Coding numerically

Coding alphabetically

FAG Rolling Bearings

Ball bearings · Roller bearings · Housings · Accessories

Catalogue WL 41 520/3 EA

1999 Edition

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For Technical Advice and Sales please see pages 709-714

FAG rolling bearing programme

This catalogue contains excerpts from the FAG rolling bearing programme for the industrial original equipment manufacture (OEM), distribution, and replacement demand.

With the products from this catalogue, most of which are produced in series, almost any application problem can be solved. To ensure quick availability of rolling bearings, housings and accessories, our stock-keeping programmes are constantly adapted to the requirements in your markets.

Your adavantages:

- fair market prices
- short delivery periods
- long-term availability
- long-term planning
- simplified stock-keeping

The current FAG product programme can be found in the current price list.

Enquiries should be directed to your FAG sales representative. (For addresses see page 709 et seq.)

FAG standardized rolling bearing programme

In the catalogue, priority is given to rolling bearings in DIN/ISO dimensions. This allows the designer to solve almost any application problem quickly and cost-effectively.

Moreover, FAG offer further rolling bearing types and design variations within outside diameters ranging from 3 millimetres to 4.25 metres.

FAG target industry programmes

FAG have compiled special programmes for certain branches of industry (page 693 et seq.).

In addition to the standardized rolling bearings, these programmes contain numerous special designs which offer efficient, cost-effective solutions for more complicated bearing applications.

To ensure product availability, please contact our Customer Service as early as possible to place orders. For technical questions and assistance, please contact our Application Engineers.

Continuous technical progress - refined life calculation - new speed indices - catalogue on CD-ROM

Evidence of continuous technical progress can be seen throughout the entire FAG rolling bearing programme. This catalogue reflects the quality improvements achieved in recent years which can be seen best in the new calculation method derived from the findings of FAG research on the dimensioning of bearings and the calculation of their rating life.

In the early eighties, FAG published new findings on the actually attainable rolling bearing life. The FAG method of adjusted life calculation was developed from these findings and is based on international standard recommendations, extensive investigations by the FAG fundamental research department, as well as practical experience. It takes into account failure probability, material, lubrication, magnitude of load, bearing

type, and cleanliness. It shows that fail-safe bearings can be a reality provided that a fully separating lubricant film, the highest degree of cleanliness, and realistic stressing are used. With the refined FAG calculation method introduced in the early nineties bearings can be safely dimensioned also for operation under contaminated lubricant conditions.

The suitability of rolling bearings for high speeds is generally determined by the permissible operating temperature. Therefore, the bearing tables show **reference speeds** which are determined by precisely defined and uniform criteria (reference conditions) on the basis of DIN 732 T1 (draft). If the operating conditions load, oil viscosity and permissible temperature deviate from the reference conditions, the **thermally permissible operating speed** can be assessed according to a method derived from DIN 732 T2 (draft). The **limiting speed**, on the other hand, takes into account mechanical limits such as the sliding velocity at rubbing seals or the strength of the bearing parts. The limiting speed may only be exceeded on consultation with FAG.

Version 1.1 of the **electronic FAG rolling bearing catalogue** is based on this printed catalogue. The programme on CD-ROM, however, is even more efficient and advantageous for the user. He is led to the best solution reliably and quickly in dialogue and saves a lot of work and time otherwise required for searching, selecting and calculating rolling bearings. Any background information can be fetched on-line in the form of texts, photos, drawings, diagrams, tables or animated pictures.

A CD-ROM will be available on request, with which bearings can be selected for a bearing location, a shaft or a shaft system.

Construction of the catalogue

In the first Section, "**Designing rolling bearing arrangements**", design engineers find, in a practical order, the data required for designing reliable and cost-effective bearing arrangements. It includes information applicable to all bearing types, for example, on dimensioning, bearing data, surrounding structures, lubrication and maintenance, mounting and dismounting.

In the second Section, "**FAG standardized rolling bearing programme**", type-specific details and explanations can be found on the pages preceding the individual bearing tables. The bearing tables of the second Section indicate dimensions, abutment dimensions, load ratings, speed indices and other technical data relevant to the bearing types.

Please note the comprehensive **FAG services programme** for more operational reliability (page 685 et seq.).

