller bearings and angular contact thrust ball bearings used for machine tool applications belong to the ISO Diameter Series 9 and 0.

By selecting suitable combinations of bearings it is thus possible to achieve an optimum bearing arrangement for specific requirements within the same radial space. For bearing arrangements where less radial space is available, angular contact ball bearings and cylindrical roller bearings belonging to ISO Diameter Series 9 can be used. Angular contact ball bearings to the ISO Diameter Series 2, despite being rarely chosen for new designs, are still common in existing applications.

To illustrate the space required, **fig 2** shows cross-sections of the most common machine tool spindle bearings belonging to the different Diameter Series.

High-precision bearings cross section for different Diameter Series

Fig 2



Loads

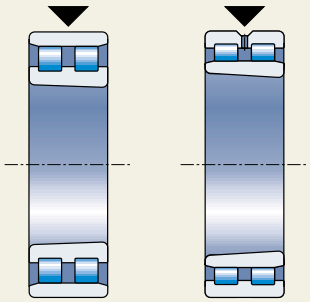
In machine tools – the main application for high-precision bearings – the load carrying capacity of a bearing is usually of much less importance when determining bearing size, than in engineering applications in general. Other criteria such as stiffness, size of the requisite bore in the spindle, machining speeds and accuracy are the decisive factors.

When selecting the bearing type for a given bearing arrangement, however, the magnitude and direction of action of the load play an important part.

As a general rule, roller bearings can carry heavier loads than ball bearings having the same boundary dimensions (➔ **Table 3**).

Radial load

Fig 3



Main features, speed rating and size range for different bearing design ➤

Codes:

A = Angular Contact Ball Bearing, Single direction;
AT2 = Angular Contact Thrust Ball Bearing, Double direction;
C1 = Cylindrical Roller Bearing, Single row;
C2 = Cylindrical Roller Bearing, Double row;
Hybrid = Bearing with steel rings and silicon nitride (ceramic) rolling elements

Table 3

Diametric Series ISO	SKF Series designation	Bearing Code	Contact angle α	Shaft diameters from	up to and incl.	Loading	Precision SKF tolerance Class	Speed rating n d_m
–	–	–	deg	mm	mm	–	–	10^6 mm/min
9	719 ACX	A	25	10	30	combined	P4A	1,5
	719 ACD	A	25	35	220	combined	P4A	1,5
	719 ACE	A	25	20	120	combined	P4A	2,2
	719 CX	A	15	10	30	combined	P4A	1,8
	719 CD	A	15	10	220	combined	P4A	1,8
	719 CE	A	15	20	120	combined	P4A	2,4
	719 ACX/HC	Hybrid A	25	10	30	combined	P4A	1,8
	719 ACD/HC	Hybrid A	25	35	140	combined	P4A	1,8
	719 ACE/HC	Hybrid A	25	20	120	combined	P4A	2,5
	719 CX/HC	Hybrid A	15	10	30	combined	P4A	2,1
	719 CD/HC	Hybrid A	15	35	140	combined	P4A	2,1
	719 CE/HC	Hybrid A	15	20	120	combined	P4A	2,8
	NNU 49 B/W33	C2	0	100	240	radial only	SP	0,8
	NNU 49 BK/W33	C2	0	100	240	radial only	SP	0,8
0	70 ACX	A	25	10	30	combined	P4A	1,5
	70 ACD	A	25	35	240	combined	P4A	1,5
	70 ACE	A	25	20	100	combined	P4A	2,2
	70 CX	A	15	8	30	combined	P4A	1,8
	70 CD	A	15	35	240	combined	P4A	1,8
	70 CE	A	15	20	100	combined	P4A	2,4
	70 ACX/HC	Hybrid A	25	10	30	combined	P4A	1,8
	70 ACD/HC	Hybrid A	25	35	100	combined	P4A	1,8
	70 ACE/HC	Hybrid A	25	20	100	combined	P4A	2,5
	70 CX/HC	Hybrid A	15	8	30	combined	P4A	2,1
	70 CD/HC	Hybrid A	15	35	100	combined	P4A	2,1
	70 CE/HC	Hybrid A	15	20	100	combined	P4A	2,8
	N 10 KTN(9)	C1	0	40	120	radial only	SP	1
	N 10 KTNHA	C1	0	40	120	radial only	SP	1,8
	N 10 KTN(9)/HC	Hybrid C1	0	40	120	radial only	SP	1,15
	N 10 KTNHA/HC	Hybrid C1	0	40	120	radial only	SP	2
	NN 30	C2	0	25	130	radial only	SP	0,8
	NN 30 K	C2	0	25	280	radial only	SP	0,8
	2344(00) B	AT2	60	35	200	thrust only	SP	0,7
	BTM – A	AT2	30	60	130	thrust only	P4C	1,05
	BTM – B	AT2	40	60	130	thrust only	P4C	0,9
2	72 ACX	A	25	10	25	combined	P4A	1,5
	72 ACD	A	25	30	120	combined	P4A	1,5
	72 CX	A	15	10	25	combined	P4A	1,8
	72 CD	A	15	30	120	combined	P4A	1,8
	72 ACX/HC	Hybrid A	25	10	25	combined	P4A	1,8
	72 ACD/HC	Hybrid A	25	30	60	combined	P4A	1,8
	72 CX/HC	Hybrid A	15	10	25	combined	P4A	2,1
	72 CD/HC	Hybrid A	15	30	60	combined	P4A	2,1
	BSA 2	AT1	60	12	35	combined	P4	0,5
3	BSA 3	AT1	60	25	40	combined	P4	0,5
–	BSD	AT1	60	20	50	combined	Special	0,5

Radial loads
Purely radial loads can be supported by cylindrical roller bearings having one ring without flanges (N, NN and NNU types **figs** 3 **page 14 and** 4). All other radial bearings can carry some axial load in addition to radial loads; see **Combined load page16**.

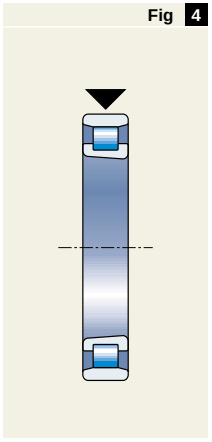
Axial loads
Double direction angular contact thrust ball bearings series 2344(00) and BTM are designed to take loads that are purely axial (→ **fig** 5). For large bearing arrangements, or those subjected to particularly heavy axial loads, special single direction thrust ball bearings or cylindrical roller thrust bearings are recommended. Please consult the SKF application engineering service for more details.

Combined loads
Combined loads consist of a radial load and a simultaneously acting axial load. This type of load can be accommodated by bearings having raceways in the inner and outer rings, situated at an angle to the bearing axis. Where high-precision bearings are concerned, the angular contact ball bearings (series 70, 719 and 72) and the single direction angular contact thrust ball bearings (series BSD and BSA) fall into this category.

The ability to carry axial loads is determined by the angle of contact; the larger the angle, the greater the axial load, which can be accommodated (→ **figs** 6 **and** 7).

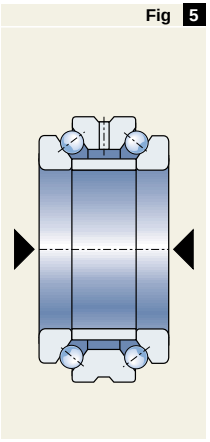
Radial load

Fig 4



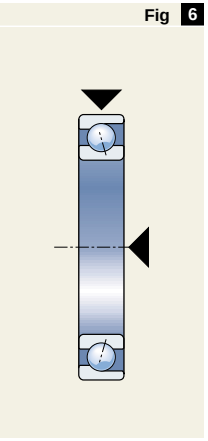
Thrust load

Fig 5



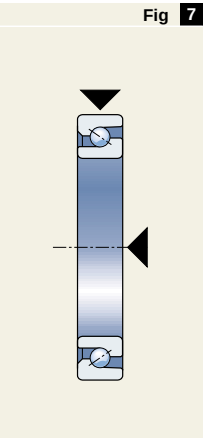
Combined loads

Fig 6



Combined loads

Fig 7



Load carrying capacity and life

General

All general information on life, basic load rating, and life equations described in the SKF General Catalogue or the SKF Interactive Engineering Catalogue also applies to high-precision bearings.

In general-machinery applications, the size of bearing to be used is initially selected on the basis of its load carrying capacity in relation to the loads to be carried, and the requirements regarding life and reliability.

For machine tool spindles, bearing size is nearly always determined by criteria such as stiffness of the system, fixed dimensions for the tool holder, or the spindle bore. The bearings selected according to such criteria give arrangements that are often required to have a very long life.

For high-precision bearings, determining the load to which a bearing will be subjected is particularly complex as it involves many influencing factors. SKF has therefore developed special computer programs for the calculation of indeterminate spindle bearing systems. Contact SKF for assistance in determining the bearing loads and in designing an optimum bearing arrangement.

Rating lives

Basic rating life equation
For calculation of simple bearing systems, the classic ISO equation for basic rating life can be used:

$$L_{10} = (C/P)^p$$

where
L_{10} = basic rating life, millions of revolutions
C = basic dynamic load rating, N
P = equivalent dynamic bearing load, N
p = exponent of the life equation (3 for ball bearings, 10/3 for roller bearings)

Adjusted rating life equation
As there are many other factors influencing bearing life besides load, ISO introduced an adjusted life equation

$$L_{na} = a_1 a_2 a_3 (C/P)^p$$

or simply $L_{na} = a_1 a_2 a_3 L_{10}$

where
L_{na} = adjusted rating life, millions of revolutions (the index n represents the difference between the requisite reliability and 100 %)
a_1 = life adjustment factor for reliability
a_2 = life adjustment factor for material
a_3 = life adjustment factor for operating conditions

SKF Life Theory – rating life equation.
Practical experience and modern research have shown that, under special conditions, SKF bearings attain a much longer life than predicted by standardised life calculation methods, particularly when loads are light. These special conditions apply when the rolling surfaces (raceways and rolling elements) are effectively separated by a lubricant film, and where the risk of the surfaces becoming damaged by contaminants are largely non-existent. In fact, under ideal conditions it is possible to speak of infinite life.

The SKF Life Theory introduces the concept of a fatigue load limit P_u analogous to that used when calculating other machinery components. This fatigue load limit represents the load below which fatigue will not occur in the bearing under ideal conditions. Due to the complexity, a detailed description of the theory is beyond the scope of this publication. For further details, please consult SKF application engineering services.

Values of the P_u limit for precision bearings are given in the bearing tables. The adjusted life equation according to the SKF Life Theory is:

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

where:
L_{naa} = adjusted rating life to SKF Life Theory, millions of revolutions
L_{10} = basic rating life, millions of revolutions
a_1 = life adjustment factor for reliability
a_{SKF} = life adjustment factor based on SKF Life Theory

For details on the a_{SKF} life adjustment factor, please consult the SKF General Catalogue or the SKF Interactive Engineering Catalogue.

Rating life equation for hybrid bearings

When using hybrid bearings, the effect of the different rolling element material should also be taken into account.

Under the same external load, the stress in the contact area between a hybrid bearing ball and each raceway will be higher than in an all-steel bearing. This is due to the greater hardness and stiffness of the ceramic material compared with steel.

As ISO does not provide guidelines for calculating basic load ratings for hybrid bearings, SKF quotes the same load rating values for hybrid bearings as for all-steel bearings.

By introducing a factor a_{HC} into the life equations, the higher contact stress is taken into consideration.

$$L_{10\,(hybrid)} = a_{HC} \, (C/P)^p$$
$$L_{na\,(hybrid)} = a_1 \, a_{23} \, a_{HC} \, (C/P)^p$$
$$L_{naa\,(hybrid)} = a_1 \, a_{SKF} \, a_{HC} \, (C/P)^p$$

Recent results from practical experience and extensive laboratory testing indicate that the factor a_{HC} can be considered equal to 1 both for hybrid ball and roller bearings.