In another Section, the **FAG target industry programmes** are introduced. They are tailored to the specific requirements of machinery and installations. Target industry programmes contain standard bearings as well as special bearing types and designs.

Your **Technical Advice and Sales representatives** at FAG (see page 709 et seq. for addresses) will be gladly prepared to assist you in selecting suitable bearings and housings. They will provide you with technical publications mentioned in the catalogue. The publications give details on general topics concerning bearing technology such as mounting and dismounting, lubrication and maintenance, life calculation, etc., and they give details on special topics which cannot be dealt with in this catalogue.

Printed in Germany by Weppert GmbH & Co. KG, Schweinfurt.

The OEM/Distribution Business Unit of FAG Kugelfischer Georg Schäfer AG supplies rolling bearings, accessories and services to original equipment customers in the sectors of machinery and plant construction and to customers in the sectors distribution and replacement. With their extensive know-how, competent advice and comprehensive customer services, FAG are a most important partner of their customers. Development and further development of our products are guided by the requirements of practical operation. In the ideal case, the spectrum of requirements is defined jointly by our researchers, application engineers, the machine producers and users. This is the basis for technically and economically convincing solutions.

The Business Unit produces at locations in Germany, Italy, Portugal, India, Korea and the USA. The market is supplied through subsidiaries and trading partners in nearly all countries of the world.

Contents

Designing rolling bearings arrangements

A long service life, a high degree of reliability and economic efficiency are the chief aims when designing rolling bearing arrangements. To reach these, design engineers draw up in specifications the conditions influencing the bearings and the requirements they have to meet. Not only the correct bearing type, bearing design and bearing arrangement must be selected when designing but also the surrounding parts, that is the shaft, housings and fastening parts, the sealing and particularly the lubrication, all of which have to be adapted to the influencing factors in the specifications.

The steps involved in designing a bearing arrangement generally follow the same order. First, an accurate survey of all influencing factors, should be made. Then the type, arrangement and size of the bearings are chosen and alternatives are reviewed. The complete bearing arrangement is then laid down in the design drawing which means bearing data (main dimensions, tolerances, bearing clearance, cage, code number) the connection parts (fits, fastening, sealing) and the lubrication. Mounting and maintenance are also taken into consideration. In order to select the most economic bearing arrangement, the degree to which alternative solutions take the influencing factors in the specifications into account is compared as well as the total cost arising.

Influences

The following data should be known:

- Machine/device and bearing locations (sketch)
- Operating conditions (load, speed, mounting space, temperature, environmental conditions, shaft arrangement, rigidity of the mating parts)
- Requirements (life, precision, noise, friction and operating temperature, lubrication and maintenance, mounting and dismounting)
- Commercial data (deadlines, numbers of items)

Before designing the bearing arrangement, the following influencing factors should be considered:

- Load and speed
  How high are radial and axial forces? Does the direction change? How high is the speed? Does the direction of rotation change? Do shock loads occur? How should the correlations between load and speed and their time shares be taken into consideration when dimensioning?
- Mounting space
  Is the mounting space firmly specified? Can dimensions be changed without the function of the machine being impaired?
- Temperature
  How high is the ambient temperature? Is external heating or cooling to be expected (temperature gradient between the bearing rings)? Which length variations may be expected as a result of thermal expansion (floating bearing)?
- Environmental conditions
  Is humidity high? Does the bearing arrangement have to be protected from more dirt? What about aggressive media? Are vibrations transferred to the bearings?
- Shaft arrangement
  Are the shafts horizontal, vertical or inclined?
- Rigidity of the mating parts
  Does a housing deformation have to be taken into consideration? May misalignment of the bearings be expected because of the shaft deflection?
- Life
  What is the required life? Can the bearing arrangement be compared with another proven bearing arrangement (nominal life L_h, index of dynamic stressing f_L)? Is the adjusted life calculation (which should always be preferred due to the greater closeness of the results to real operating conditions) to be applied?
- Precision
  Are greater demands made on the running accuracy, e.g. with machine tool bearing arrangements?
- Noise
  Is particularly low noise required, e.g. in the case of electric equipment in houshold appliances?
- Friction and operating temperature
  Should the energy loss be particularly slight? Must the temperature increase be limited, so that precision is not endangered?

- Lubrication and maintenance
  Are certain conditions, e.g. oil sump lubrication or circulation lubrication, specified for bearing lubrication?
  Does lubricant escape have to be prevented from the bearings in order to ensure the quality of the manufacturing process, e.g. in food industry? Is there a central supply of lubricant? Should the bearings be maintenance-free?
- Mounting and dismounting
  Is special mounting equipment required? Is the inner ring mounted on a cylindrical shaft or on a tapered shaft? Should the bearings be seated directly on the shaft or be fastened with adapter or withdrawal sleeves? Does dismounting occur frequently, e.g. with rolling mill bearings?
- Commercial data
  How high is demand? When should the bearings be available? Can basic designs (see FAG price list) which can be supplied on the short term be used? Are variants of basic bearing designs required or are new designs necessary in the case of special operational conditions? The FAG customer service representative informs you on price and delivery time for these bearings.

These influences are taken into account for each of the following steps for the bearing arrangement draft:

- Choice of the bearing type
- Choice of the bearing arrangement
- Determination of the bearing size (life, index of static safety)
- Definition of the bearing data
- Structure of the surrounding parts
- Lubrication and maintenance
- Mounting and dismounting

In most cases the extent of work and cost required for a bearing arrangement draft is relatively slight as past experience with comparable bearing arrangements can be applied. The data of this catalogue refer to such applications.

New bearing arrangements or extreme conditions frequently require more extensive calculations and constructive action which cannot be presented in this catalogue. In such cases FAG services should be availed of. Specialized publications are also available for many applications. They are indicated in various places in the catalogue.

PC programmes

Version 1.1 of the **electronic FAG rolling bearing catalogue** is based on this printed catalogue. The programme on CD-ROM, however, is even more efficient and advantageous for the user. He is led to the best solution reliably and quickly in dialogue and saves a lot of work and time otherwise required for searching, selecting and calculating rolling bearings.

Ordering code: CD41520/3D-E.

A CD-ROM will be available on request for selecting and calculating rolling bearings for a bearing location, a shaft or a shaft system.

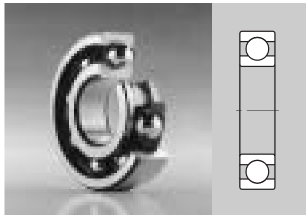
Details on PC programmes can be found in Section "FAG services programme", page 689 et seq.

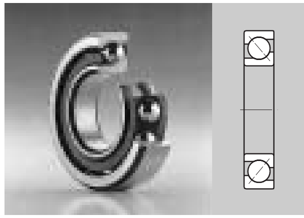
Bearing Types
Ball bearings

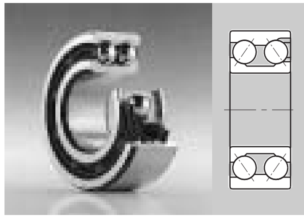
Selection of bearing type

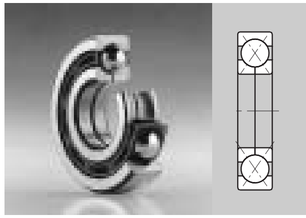
The FAG delivery programme contains a multitude of bearing types from which design engineers can select those most suitable for their requirements. Ball bearings and roller bearings are differentiated by the type of rolling elements. The following tables show examples:

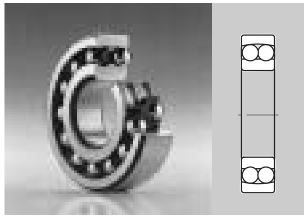
▼ Ball bearings

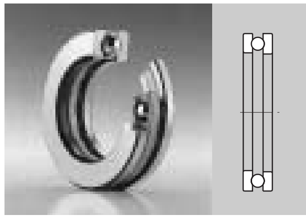

Deep groove ball bearing
single row

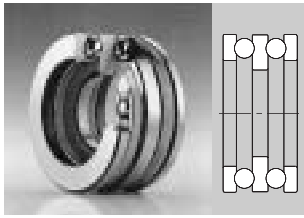

Angular contact ball bearing
single row

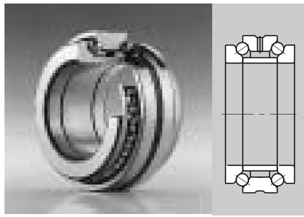

Angular contact ball bearing
double row


Four-point bearing


Self-aligning ball bearing

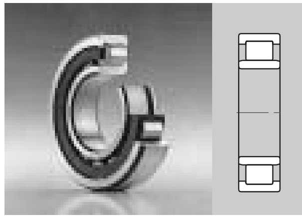

Thrust ball bearing
single direction

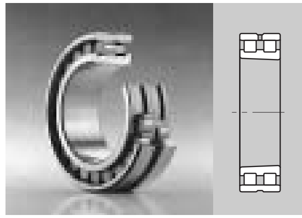

Thrust ball bearing
double direction

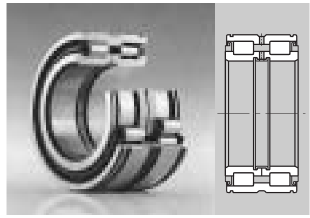

Angular contact thrust ball bearing
double direction

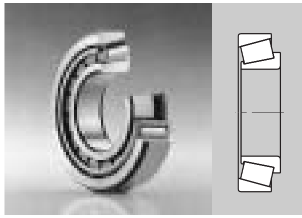
Bearing Types
Roller bearings

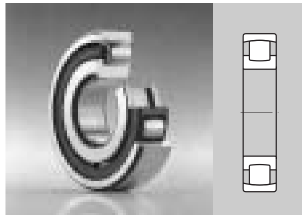
▼ Roller bearings

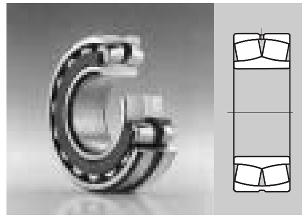

Cylindrical roller bearing
single row

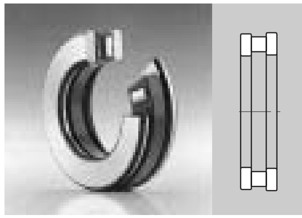

Cylindrical roller bearing
double row

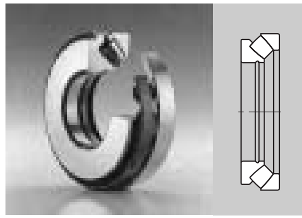

Cylindrical roller bearing
double row, full complement


Tapered roller bearing


Barrel roller bearing


E design spherical roller bearing


Cylindrical roller thrust bearing


Spherical roller thrust bearing

The most important characteristics of each bearing type are summarized in the overview on pages 20 to 23. They are, however, only a rough guide for selection. Several criteria have to be weighed prior to deciding on one certain type. A lot of requirements can be met, for example, with deep groove ball bearings. They accommodate medium radial loads and also axial loads, are suitable for very high speeds and run quietly. Deep groove ball bearings are also available with dust shields and seals. As they are very reasonably priced as well, they are used more than any other bearing.

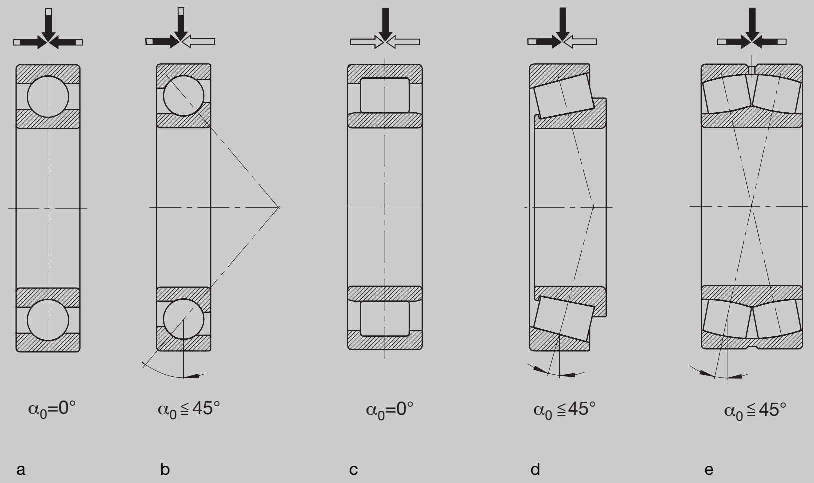
More details on the characteristics of the bearing types and designs possible can be found on the pages prior to the individual sections of the tables.