In general, experience shows that the service life of hybrid bearings is significantly longer than that of all-steel bearings under the conditions normally encountered in machine tool operations. Hybrid bearings are much less susceptible to wear, and lubrication conditions are generally superior to those in an all-steel bearing.

Load carrying capacity of bearing sets

The basic load ratings listed in the product tables for angular contact ball bearings and single direction angular contact thrust ball bearings apply to single bearings. The basic dynamic load rating for sets of bearings arranged back-to-back, face-to-face or in tandem, is obtained by multiplying the C value for a single bearing by

1,62 for sets comprising two bearings
2,16 for sets comprising three bearings
2,64 for sets comprising four bearings.

For the basic static load rating, values for bearing sets can be obtained by multiplying the single bearing value by the number of bearings in the set, viz. by 2, 3, or 4.

For single direction angular contact thrust ball bearings, refer to the product section, **Table 10 page 247**.

Equivalent bearing loads

Equivalent dynamic bearing load
Angular contact ball bearings are generally subjected to combined loads, i.e. radial and axial loads acting simultaneously. Preload forces are included in the axial load. In all such cases, it is necessary to calculate the equivalent dynamic bearing load that has the same influence on bearing life as the actual load to which the bearing is subjected. The equivalent dynamic bearing load can be obtained from the general equation:

$$P = XF_r + YF_a$$

where
P = equivalent dynamic bearing load, N
F_r = actual radial bearing load, N
F_a = actual axial bearing load, N
X = radial load factor for the bearing
Y = axial load factor for the bearing

All information necessary for calculating the equivalent dynamic bearing load for single bearings and sets of two bearings is given in the relevant product sections. The information does not apply to sets of three, or more bearings, as it cannot be assumed that the load is evenly distributed over the bearings, and reaction forces caused by spindle deflection cannot be ignored. Special SKF computer programs are used to calculate the equivalent dynamic loads of such bearing arrangements and other parameters, e.g. spindle deflection. Further information will be supplied on request.

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

$$P_0 = X_0F_r + Y_0F_a$$

where
P_0 = equivalent static bearing load, N
F_r = actual radial bearing load, N
F_a = actual axial bearing load, N
X_0 = radial load factor for the bearing
Y_0 = axial load factor for the bearing

All information and data necessary for the calculation of the equivalent static bearing load for single bearings and sets of two bearings is given in the relevant product sections. The information does not apply to sets of three, or more bearings, as it cannot be assumed that the load is evenly distributed over the bearings, and reaction forces caused by spindle deflection cannot be ignored. Further information will be supplied on request.

Rigidity

System rigidity

Bearing rigidity influences the rigidity of a bearing arrangement and thus the rigidity of the complete spindle/bearing system. It is of particular importance in machine tool applications, as the magnitude of the deflection under load determines the machining accuracy of a machine tool.

The rigidity of a bearing depends on its type and size, the most important criteria being:
- type of rolling elements (rollers or balls)
- number and size of rolling elements, and
- contact angle.

To enhance the rigidity of the bearing arrangement, or to increase running accuracy, bearings can be preloaded. These are two important reasons why machine tool spindles are almost always fitted with preloaded bearings or bearing sets.

Apart from the bearings, other components might influence the rigidity of a system i.e. bending of the spindle itself, position and number of support bearings, and tool overhang. Thus, it is necessary to consider how different bearing arrangements behave and how they contribute to the system's overall rigidity.

The rigidity indexes given in **Table** 1 **page 22** are guideline values and must not be taken as tools for precise calculations of system rigidity.

General guidelines for designing a rigid spindle are:
- select the largest possible shaft diameter compatible with other restrictions
- minimise the distance between the front support position and the spindle nose
- bearing spacing (between rear and front supports) should be fairly short. As a guideline, a ratio l/d = 3 – 3,5 provides the best compromise, where l = distance between the rearmost bearing row and the first front side bearing row, and d = bore diameter of the front bearing.

Please consult the SKF application engineering service for advanced system analysis.

Table 1 shows the radial and axial stiffness of a 100 mm shaft diameter spindle where different bearing arrangements are compared.

Influence of a loose fit

In general, a machine component is supported in a locating and a non-locating bearing.

Non-locating bearings can be displaced axially and so prevent the bearings from being subjected to extra stresses, e.g. as a result of changes in shaft length caused by thermal expansion.

If non-separable bearings, e.g. angular contact ball bearings, are used as non-locating bearings, one of the bearing rings must have a loose fit. Usually the non-rotating outer ring is given a loose fit in the housing, although this has a negative influence on the total stiffness of the bearing arrangement.

Table 1

Relative stiffness of spindles equipped with different bearing arrangements

Bearing arrangement		Rigidity index	
Rear side	Tool side	Radial	Axial
NN 30 KTN	**2344(00) + NN 30 KTN**	100	100
NN 30 KTN	**BTM – B + NN 30 KTN**	100	67
NN 30 KTN	**BTM – A + NN 30 KTN**	100	44
N 10 KTN	**BTM – A + NN 30 KTN**	98	44
N 10 KTN	**70 CD/QBCB**	76	45
N 10 KTNHA	**719 CE QBCB**	74	34
NN 30 KTN	**719 ACD/TBTB**	67	75
719 CD/DBA	**719 ACD/TBTB**	67	75
NN 30 KTN	**70 CD/TBTB**	67	36
70 CD/DBA	**70 CD/TBTB**	67	36
70 CD/DBA	**70 ACD/TBTB**	63	74
N 10 KTNHA	**70 CE QBCB**	63	30
70 CE/HCDT	**70 CE/HCDT**	55	33
70 CD/DBA	**70 ACD/DBB**	54	48
719 CE/HCDBA	**719 ACE/HCTBTA**	49	48
70 CE/DT	**70 CD/DT**	49	30
N 10 KTNHA	**70 CE/DBB**	47	16
719 CE/DBA	**719 ACE/TBTA**	40	42
70 CD/HCDBA	**70 ACE/HCDBA**	39	28
N 10 KTNHA/HC5	**70 ACE/HCDBA**	38	28
70 CE/DBA	**70 ACE/TBTA**	37	41
N 10 KTNHA	**719 ACE/DBA**	36	26
719 CE/DBA	**719 ACE/DBA**	36	26
70 CE/DBA	**70 ACE/DBA**	34	25
N 10 KTNHA	**70 ACE/DBA**	34	25

The table above specifies the relative stiffness, calculated on a real spindle design having 100 mm shaft diameter at the tool side and 90 mm shaft diameter at the rear side. It must be taken into account that the shaft length (i.e. the distance from the first and last bearing centres) has an important influence on actual rigidity, while axial stiffness may be influenced by the actual preload in operation

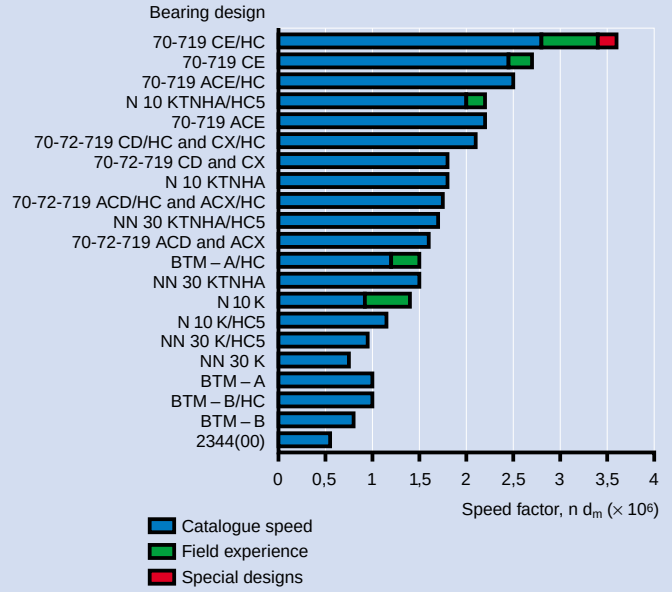
Speed

Rotational speed

The permissible operating temperature governs largely the speed at which rolling bearings can operate. Bearing types with low friction and thus low heat generation in the bearing itself are the most suitable for high-speed operation. Generally speaking, ball bearings are preferred for high speed. The following graphs gives guidelines on the attainable speeds from different bearing designs and executions with grease and oil lubrication (→ **Diagrams 1 and 2 page 24**).

Diagram 1

Speed guidelines for different bearing designs
Oil spot lubrication



However, the primary parameter that sets the actual limit for the operating speed in bearing systems is the maximum permissible temperature for safeguarding the lubricant life and the complete system's thermal stability.

The operating temperature of the system depends on a number of factors, and bearings are only one consideration. To obtain thermal stability it is always necessary that

$$W_g = W_d = W_n + W_c$$

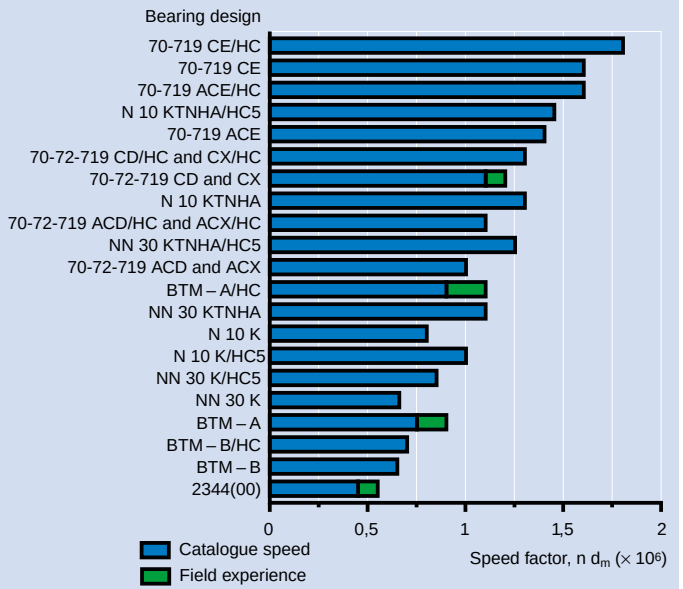
where W_g is the generated heat in the system and W_d is the heat dissipated by the system, which comes partly from natural dissipation, W_n, and partly is obtained by forced cooling, W_c. In many cases where no cooling acts on the system, $W_c = 0$.

W_g is the result of several factors such as the temperature of the environment, the heat generated by motors, electrical losses, friction in the bearings, friction of the lubricant, etc.

The heat generated by the bearings themselves has many causes, i.e. the bearings internal design, the material of the rings and rolling elements, the type of lubrication, the loads acting on the bearings (including preloading) etc.

When W_n is insufficient to stabilise the system temperature to the desired value, it is necessary to modify the design of the system to reduce W_g. If this is not possible, a cooling system must be introduced to re-establish the thermal equilibrium. This can be done by air-cooling the entire system, by using chilled air in the oil spot lubrication system, or by using the oil jet method, making the oil the vehicle for heat, transfer from the system to another area.

Diagram 2



Speed guidelines for different bearing designs
Grease lubrication

Speed ratings

The speed ratings quoted in the product tables are guideline values and are valid, provided that the bearings are lightly loaded, that they are lightly preloaded and that the transport of heat away from the bearing position is good.

The values under oil spot lubrication are maximum values and should be reduced for other methods of oil lubrication, other than those involving minimum oil quantities are applied, or where additional cooling is not provided. A 0,3 – 0,4 reduction factor should be considered with oil bath, while a 0,95 factor should be considered for oil mist lubrication. Conversely, oil jet might allow higher speeds than those given for oil spot, but it depends very much on oil type, oil supply rate, oil inlet temperature, oil drainage efficiency, etc. Please consult the SKF application engineering service for details.

The values under grease lubrication are maximum values that can be attained using a good quality grease of soft consistency.