Radial load

Bearings which are chiefly used for radial loads are referred to as radial bearings. They have a nominal contact angle $\alpha_0 \leqq 45°$. Roller bearings are suitable for higher radial loads than ball bearings of the same size.

Cylindrical roller bearings of the designs N and NU may only be loaded radially. The radial bearings of the other types accommodate both radial and axial loads.

▼ Radial bearings with a nominal contact angle $\alpha_0 \leqq 45°$ predominantly for radial loads
a = deep groove ball bearing, b = angular contact ball bearing, c = cylindrical roller bearing NU, d = tapered roller bearing, e = spherical roller bearing

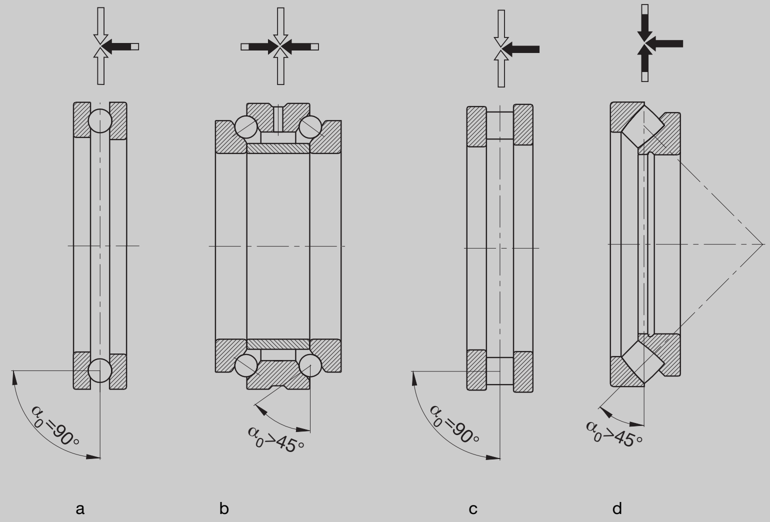


Axial load

Bearings which are chiefly for axial loads (axial bearing) have a nominal contact angle $\alpha_0 > 45°$. Thrust ball bearings and angular contact thrust ball bearings can accommodate axial forces in one or both directions depending on the design. For especially high axial loads, cylindrical roller thrust bearings or spherical roller thrust bearings are given preference.

Spherical roller thrust bearings and single-direction angular contact thrust ball bearings accommodate combined axial and radial loads. The remaining thrust bearing types are only suitable for axial loads.

▼ Axial bearings with a nominal contact angle $\alpha_0 > 45°$ predominantly for axial loads
a = thrust ball bearing, b = angular contact thrust ball bearing, c = cylindrical roller thrust bearing, d = spherical roller thrust bearing



Length compensation within the bearing

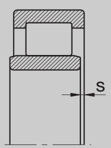
Usually, a locating bearing and a floating bearing are used for the bearing arrangement of a shaft. The floating bearing compensates for axial length tolerances and heat expansion.

Cylindrical roller bearings of the designs NU and N are the ideal floating bearings. Length differences are compensated for in these bearings themselves. The bearing rings can be firmly fitted.

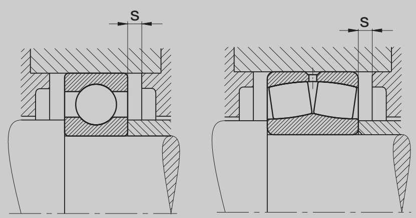
Length compensation with sliding fit

Non-separable bearings, such as deep groove ball bearings and spherical roller bearings, are also mounted as floating bearings. One of the two bearing rings is then provided with a loose fit and needs no axial mating surface, so that the loose outer ring can move in the housing bore and the loose inner ring on the shaft seat.

▼ Displacement (s) in the bearing is possible with cylindrical roller bearings

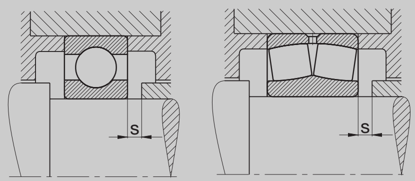


▼ A loose fit in the housing bore makes axial displacement (s) of the deep groove ball bearing (a) and of the spherical roller bearing (b) possible



a b

▼ A loose fit on the shaft of the deep groove ball bearing (a) and of the spherical roller bearing (b) makes axial displacement (s) possible



a b

Separable bearings

These are bearings whose rings can be mounted separately. This is advantageous where both rings have tight fits.