Table 1

Relative speed capability of bearing systems for machine tool spindles equipped with different bearing arrangements

Bearing arrangement[1)] Rear side	Work side	Speed index Oil	Grease
NN 30 KTN	**2344 + NN 30 KTN**	84	70
NN 30 KTN	**BTM – B + NN 30 KTN**	100	84
NN 30 KTN	**BTM – A + NN 30 KTN**	100	89
N 10 KTNHA	**BTM – A + NN 30 KTN**	100	89
N 10 KTNHA	**70 CD QBCB**	102	60
N 10 KTNHA	**719 CE QBCB**	107	64
NN 30 KTN	**719 ACD/TBTB**	100	70
719 CD/DBA	**719 ACD/TBTB**	118	70
NN 30 KTN	**70 CD/TBTB**	100	73
70 CD/DBA	**70 CD/TBTB**	124	73
70 CD/DBA	**70 ACD/TBTB**	110	66
N 10 KTNHA	**70 CE QBCB**	88	56
70 CE/HCDT	**70 CE/HCDT**	312	204
70 CD/DBA	**70 ACD/DBB**	140	84
719 CE/HCDBA	**719 ACE/HCTBTA**	225	144
70 CE/DT	**70 CD/DT**	262	180
N 10 KTNHA	**70 CE/DBB**	176	120
719 CE/DBA	**719 ACE/TBTA**	190	121
70 CD/HCDBA	**70 ACE/HCDBA**	240	150
N 10 KTNHA/HC5	**70 ACE/HCDBA**	240	150
70 CE/DBA	**70 ACE/TBTA**	173	112
N 10 KTNHA	**719 ACE/DBA**	213	140
719 CE/DBA	**719 ACE/DBA**	220	140
70 CE/DBA	**70 ACE/DBA**	200	130
N 10 KTNHA	**70 ACE/DBA**	200	130

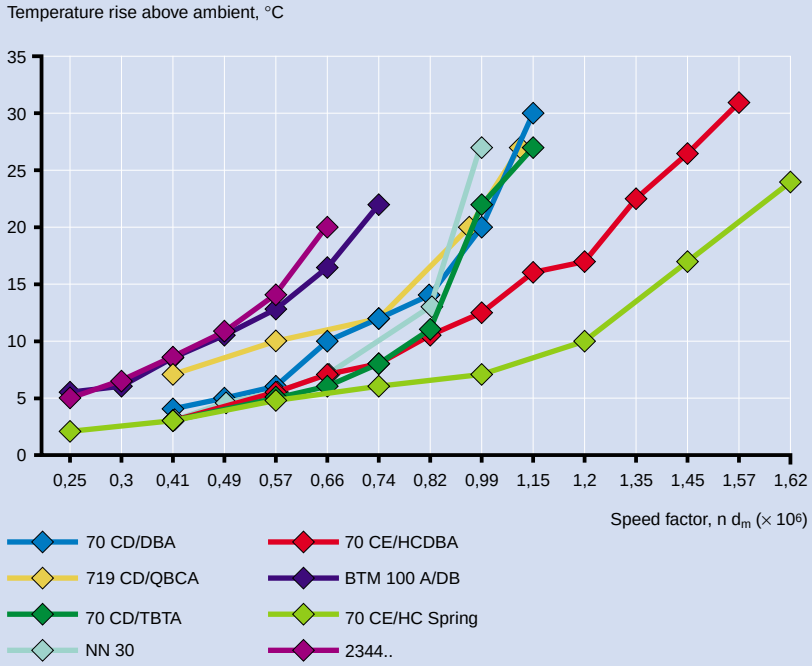
[1)] Reference size: work size bearings 80 mm bore diameter; rear side bearings 70 mm bore diameter

Speed capability of bearing systems

When designing a spindle, a bearing system will be used. The system may be composed of various bearing arrangements, normally a set of bearings at the work side, and another set at the drive side (rear end). The spindle speed must be evaluated on the most critical bearing set, normally the one at the tool side, which is bigger in bore diameter, forcing it towards high values of the speed factor n d_m. **Table 1 page 25** provides a comparison of possible choices in this respect.
A comparison of temperature rise versus speed for grease-lubricated spindles based on actual field results is shown in **Diagram 3**.

Temperature rise from existing applications
Grease lubrication

Diagram 3



Bearing data – general

Dimension

High-precision bearings, similarly to rolling bearings in general, are manufactured with very few exceptions, according to Dimension Plans for the boundary dimensions issued by the International Organisation for Standardisation (ISO).

More precisely, boundary dimensions follow the ISO Dimension Plan for radial bearings ISO15: 1998, where a progressive series of standardised outside diameters (Diameter Series 7, 8, 9, 0, 1, 2 etc., in order of increasing outside diameter), are set for every standard bore diameter. Within each Diameter Series different Width Series are also established (Width Series 8, 0, 1, 2, 3 etc., in order of increasing width).

The Width Series for radial bearings correspond to the Height Series for thrust bearings (Height Series 7, 9, 1 and 2 in order of increasing height). By combining a Diameter Series with a Width or Height Series, Dimension Series, designated by two figures, are arrived at. The first figure identifies the Width Series and the second the Diameter Series.

The bearings in this catalogue comply with the ISO Dimension Plans, with the exception of double direction angular contact thrust ball bearings and single direction angular contact thrust ball bearings belonging to metric series, whose boundary dimensions are not standardised, but recognised by the market and used by the manufacturers as such.

Tolerances

Tolerance classes for bearings are internationally standardised. Depending on the bearing type, SKF High-precision bearings are manufactured to the following tolerances (→ **Table** 1).

Actual tolerance values are equal to or better than those specified by the following international standards:

- ISO 199:1997
  Rolling bearings-Thrust bearings-Tolerances
- ISO 492:2002
  Rolling bearings-Radial bearings-Tolerances
- ANSI/ABMA Std. 20-1996
  Radial bearings of ball, cylindrical roller and spherical roller types, Metric design
- DIN 620-2:1999
  Rolling bearings-Tolerances for radial bearings
- DIN 620-3:1982
  Rolling bearings-Tolerances for thrust bearings

Tolerance symbols

Following are explanations for the symbols used in the tolerance tables:

Symbol	Definition
d	Nominal bore diameter
d_{mp}	Mean bore diameter, arithmetical mean of the largest and smallest single bore diameters in one plane
d_s	Single diameter of bore
Δ_{dmp}	Deviation of the mean bore diameter from the nominal
Δ_{d2mp}	Deviation of the mean bore diameter at large end of tapered bore (for cylindrical roller bearings), arithmetical mean of the largest and smallest single bore diameters at distance “a”
Δ_{d3mp}	Deviation of the mean bore diameter at small end of tapered bore (for cylindrical roller bearings), arithmetical mean of the largest and smallest single bore diameters at distance “a”
Δ_{ds}	Deviation of a single bore diameter from the nominal
V_{dp}	Bore diameter variation; difference between the largest and smallest single bore diameters in one plane
V_{dmp}	Mean bore diameter variation; difference between the largest and smallest mean bore diameters of one ring
D	Nominal outside diameter
D_{mp}	Mean outside diameter, arithmetical mean of the largest and smallest single outside diameters in one plane
D_s	Single diameter of outside cylindrical surface
Δ_{Dmp}	Deviation of the mean outside diameter from the nominal
Δ_{Ds}	Deviation of single outside diameter from the nominal
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
B_s, C_s	Single width of inner ring and outer ring, respectively
C_s	Single height (width) of housing washer (For bearing series 2344(00))
B_{1s}, C_{1s}	Single width of inner ring and outer ring, respectively, of bearings made for paired mounting
Δ_{Bs}, Δ_{Cs}	Deviation of single inner ring width or single outer ring width from the nominal
Δ_{Cs}	Deviation of single housing washer height (width) from the nominal
Δ_{B1s}, Δ_{C1s}	Deviation of single inner ring width or single outer ring width from the nominal of a bearing specially manufactured for paired mounting
V_{Bs}, V_{Cs}	Ring width variation; difference between the largest and smallest single width of inner ring and outer ring, respectively,
K_{ia}, K_{ea}	Radial runout of assembled bearing inner ring and assembled bearing outer ring, respectively
S_d	Side face runout with reference to bore (of inner ring)
S_D	Outside inclination variation; variation in inclination of outside cylindrical surface to outer side-face
S_{ia}, S_{ea}	Side face runout with reference to raceway of assembled bearing inner ring and assembled bearing outer ring respectively
S_i, S_e	Thickness variation, measured from middle of raceway to back (seating face) of shaft washer and housing washer, respectively
T_s	Single height of thrust bearing
Δ_{Ts}	Deviation of single height of thrust bearing from the nominal

Limits of chamfer dimensions

To prevent the improper dimensioning of associated components for rolling bearings and to facilitate the calculation of retaining ring location arrangements, minimum values for the chamfer dimensions in the radial direction (r_1, r_3) and the axial direction (r_2, r_4) are given in the product tables

These values conform to ISO 582:1995 for metric bearings with series designations. Maximum values associated with these minimum chamfer dimensions are found in the standards

High-precision bearing tolerance classes

Table 1

Radial angular contact ball bearings	**P4A and PA9A**
Radial cylindrical roller bearings	**SP and UP**
Single direction angular contact thrust ball bearings (ball screw support bearings)	**P4**
Double direction angular contact thrust ball bearings series 2344(00)	**SP and UP**
Double direction angular contact thrust ball bearings series BTM – A and BTM – B...	**P4C**

Bearing internal clearance or preload

All bearings, are designed in a way to get, in unfitted conditions, a certain amount of internal clearance that can be radial or axial, depending on the direction one of the rings can be moved, with respect to the other. This clearance value, fixed at the design stage, nevertheless, is reduced after mounting; because of the fitting conditions (interference) the bearing is subjected to and/or because of the thermal expansion or compression of the rings. Thus, operational clearance is the key parameter for a correct bearing operation.

To ensure maximum running accuracy and rigidity of the system, bearings used in machine tool spindles should have a minimum radial internal clearance or a preload after mounting. This is most valid for angular contact ball bearing sets, which are prepared at a certain step of the manufacturing process, in a way, that after fitting them on the shaft, the required preload is obtained.

Similarly, double direction angular contact thrust ball bearings are set up, in a way, to get a fixed static preload after mounting on the shaft.

Conversely, cylindrical roller bearings are supplied with radial clearance according to different possible classes. Depending on the speed operations, and on the thermal expansion or compression of the rings expected, the operational clearance can be set up at the required value, by driving the inner ring of the bearing on its tapered seat. For relatively low speed a certain amount of radial preload can be applied.

The clearance values for cylindrical roller bearings and the preload values for matched single row angular contact ball bearings, and single and double row angular contact thrust ball bearings will be found in the relevant tables in the product sections.

Materials for high-precision bearings

The performance and reliability required from high-precision bearings implies the use of adequate materials for rings, rolling elements and cages.

Steel for bearing rings and rolling elements

The standard SKF steel for high-precision bearings is, according to the usual classification, a “high carbon” content one (more than 0,5 %). Additional alloying elements such as Chromium, Manganese and Molybdenum are included in the steel composition to obtain the necessary properties in the finished components.

By means of the heat treatment and the resulting changes in the microstructure of the steels, the mechanical properties of the components can be addressed to specific requirements.

Rings and rolling elements of SKF high-precision bearings are made of typical martensitic through-hardened steel that provide sufficient resistance to sub-surface rolling contact fatigue, sufficient static capacity and structural strength combined with adequate dimensional stability.

SKF high-precision bearings can generally be used up to 120°C. When operating temperatures exceed this value a special heat treatment (stabilisation) has to be adopted, so that it is possible to get the required dimensional stability, avoiding changes in dimension that could result in premature bearing failure.

In order to significantly improve the resistance to corrosion, the use of nitrogen as an alloying element has been introduced in a newly developed bearing steel. Nitrogen leads to precipitation of chromium nitrides rather than chromium carbides, enabling a much higher content of chromium to be dissolved in the steel matrix, resulting in a better resistance to oxidation, and in a longer service life of the bearings.

It is advisable to contact SKF regarding the selection and application of nitrogen steel bearings.

Ceramic materials for rolling elements

Within the various ceramic materials, the hot isostatic pressed silicon nitride identified with the chemical formula Si_3N_4 is commonly used for rolling elements, in both balls and rollers.

Silicon nitride is hard and its main properties are high modulus of elasticity, low density and thermal expansion.

The modulus of elasticity of silicon nitride is some 50 % higher than for steel. This means that a ceramic rolling element under load does not distort to the same extent, and in turn, the contact between rolling elements and rings is smaller. So there is less friction.

The density of silicon nitride rolling elements is 60 % lower than that of steel of the same size. This leads to considerably reduced centrifugal loads and consequently reduced stresses. Furthermore, for angular contact ball bearings, the lower density has a beneficial effect on contact angle variation at high speeds, leading to reduced ball sliding and consequently to reduced friction.

The thermal expansion of silicon nitride is less than 30 % that of steel and this, in cases, where the difference in temperature between inner ring (usually warmer than outer ring) and outer ring is not negligible, reduces the clearance reduction and/or preload increase, hence reducing the risk of drastic failure.

A comparison of the physical properties of silicon nitride Si_3N_4 and bearing steel is shown in **Table 2**.

Bearings assembled with steel rings and ceramic rolling elements are commonly called hybrids. Compared to standard all-steel bearings they can run at higher speeds with a certain temperature increase, or at the same speeds with a lower temperature increase. They have also a greater rigidity, both under dynamic and static conditions. They are less sensitive to temperature differences between the rings, and the preload increase is smaller than for normal bearings. The low coefficient of friction of silicon nitride enhances their wear resistance and enables them to run cooler under poor lubrication conditions. Moreover, silicon nitride is hard, tough and strong and these properties combined with a better surface finish give better resistance to damage from particles and impurities that may enter the bearing.

Properties of silicon nitride and bearing steel

Table 2

Property	Unit	Silicon nitride Si_3N_4	Bearing steel, hardened
Density, ρ	g/cm^3	3,2	7,9
Coefficient of thermal expansion	×10^{-6}/K		
20 – 1 000 °C		3,2	
20 – 300 °C			11,5
Modulus of elasticity, E	Gpa	310	210
Poisson's ratio	-	0,26	0,3
Hardness HV10	kg/mm^2	1 600	700
Tensile strength	MPa		
20 °C		800	2 400
1 000 °C		800	0
Fracture toughness, KIC	Mpa.m$^{1/2}$	7	25
Thermal conductivity	W/mK	30 – 40	40 – 50
Specific electrical resistivity	Ωm	10^{12}	$0,4 \times 10^{-6}$

Material for cages

The main purpose of the cage is to keep the rolling elements at an appropriate distance from each other and to prevent immediate contact between two neighbouring rolling elements, in order to keep friction and consequently heat generation at a minimum. Materials and the shape of cages for high-precision bearings have been developed parallel to the development of the bearings and to the requirements they have to satisfy.

High-precision bearing cages are mechanically stressed by frictional strain and inertia forces. They might also be subjected to chemical action of certain lubricants. Thus, the design and choice of material are of paramount importance for the performance of the cage, as well as for the operational reliability of the bearing as a whole.

For each of the bearings shown in the product tables one particular cage design is established as the standard cage for that bearing. The standard cage is always well proven in service and is the design considered most suitable for the majority of applications. In the introductory text to each table section information is provided regarding the standard cages bearings are fitted with and also the possible alternatives. Machined outer ring land-riding fabric-reinforced phenolic resin cages are used for most of the angular contact ball bearings. This material is lightweight resulting in minimal centrifugal forces and having the capability of retaining part of the lubricant, ensuring optimum lubrication. Fabric-reinforced phenolic resin cages should not be used at temperatures exceeding 120°C.

Moulded glass fibre reinforced polyamide cages are normally fitted in cylindrical roller bearings and single and double row angular contact thrust ball bearings. This material is characterised by a favourable combination of strength and elasticity. The good sliding properties of the polyamide on lubricated steel surfaces and the smoothness of the cage surface in contact with the rolling elements results in little friction from the cage, with minimum heat generation and wear in the bearing. They can be used for operating temperatures not exceeding 120°C.

When higher speeds or better performances need to be reached, as in the case of angular contact ball bearings or single row cylindrical roller bearings, glass fibre reinforced PEEK (poly-ether-ether-ketone) cages are chosen. The injection moulding process used to produce these cages allows functionally suitable designs to be realised. Moreover, PEEK offers a better stability and rigidity than polyamide.

For lower speeds and higher loads, as in the case of large size double row cylindrical roller bearings or double direction angular contact thrust ball bearings, machined brass cages generally centred on the rolling elements are selected.

For special applications, where high temperatures or poor lubrication are involved, silver-plated steel or brass cages can represent the optimal solution.

Application of bearings

Bearing arrangements

General
The classic application field for high-precision bearings is machine tools spindles, which may have different requirements depending on the working operations they are designed for. Generally, lathe spindles are used to cut metals at rather low speeds and in combination with relatively large cutting loads. Such types of spindles usually have the driving torque transmitted through a pulley or toothed gears. This means that loads at the rear side of the shaft are also rather heavy. For such applications, the requirements in terms of speed are not so tough; the more important parameters are rigidity and load carrying capacity. It is quite common to have, at the front side of the spindle, a double row cylindrical roller bearing in combination with a double row angular contact thrust ball bearing, while having a double row cylindrical roller bearing at the rear end of the shaft. This type of arrangement ensures a long calculated life and an excellent rigidity, so that a good quality of the workpiece is obtained. Also, from a kinematic point of view, the bearings run in a stable way, as there are two types of bearing (radial and axial) that carry independently, the loads applied on the shaft (in fact, to avoid that angular contact thrust ball bearings carry radial loads, the outer ring outside diameter has a special tolerance so that it is never in contact with the housing). When designing these types of spindles (this applies in general when rather heavy loads are involved) a good rule of thumb concerning the position of the bearings along the shaft, is to have the distance between the centre of the front and rear support in the range of 3 – 3,5 times the bearing inner diameter.

When higher speeds are requested (i.e. high-speed machining centre or internal grinding) different bearing solutions need to be found. Obviously, in such cases something has to be paid in terms of rigidity, as well as carrying capacity. High-speed applications usually have direct-driven spindles driven by direct coupling and/or electrical motors (i.e. the so called motorised spindles). There are no loads due to transmission of power and consequently single row angular contact ball bearings paired in sets or single (for extremely high speed) and single row cylindrical roller bearings are frequently adopted, if enhanced performances are required, equipped with silicon nitride rolling elements. The front side bearing set is axially located, whilst mounting a cylindrical roller bearing at the rear side permits axial displacement, due to spindle elongation.

When very high speeds are involved ($n\,d_m$ factor over 2 million) it is quite common to see angular contact ball bearings on both sides, preloaded by springs. This is done to control the heat generation. If sets of angular contact ball bearings arranged in a constant position are chosen, preload would increase with the speed, and at high speed produce an amount of heat which is not sustainable.

Conversely, spring preload remains constant with the speed, thus ensuring a more correct kinematic behaviour and a limited amount of heat generation.

An even better solution is represented by the possibility to preload the bearings (angular contact) by a hydraulic system. In such a case, the amount of preload can be adjusted according to the speed of the spindle, thus reaching the best possible combination among rigidity, heat generation and theoretical life of the bearings.

Radial location of bearings

General
In order to work properly, bearings must have their rings or washers fully supported around their complete circumference and across the whole width of the raceway. The support must be firm and even, and can be provided by a cylindrical or tapered seating or, for thrust bearing washers, by a flat (plane) support surface. This means that the seatings must be made with adequate accuracy and that their surface should be uninterrupted by grooves, holes or other features. This is particularly important with high-precision bearings since they have relatively thin-walled rings, which adapt themselves to the form of the shaft or housing bore. In addition, the bearing rings must be reliably secured to prevent them from turning on or in their seatings under load.

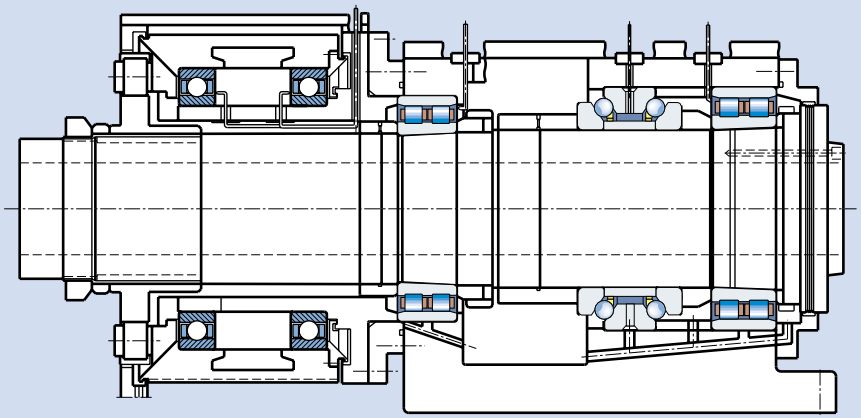
A satisfactory radial location and an adequate support can generally be obtained when the rings are mounted with an appropriate degree of interference. Inadequately or incorrectly secured bearing rings generally cause damage to the bearings and associated components. However, when easy mounting and dismounting are desirable, or when axial displacement is required with a non-locating bearing, an interference fit cannot be used. In certain cases where a loose fit is employed it is necessary to take special precautions to limit the inevitable wear, for example, surface hardening of the seating and abutments.

Application examples

Spindle arrangements for heavy machining, CNC lathes and conventional milling machines (➔ **figs** 1, 2 **and** 3).

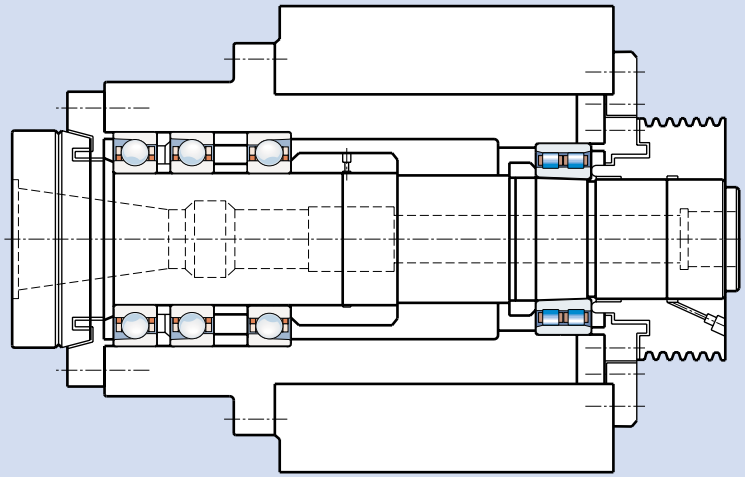
Bearing arrangement work side: NN 30 K + 2344(00); drive side NN 30 K

Fig 1



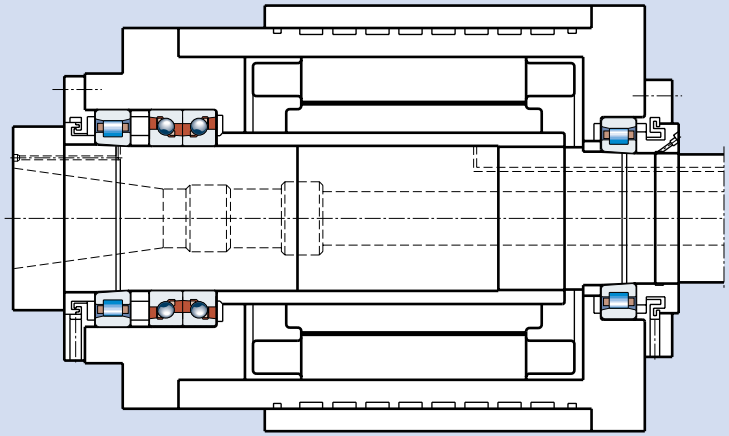
Bearing arrangement work side: 70 ACD/P4ATBT; drive side NN 30 K