Examples: Four-point bearings, double row angular contact ball bearings with split inner ring, cylindrical roller bearings, tapered roller bearings, thrust ball bearings, cylindrical roller thrust bearings and spherical roller thrust bearings.

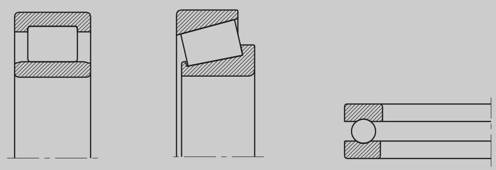
Non-separable bearings: Deep groove ball bearings, single-row angular contact ball bearings, self-aligning ball bearings, barrel roller bearings and spherical roller bearings.

Precision

The normal dimensional and running precision of rolling bearings (tolerance class PN) is sufficient for most applications. When requirements are high, for example, in machine tool spindles, bearings must have a higher degree of precision. For this purpose the tolerance classes P6, P6X, P5, P4, and P2 have been standardized. The tolerance classes P4S, SP, and UP according to FAG plant standards also exist for individual bearing types.

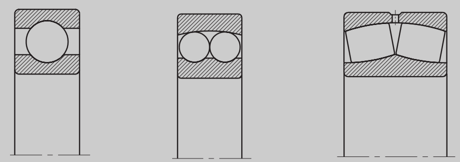
FAG deliver the following bearings with increased precision: Spindle bearings, cylindrical roller bearings, and angular contact thrust ball bearings (see publication no. AC 41 130 "Super Precision Bearings"). The tolerance classes for each are indicated in the introduction sections to the tables.

▼ Separable cylindrical roller bearing (a), tapered roller bearing (b) and thrust ball bearing (c)



a b c

▼ Non-separable deep groove ball bearing (a), self-aligning ball bearing (b) and spherical roller bearing (c)



a b c

Bearing Types
Compensation of misalignments · Speeds · Low-noise operation

Compensation of misalignments

Misalignment can occur when machining the bearing seats of a shaft or a housing, particularly when the seats are not machined in one setting. Misalignment can also be expected when using single housings, such as flanged or plummer block housings. Tilting of the bearing rings due to shaft deflection as a result of the operating load also leads to misalignment.

Self-aligning bearings such as self-aligning ball bearings, barrel roller bearings, radial and axial spherical roller bearings, compensate for misalignment and tilting. The bearings have a hollow spherical outer ring raceway in which the inner ring together with the rolling element set can swivel out. The angle of alignment of these bearings depends on their type and size as well as load.

S-type bearings and thrust ball bearings with a seating washer have a spherical support area; they can adjust themselves during mounting in the hollow spherical mating surface.

Values for the permissible angles of alignment are to be taken from the introduction preceding the tables for each bearing type.

Speeds

The reference speeds and limiting speeds listed in the dimensional tables indicate the suitability of the bearings for high speeds. Single-row bearing types with particularly low friction reach the highest speeds. These are deep groove ball bearings with radial load only and angular contact ball bearings with combined load.

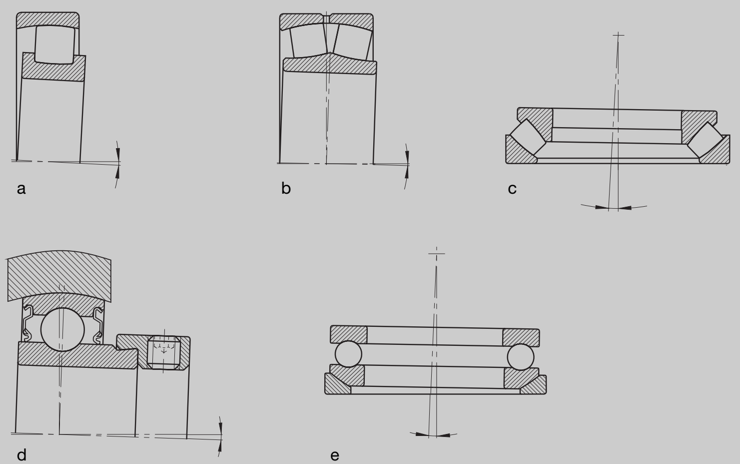
Increased dimensional and running precision of bearing and mating parts, cooling lubrication, and special cage types and cage materials generally have a positive effect on the speed suitability of the bearings.