Fig 2



Bearing arrangement work side: N 10 KTN + BTM-A/HC; rear side N 10 KTN

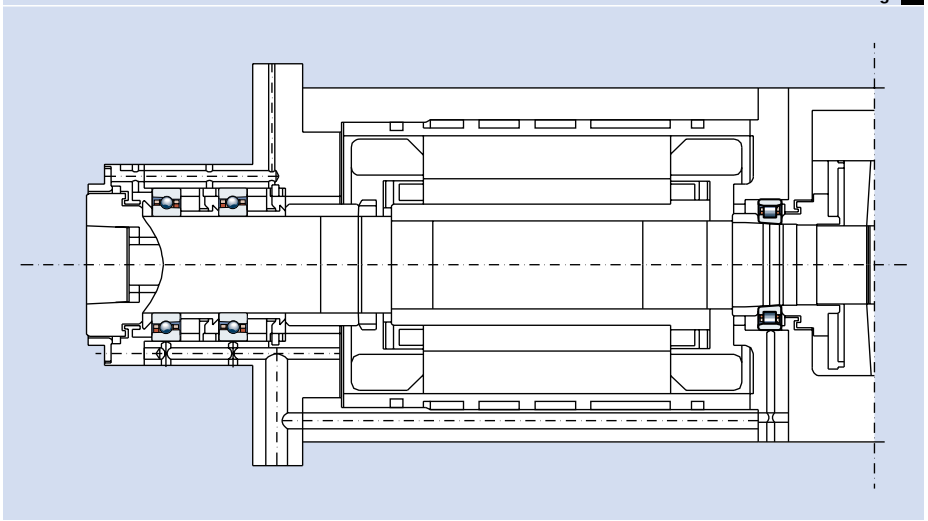
Fig 3



Spindle arrangements for great rigidity and high speed, machining centres, high speed turning centres and high speed milling (→ **figs 4, 5, 6 and 7**).

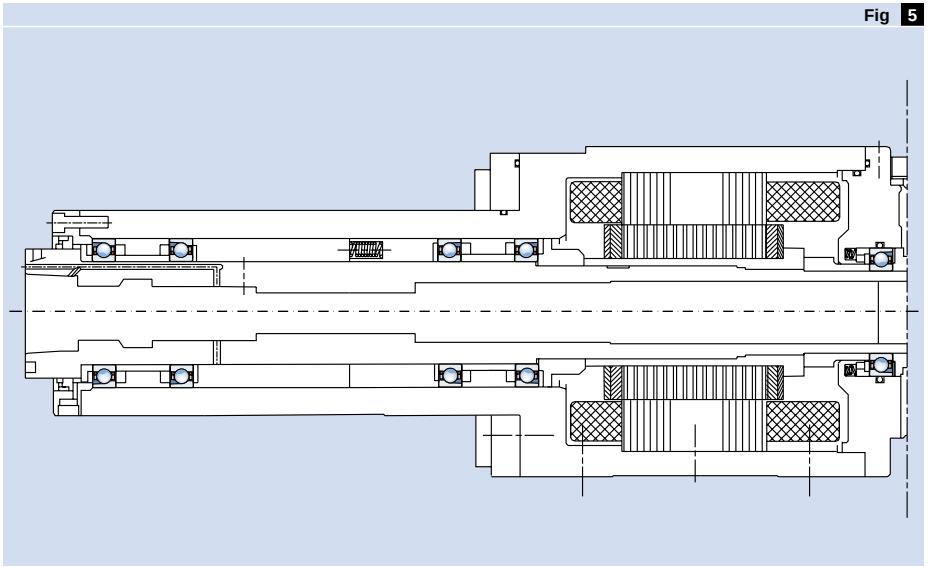
Bearing arrangement work side: 70 CE/HCP4ADB; rear side N 10 KTN

Fig 4



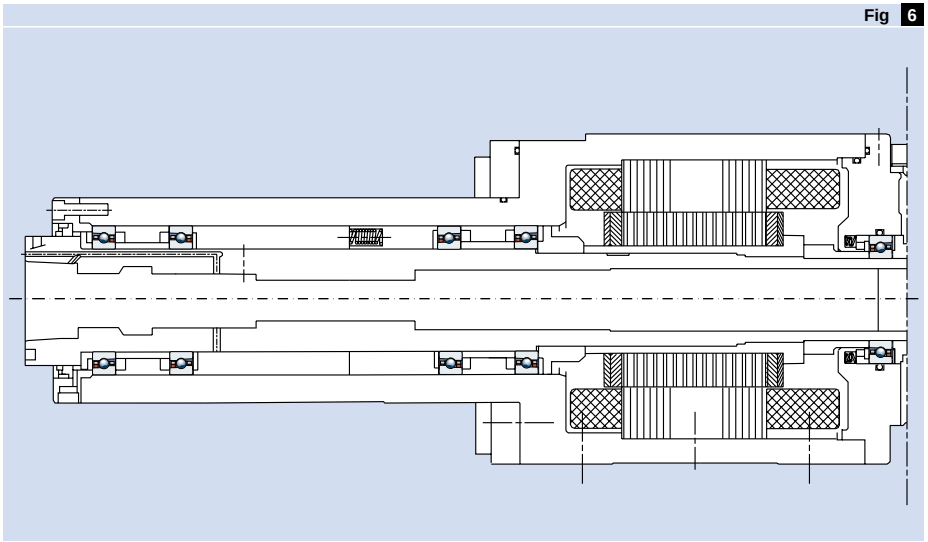
Bearing arrangement work side: 70 CD/P4AQBC; rear side 70 CD/P4A

Fig 5



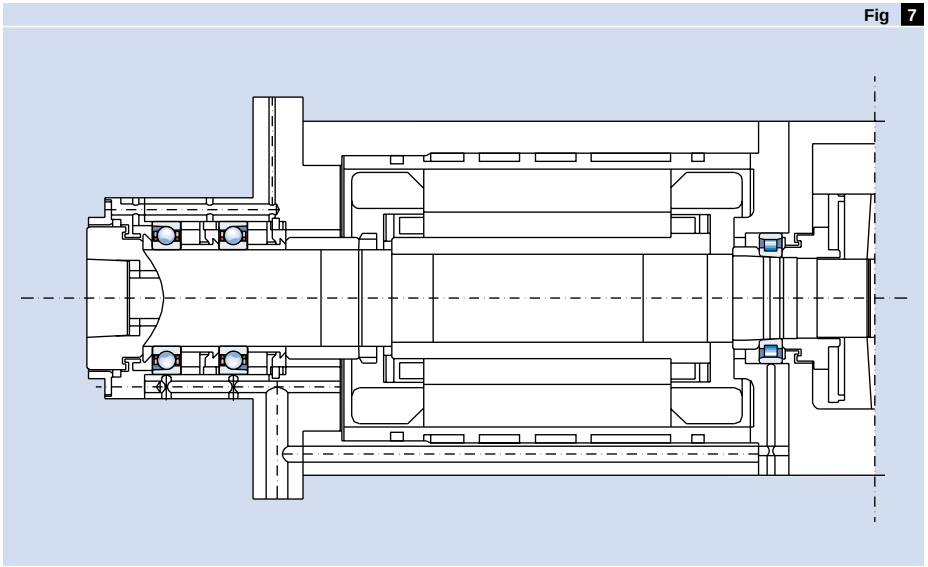
Bearing arrangement work side: 70 CE/HCP4AQBC; rear side 70 CE/HCP4A

Fig 6



Bearing arrangement work side: 70 CD/P4ADB; rear side N 10 KTN

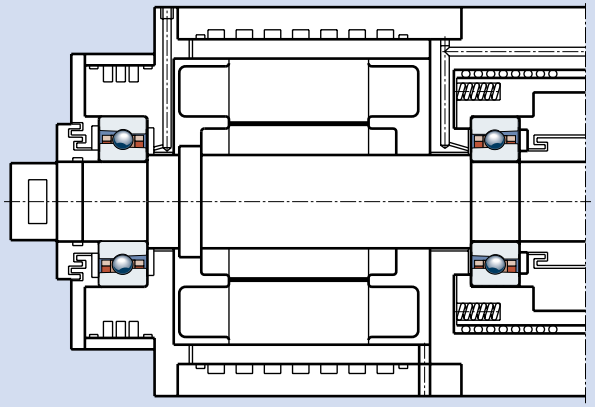
Fig 7



Spindle arrangements for maximum speed, internal grinding machines (➔ **figs** 8 **and** 9).

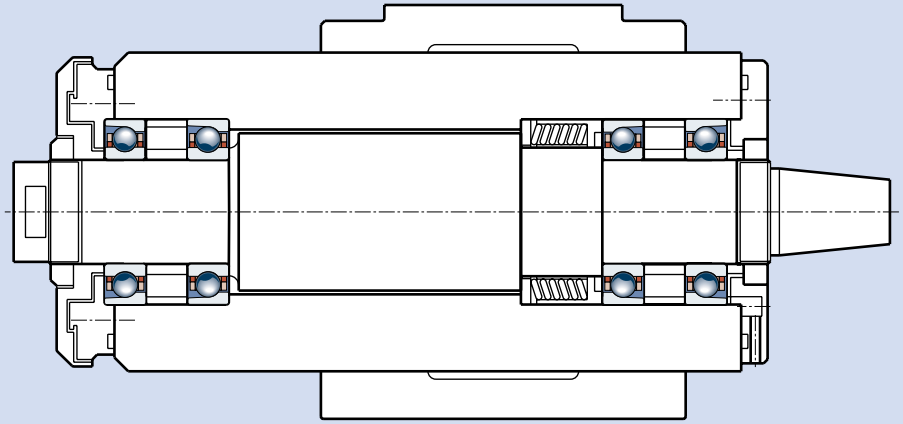
Bearing arrangement work side: 70 CE/HCP4A; rear side 70 CE/HCP4A

Fig 8



Bearing arrangement work side: 70 CD/HCP4ADT; rear side 70 CD/HCP4ADT

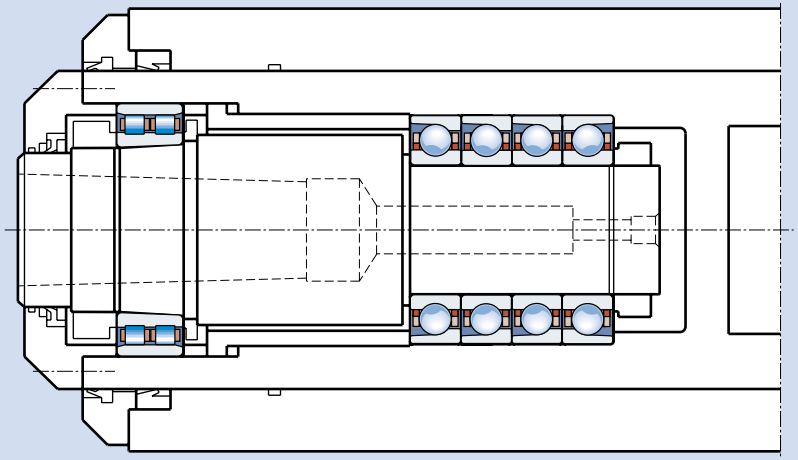
Fig 9



Bearing arrangement for live centres (➔ **fig** 10).

Bearing arrangement work side: NN 30 K ; rear side 72 ACD/P4AQBT

Fig 10



Recommended fits

Appropriate shaft and housing tolerances for high-precision bearings are shown in **Tables 1 and 2**. However, in the specific case of spindle applications with normal load and speed conditions, the interference/clearance fit between bearings, shaft and housing shown in **Tables 3 and 4 page 44** are recommended.

The table of housing tolerance recommendations also gives information as to whether the outer ring can be axially displaced in the housing bore.

For extreme conditions, such as very high speed or exceptional loading, please consult SKF application engineering services.

Fits for steel shafts (solid and hollow)

Table 1

Bearing type	Shaft diameter over	Shaft diameter incl.	Tolerance Bearings of tolerance class P4A, SP, P4C	Tolerance Bearings of tolerance class PA9A, UP
	mm	mm		
Angular contact ball bearings				
with rotating outer ring load	–	240	**h4**	**h3**
with rotating inner ring load	–	240	**js4**	**js3**
Cylindrical roller bearings				
with cylindrical bore	–	40	**js4**	**–**
	40	140	**k4**	**–**
	140	200	**m5**	**–**
	200	240	**n5**	**–**
Angular contact thrust ball bearings				
Single direction	–	130	**h4**	**–**
Double direction	–	200	**h4**	**h3**

Table 2

Bearing type	Operating conditions	Tolerance Bearings of tolerance class P4A, SP, P4C	Tolerance Bearings of tolerance class PA9A, UP
Angular contact ball bearings	Non-locating bearings, displacement of outer ring desired	**H5**[1]	**H4**[1]
	Locating bearings, displacement of outer ring not required	**JS5**	**JS4**
	Rotating outer ring load	**M5**	**M4**
Cylindrical roller bearings	Normal and light loads	**K5**	**K4**
	Heavy loads, rotating outer ring loads	**M5**	**M4**
Angular contact thrust ball bearings			
Single direction	–	**H5**	**–**
Double direction[2]	–	**K5**	**K4**

[1] The upper half of the recommended tolerance range should be used, especially when important drive loads (belts, gears) are acting at the spindle drive side

[2] The 2344(00) and BTM series bearings are mounted with radial clearance in the same housing bore seating as the appropriate cylindrical roller bearing. Fits tighter than those recommended should never be used even if they are required for cylindrical roller bearings

Fits for cast iron and steel housings

Preferred fits on the shafts

Table 3

Bearing type	Bearing bore over	Bearing bore incl.	Interference
	mm	mm	µm
Angular contact ball bearings	–	50	0 – 2
	50	80	1 – 3
	80	120	1 – 4
	120	150	2 – 5
	150	200	2 – 6

Table 4

Bearing type	Outside diameter over	incl.	Recommended clearance for bearings axially located	free (non-located)
	mm	mm	µm	
Angular contact ball bearings	–	50	0 – 2	5 – 8
	50	120	0 – 3	6 – 10
	120	150	0 – 4	8 – 12
	150	250	0 – 5	10 – 15

Bearing type	Outside diameter over	incl.	Recommended interference
	mm	mm	µm
Cylindrical roller bearings	–	460	0 – 2

Preferred fits in the housings

Tolerances for shaft and housing
Appropriate shaft and housing tolerances for high-precision bearings will be found in the **Table** 5 **page 46** : ISO shaft limits and **Table** 6 **page 47** : ISO housing limits.

The positioning of tolerance zones given in the tables, in relation to the nominal bore and outside diameter of the bearings is shown in **Diagram** 1.

Specific recommendations for single direction angular contact thrust ball bearing seatings are given in the relevant bearing section.

ISO shaft and housing limits, position of the tolerance zones

Diagram 1

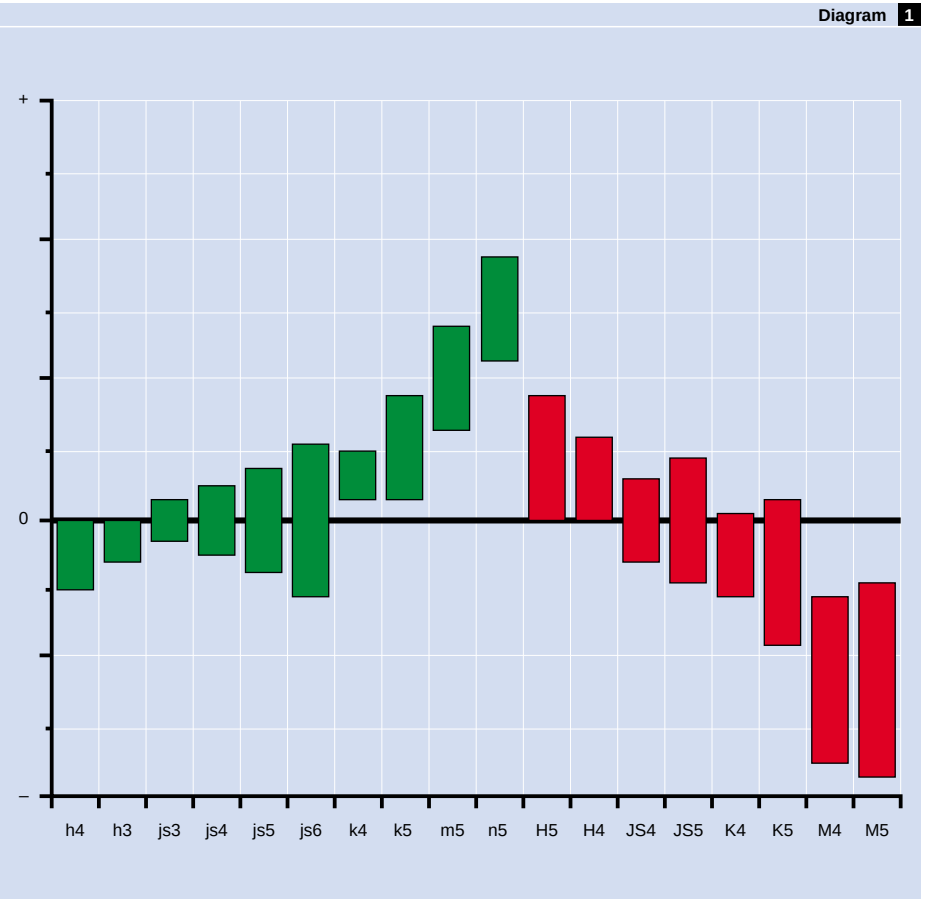


Table 5

Shaft diameter Nominal		Tolerances h4 Deviations		h3		js3		js4		js5	
over	incl.	high	low	high	low	high	low	high	low	high	low
mm		µm									
6	**10**	0	–4	0	–2,5	+1,25	–1,25	+2	–2	+3	–3
10	**18**	0	–5	0	–3	+1,5	–1,5	+2,5	–2,5	+4	–4
18	**30**	0	–6	0	–4	+2	–2	+3	–3	+4,5	–4,5
30	**50**	0	–7	0	–4	+2	–2	+3,5	–3,5	+5,5	–5,5
50	**80**	0	–8	0	–5	+2,5	–2,5	+4	–4	+6,5	–6,5
80	**120**	0	–10	0	–6	+3	–3	+5	–5	+7,5	–7,5
120	**180**	0	–12	0	–8	+4	–4	+6	–6	+9	–9
180	**250**	0	–14	0	–10	+5	–5	+7	–7	+10	–10
250	**315**	0	–16	0	–12	+6	–6	+8	–8	+11,5	–11,5

Shaft diameter Nominal		Tolerances js6 Deviations		k4		k5		m5		n5	
over	incl.	high	low	high	low	high	low	high	low	high	low
mm		µm									
6	**10**	+4,5	–4,5	+5	+1	+7	+1	+12	+6	+16	+10
10	**18**	+5,5	–5,5	+6	+1	+9	+1	+15	+7	+20	+12
18	**30**	+6,5	–6,5	+8	+2	+11	+2	+17	+8	+24	+15
30	**50**	+8	–8	+9	+2	+13	+2	+20	+9	+28	+17
50	**80**	+9,5	–9,5	+10	+2	+15	+2	+24	+11	+33	+20
80	**120**	+11	–11	+13	+3	+18	+3	+28	+13	+38	+23
120	**180**	+12,5	–12,5	+15	+3	+21	+3	+33	+15	+45	+27
180	**250**	+14,5	–14,5	+18	+4	+24	+4	+37	+17	+51	+31
250	**315**	+16	–16	+20	+4	+27	+4	+43	+20	+57	+34

ISO shaft limits

Table 6

Housing bore diameter Nominal		Tolerances H5 Deviations		H4		JS4		JS5	
over	incl.	high	low	high	low	high	low	high	low
mm		µm							
18	**30**	+9	0	+6	0	+3	–3	+4,5	–4,5
30	**50**	+11	0	+7	0	+3,5	–3,5	+5,5	–5,5
50	**80**	+13	0	+8	0	+4	–4	+6,5	–6,5
80	**120**	+15	0	+10	0	+5	–5	+7,5	–7,5
120	**180**	+18	0	+12	0	+6	–6	+9	–9
180	**250**	+20	0	+14	0	+7	–7	+10	–10
250	**315**	+23	0	+16	0	+8	–8	+11,5	–11,5
315	**400**	+25	0	+18	0	+9	–9	+12,5	–12,5
400	**500**	+27	0	+20	0	+10	–10	+13,5	–13,5

Housing bore diameter Nominal		Tolerances K4 Deviations		K5		M4		M5	
over	incl.	high	low	high	low	high	low	high	low
mm		µm							
18	**30**	0	–6	+1	–8	–6	–12	–5	–14
30	**50**	+1	–6	+2	–9	–6	–13	–5	–16
50	**80**	+1	–7	+3	–10	–8	–16	–6	–19
80	**120**	+1	–9	+2	–13	–9	–19	–8	–23
120	**180**	+1	–11	+3	–15	–11	–23	–9	–27
180	**250**	0	–14	+2	–18	–13	–27	–11	–31
250	**315**	0	–16	+3	–20	–16	–32	–13	–36
315	**400**	+1	–17	+3	–22	–16	–34	–14	–39
400	**500**	0	–20	+2	–25	–18	–38	–16	–43

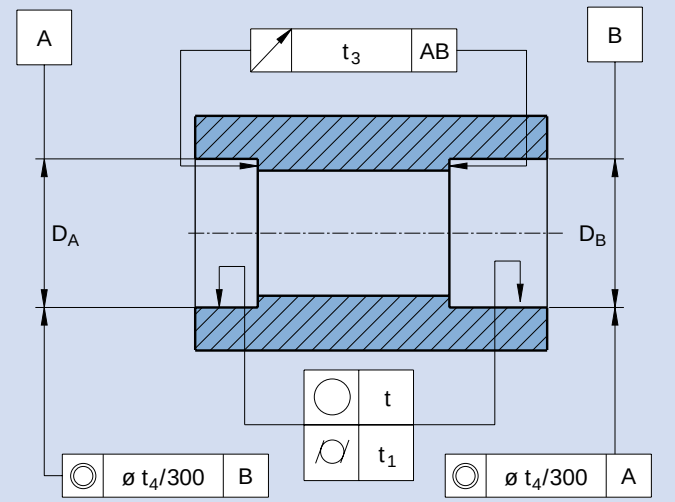
ISO housing limits

Accuracy of associated components
Maximum running accuracy, high speeds and low operating temperatures can only be achieved, even with precision bearings, if the mating parts and other associated components are made with equal precision. Deviations from geometric form must therefore be kept as small as possible where the mating parts are concerned.

The rings of the bearings shown in this catalogue are relatively thin-walled, so they will adapt themselves to the form of the shaft or housing bore. Any errors of form or other inaccuracies of the shaft and housing bore seating will thus be transmitted to the raceways of the bearing rings. Angular misalignment of one bearing ring in relation to the other can cause high operating temperatures; the axial support surfaces for the faces of the bearing rings must therefore, be precisely machined. This is particularly important where high-speed operation is intended. Consequently, an important prerequisite to achieving highly accurate bearing arrangements is that the recommendations, concerning accuracy of form and position, as well as surface finish, are adhered to when machining the mating parts (➔ **figs 11 and 12**).

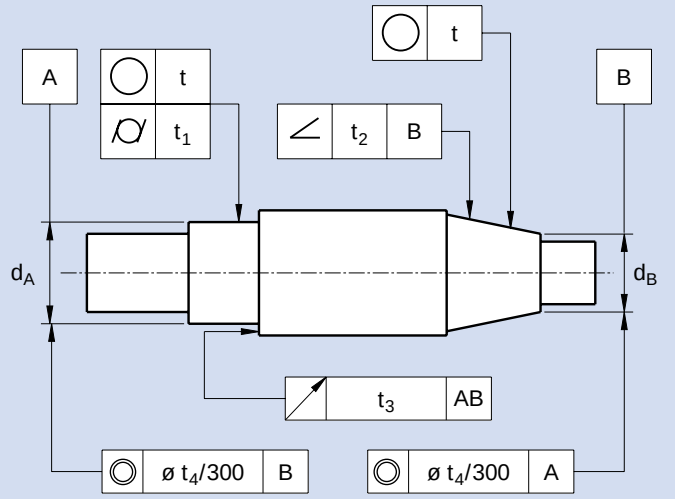
Limits for tolerance grades and roughness classes can be found in **Tables 7 and 8 page 50**.

Fig 11 ***Accuracy of form for housings*** [1)]



[1)] The housing form tolerances, symbols and reference surfaces are in accordance with ISO 1101

Fig 12 ***Accuracy of form for shafts*** [1)]



[1)] The shaft tolerances, symbols and reference surfaces are in accordance with ISO 1101

Table 7

Tolerance grades appropriate to spindle bearings

Tolerance parameter		Shaft Tolerance class P4A, SP, P4C	Shaft Tolerance class PA9A, UP	Housings Tolerance class P4A, SP, P4C	Housings Tolerance class PA9A, UP
Circularity	t	IT2/2	IT1/2	IT2/2	IT1/2
Cylindricity	t_1	IT2/2	IT1/2	IT2/2	IT1/2
Angularity	t_2	IT3/2	IT2/2		
Runout	t_3	IT1	IT0	IT1	IT0
Coaxiality	t_4	IT4	IT3	IT4	IT3
Roughness[1] d, D over	incl.				
–	80	N4	N3	N5	N4
80	250	N5	N4	N6	N5
250		N6	N5	N7	N6

[1] For surface roughness Ra see **Table 8**; roughness classes N to ISO 1302

Limits for tolerance grades, surface roughness class values[1]

Table 8

Diameter Nominal over	incl.	Tolerance grades IT0	IT1	IT2	IT3	IT4	IT5	Surface roughness class	Roughness Ra
mm		µm						–	µm
6	**10**	0,6	1	1,5	2,5	4	6	**N3**	0,1
10	**18**	0,8	1,2	2	3	5	8	**N4**	0,2
18	**30**	1	1,5	2,5	4	6	9	**N5**	0,4
30	**50**	1	1,5	2,5	4	7	11	**N6**	0,8
50	**80**	1,2	2	3	5	8	13	**N7**	1,6
80	**120**	1,5	2,5	4	6	10	15		
120	**180**	2	3,5	5	8	12	18		
180	**250**	3	4,5	7	10	14	20		
250	**315**	4	6	8	12	16	23		
315	**400**	5	7	9	13	18	25		
400	**500**	6	8	10	15	20	27		

[1] Surface roughness in Ra to DIN 7184; roughness classes N to ISO 1302

Axial location of bearings

General

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 locating bearings should be axially secured at both sides. For non-locating bearings, on the other hand, where they are of a non-separable design, it is sufficient if the ring having the tighter fit – usually the inner ring – is axially secured; the other ring must be free to move axially with respect to its seating.

In machine tool applications, usually, the work side bearings ensure the shaft location to support the axial load transmission from the shaft to the housing. Generally, then, work side bearings are axially located, while rear side supports are axially free.

Methods of location

Bearing rings having an interference fit are generally mounted so that the ring abuts a shoulder on the shaft or in the housing at one side. At the opposite side, inner rings are normally secured using a lock nut (of series KMT or KMTA). Outer rings are usually retained by a housing end cover.

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.

Other methods of axial location which are suitable, above all, for high precision bearing arrangements involve the use of press fits, e.g. in the form of stepped sleeve arrangements.

Bearings with tapered bore mounted directly on tapered journals are generally retained by a lock nut on the shaft.

Stepped sleeves

Stepped sleeves may be used to axially lock bearings or other precision components on a shaft.

Compared to threaded locking nuts, stepped sleeves ensure a superior accuracy, provided they are manufactured to a high degree of accuracy. Conversely, stepped sleeves are expensive to manufacture, have to be designed properly, and require proper mounting procedure.

Stepped sleeves are generally used in very high-speed spindles, where the accuracy granted by conventional locking devices may not be sufficient.

Stepped sleeves are not standardised and can be designed in many different ways to suit the spindle design.

Bearing preload

Depending on the application, it is necessary to have a positive, or a negative operational clearance in a bearing arrangement. In the majority of high-precision bearing applications a negative operational clearance i.e. a preload is desirable in order to enhance the stiffness of the bearing arrangement, or to increase the running accuracy. The application of a preload is also recommended where bearings are to operate without load, or under very light load and at high speeds. In such cases the preload serves to guarantee a minimum load on the bearing and thus prevents bearing damage resulting from sliding movements.

Types of preload
Depending on the type of bearing the preload may be either radial or axial. Cylindrical roller bearings can only be radially preloaded and thrust ball bearings can only be axially preloaded. Single row angular contact ball bearings that are normally subjected to axial preload, are generally mounted together, with a second bearing of the same type in a back-to-back or face-to-face arrangement.

The distance between the pressure centres of two angular contact ball bearings is longer when the bearings are arranged back-to-back and shorter when they are arranged face-to-face. This means that the bearings arranged back-to-back can accommodate large tilting moments even if the distance between bearing centres is relatively short.

Reasons for bearing preload
The main reasons of bearing preload are:
- enhance stiffness
- reduce running noise
- enhance the accuracy of shaft guidance
- compensate for wear and settling (bedding down) processes in operation
- give a long service life.

It is easy to understand that preloading bearings have a very positive effect for machine tool spindles.

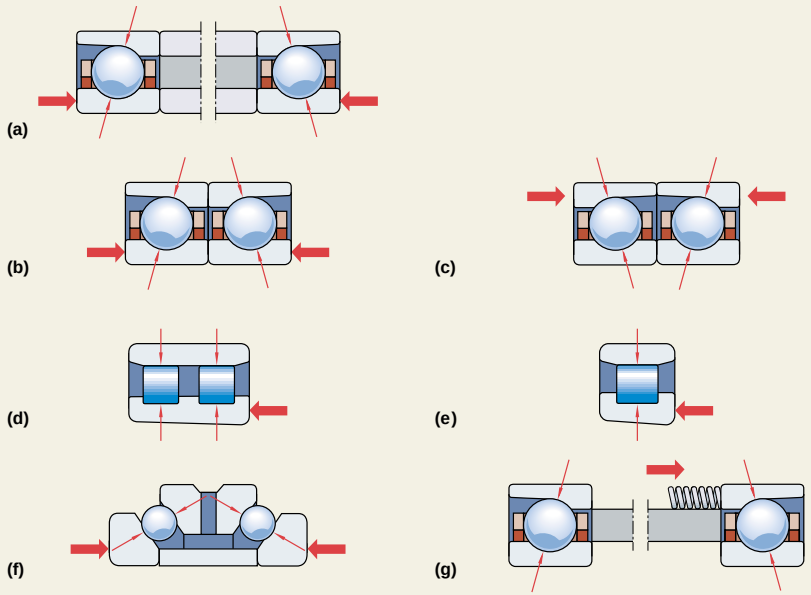
Adjustment procedures (→ fig 13)
Single row angular contact ball bearings are generally adjusted against each other by axial displacement of the inner or outer rings until a certain preload (or a certain clearance) is obtained in the bearing arrangement, (a). Single row angular contact ball bearings that are mounted in sets, (b) and (c), are matched in production so that when they are mounted immediately adjacent to each other, predetermined values of preload are obtained. Cylindrical roller bearings with tapered bore are preloaded by driving the inner ring up on to its tapered seating, (d) and (e); see also mounting instructions from **page 79**.

With regard to double direction angular contact thrust ball bearings, the spacer sleeve arranged between the shaft washers is dimensioned so that a suitable preload will be obtained once the bearing has been mounted, (f). For high speed bearing arrangements incorporating angular contact ball bearings it is customary to axially preload the bearings by means of springs, (g). This way it is possible to maintain a constant preload in the bearing arrangement throughout the whole range of operating conditions.

Specific information concerning preload values is given in the relevant bearing section.

Bearing preload - adjustment procedures

Fig 13



Preloading by springs

The simplest method of applying preload is by springs or spring “package” (➔ **fig** 14). The spring acts on the outer ring of the bearings that are 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.

Spring loading is a common method of applying preload to the angular contact ball bearings of high speed grinding spindles. The method is not suitable, however, for bearing applications where high stiffness is required, where the direction of load changes, or where undefined shock loads can occur.

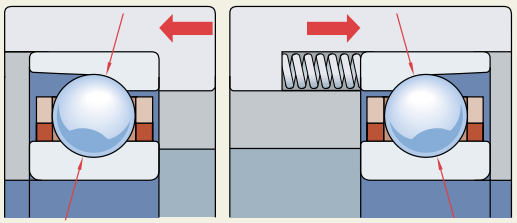
Specific recommendations for preloading by springs are given in the angular contact ball bearing section.

External axial loads on preloaded bearing sets

Angular contact ball bearing sets, and double direction angular contact thrust ball bearings are matched during manufacture to have a predetermined preload once in operation. External axial load may release bearings not supporting the external load from preload (➔ **Diagram** 2).

When two bearing rows are preloaded against each other, the same force is exchanged between the two rows. Force will induce a certain displacement in each row, δ. When an external axial load is applied to bearing 1, K_a, deflection of bearing 1 will follow the relevant curve on the above-mentioned diagram. At the same time bearing 2 will be subjected to progressively less axial preload which will move from the original static preload value F_0 down to the residual force F_2. When K_a reaches the value F_b, bearing 2 has no preload left. The force F_b is then the axial external load which releases bearing 2 from preload, the so-called ‘Breakaway’ load.

Fig 14



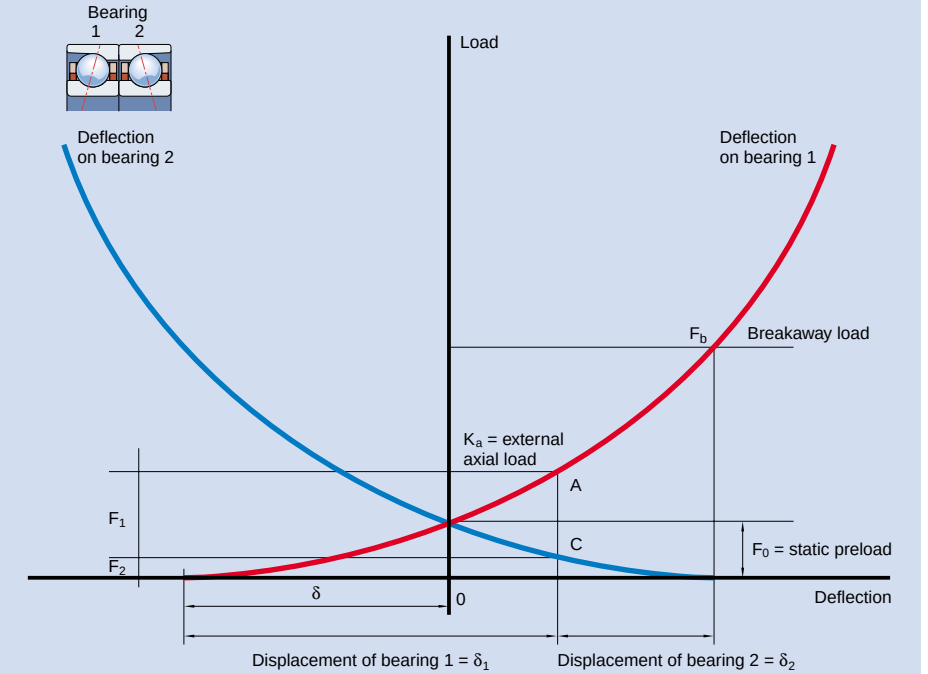
For an application where extremely high speed is required and moderate stiffness is acceptable, preloading should be done through calibrated springs acting against a bearing ring

Breakaway load varies depending on preload. For angular contact ball bearing sets of two bearings matched back-to-back or face-to-face, for series 2344(00) and BTM the relation between breakaway load and preload is shown in **Table** 9 **page 56**.

It sometimes happens that the external axial load is greater than the breakaway load F_b. This usually occurs when the spindle is cutting with heavy axial forces and rotating at relatively low speed. In such conditions, and if the spindle, while working is not subjected to strong acceleration, no consequences to operation will result. However, to avoid working beyond the breakaway load it is possible to increase the preload of the bearing sets, or if this is not possible, bearing sets with mixed contact angles may be used. For advice on this, please consult the SKF application engineering service.

External axial loads on preloaded bearing sets

Diagram 2



Preload for customised needs

As it is mentioned above, high-precision angular contact ball bearing preload is predetermined during manufacture. In certain special cases it might happen that a different preload is required to achieve the best performances. In these cases it is recommended to use spacers between the bearings with different length.

All details covering spacers and spacer reworking are shown in the angular contact ball bearing section.

Guidelines for breakaway load calculation

Table 9

Bearing type & arrangement	Breakaway load vs static preload
Series 70, 72 and 719 CD or ACD, CX or ACX in DB or DF sets	$F_b = 2{,}8\ F_0$
Series 70, 72 and 719 CD or ACD, CX or ACX in TBT or TFT sets (where two bearings are supporting the load)	$F_b = 4{,}2\ F_0$
2344(00), BTM – A and BTM – B series	$F_b = 2{,}85\ F_0$
Ball screw support bearings	as for 70, 72 and 719 series

Note: for bearings series or designs not listed above please contact SKF application engineering service

Seals

Bearing positions must be efficiently sealed both outwards and inwards, so that contaminants and damp cannot penetrate, and the lubricant cannot escape, if reliable performance is to be obtained. This is particularly true of bearing arrangements incorporating high-precision bearings where the demands in respect of running accuracy, bearing life and reliability are generally very high.

Seals can be of two basic types: non-contacting (non-rubbing) and contacting (rubbing).

Contacting seals

The following are some examples of contacting seal arrangements (→ **fig 15**).

Because of the friction, contacting seals will raise the temperature of the system and therefore cannot be considered for the majority of machine tool spindles. Thus, in general, contacting seals are used only where low speeds – n d_m below 200 000 – are involved and the influence of higher temperature has no important effects on the spindle working conditions.

Because of this, non-contacting seals are almost always used for high-precision bearing arrangements.

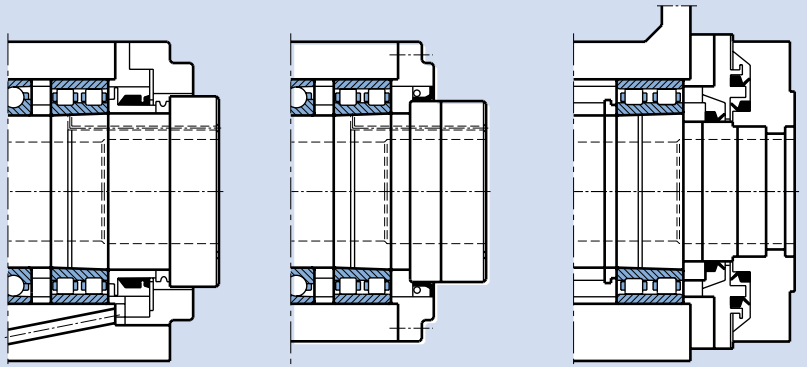
Non-contacting seals

Non-contacting seals are the most commonly used arrangements in machine tools applications, despite their being more difficult to manufacture, more expensive and more complex. Some examples of non-contacting seals are shown in **figs 16 and 17 page 58**.

Among the non-contacting seals, labyrinth seals are the most widely used in spindle applications. They make access to bearings and thus contamination difficult, and prevent cutting fluids from entering the bearing area. The main design features of a labyrinth seal, starting from the external side, are: splashguard, narrow gaps, large drainage chamber(s), and if there is adequate room, further gaps and drainage chambers. The splashguard prevents the fluids gaining access directly to the first gap. The gaps

Contacting seals, examples of application

Fig 15



prevent most of the fluid proceeding further in. The drainage chambers also serve to reduce the velocity of the fluid arising from the rotation of the shaft.

In order to avoid pumping effects inward, the labyrinth components should progressively decrease in diameter inwards from the outside. Machining spirals that can direct the fluid inwards should be avoided. If the spindle is designed to rotate in both directions – clockwise and counter clockwise – spirals have to be avoided. Additional protection is achieved by creating an overpressure inside the spindle. This is the case when oil spot or oil mist lubrication systems are used.

Under severe conditions, an air barrier can be created by blowing air into the labyrinth. It is important that the flow is balanced, so that the dominant flow is outwards. An air barrier can provide a reasonably efficient sealing even with a fairly simple labyrinth design. The following is an example of an efficient sealing system.

Non-contacting seals, examples

Fig 16

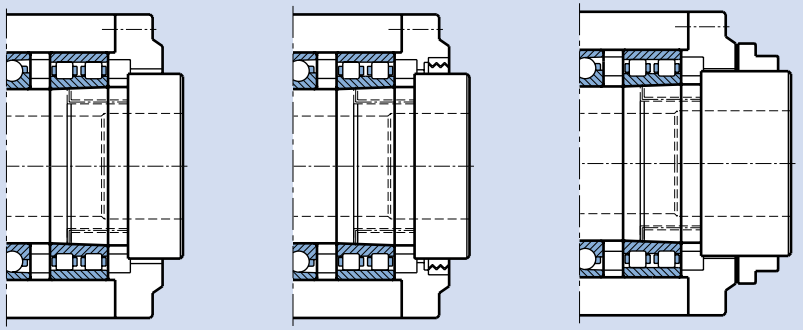
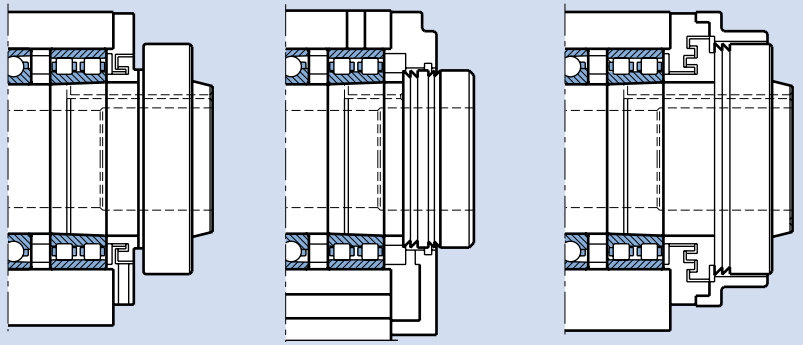


Fig 17



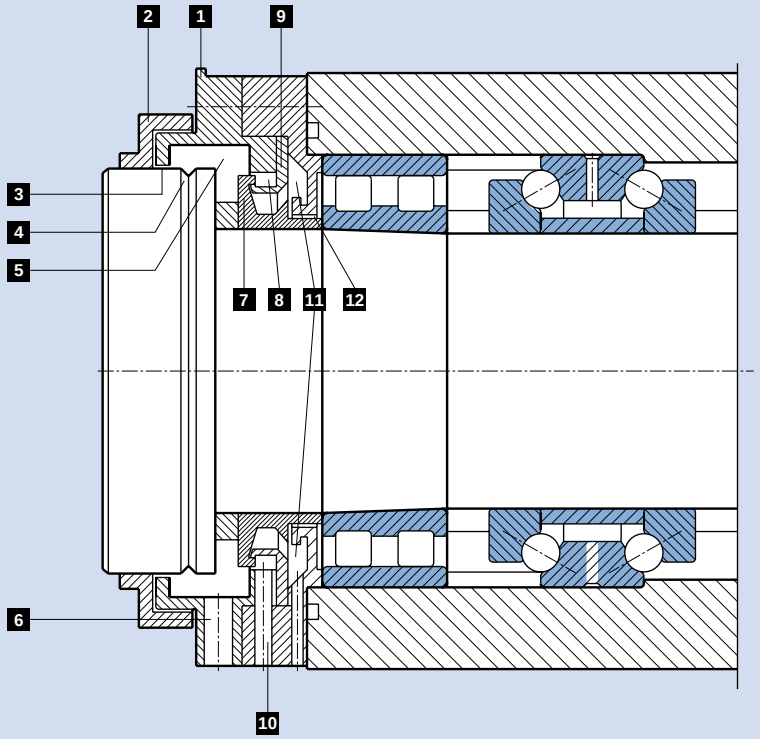
Preventive stages

The sealing system can be divided into the following preventive stages (➔ **fig** 18).

1 Direct access to the labyrinth of fluid washing over the spindle housing is prevented.
2 Splashguard, designed as a labyrinth, together with the housing cover, throws fluids outward. Both the splashguard and the housing cover are provided with one or several annular grooves to direct the fluid. When positioned on a rotating body, the grooves have little influence during rotation.
3 Gap with a height of 0,1 – 0,2 mm.
4 Groove or grooves on the shaft to direct the fluid under non-rotating conditions.

Labyrinth sealing system, example

Fig 18



5 Large drainage chamber where the velocity of the fluid is reduced. The chamber should be relatively large as the amount of fluid at this stage can be important.
6 Drainage using a large outlet area around 250 mm^2, so that no fluid stays inside the chamber (5).
7 Labyrinths with gap heights of 0,2 – 0,3 mm.
8 Chamber for fluid retardation.
9 Collector to guide the fluid to the lower side and prevent it from penetrating further.
10 Drainage using an area of 100 – 150 mm^2.
11 Chamber with collector and drainage as in steps (8) – (10). Only very little fluid should be present here and a drainage area of around 50 mm^2, should be adequate.
12 Gap with a height of about 1 mm to avoid capillary action.

This design is rather complex and more costly than conventional labyrinths, however, if the environmental conditions are severe, a very efficient sealing should be considered. The service life of the spindle will otherwise suffer, and downtime and replacement costs can be very high. A sealing system that takes up considerable space axially is favourable, as large drainage areas and collectors can be designed, thus improving sealing efficiency. However, the larger the space taken axially, the longer the overhang from the front bearing and the cutting force position, thus making the spindle radially less rigid.

Retaining covers
Retaining covers and their securing screws may be a source of deterioration in bearing accuracy. If the wall thickness between the bearing seating and the screw holes is too small, and/or the screws are tightened too hard, the outer ring raceway may be deformed. Bearings of the lightest ISO Dimension Series, 19 may be more seriously affected than the thicker rings of bearings belonging to ISO Dimension Series 10 or above.

It may be advantageous to use a large number of small diameter screws. Using only 3 or 4 screws should be avoided as such a small number of tightening points may produce lobes in the housing bore diameter. This can produce changeable friction torque, noise and unstable preload (when angular contact ball bearings are used). For spindles where the design is complex, space is limited, only a few screws can be used and thin-section bearings are used, an FEM (finite element) analysis may be recommended to accurately monitor deformation.

In addition, the play between the housing and the front cover should be checked. A guideline value may fall between 10 and 15 µm per 100 mm housing bore diameter (or bearing outside diameter) (→ **fig** 19).

Checking the play between the housing and the front cover

Fig 19