Axial bearings allow lower speeds than radial bearings. See section "Suitability for high speeds" on page 87 for further details.

Low-noise operation

Low noise is frequently required for small electrical machines, office machines, household appliances etc. FAG deep groove ball bearings are especially suitable for such applications. These bearings run so quietly that no special design is required for low noise. Axial adjustment of the bearings with springs is advantageous.

▼ Self-aligning rolling bearings:
Barrel roller bearing (a), spherical roller bearing (b), spherical roller thrust bearing (c);
S-type bearing (d) and thrust ball bearing with a seating washer (e) have a spherical support area

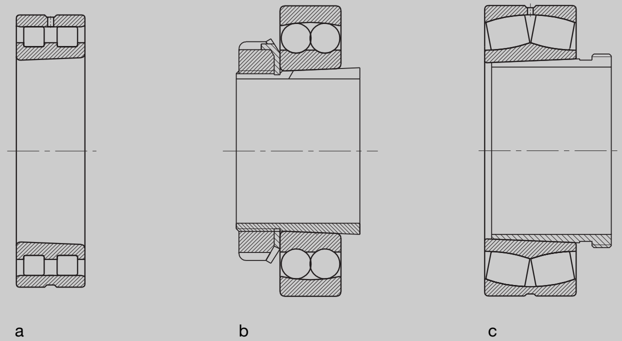


Bearing Types
Tapered bore · Sealed bearings · Rigidity · Friction

Tapered bore

Bearings with a tapered bore can be mounted directly onto a tapered shaft seat, e.g. single and double row cylindrical roller bearings in precision design. When mounting these bearings a defined radial clearance can be set.

▼ Bearings with tapered bore: a = double row cylindrical roller bearing, b = self-aligning ball bearing with adapter sleeve, c = spherical roller bearing with withdrawal sleeve



At moderate demands on the running accuracy, mainly self-aligning ball bearings, barrel roller bearings, and spherical roller bearings with a tapered bore are fixed on a cylindrical shaft seat with adapter or withdrawal sleeves. It is particularly easy to mount and dismount such bearings.

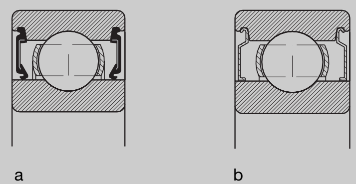
Sealed bearings

FAG deliver rolling bearings with seals at one or both sides. Bearings with rubbing sealing washers (also see page 125) or with non-rubbing dust shields (also see page 124) allow the construction of plain designs. Sealed bearings which are provided with a grease filling by the manufacturer are listed on page 130 under "Grease supply to bearings". The most common examples are deep groove ball bearings of the designs .2RSR (sealing washers at both sides) and .2ZR (dust shields at both sides).

Rigidity

By rigidity we mean, the elastic deformation in the bearing under load. Particularly high system rigidity is desirable in the case of main spindle bearings in machine tools and pinion bearing arrangements. Due to the contacting conditions between rolling elements and raceways the rigidity of roller bearings is higher than that of ball bearings. To increase rigidity, spindle bearings, for example, are preloaded (also see FAG publ. no. AC 41 130).

▼ Deep groove ball bearing sealed on both sides with seals (a) and dust shields (b)



Friction

In addition to heat supply and dissipation, bearing friction is a particularly decisive factor for the operating temperature of bearings. Examples of low-friction bearings are: deep groove ball bearings, single row angular contact ball bearings and caged cylindrical roller bearings under radial load. Relatively high friction may be expected in the case of bearings with rubbing seals, full complement cylindrical roller bearings and axial roller bearings. The calculation of the frictional moment is described on page 96.

Bearing Types

Synoptic table: Bearing types and their characteristics

Suitability

- ● very good
- ◕ good
- ◑ normal/possible
- ◔ limited
- ○ not suitable/not applicable

Characteristics:

Bearing type	Radial loadability	Axial loadability in both directions	Length compensation within the bearing	Length compensation with sliding fit	Separable bearings	Compensation of misalignment	Increased precision	Suitability for high speeds	Quiet running	Tapered bore	Seal at one or both sides	High rigidity	Low friction	Locating bearings	Floating bearings
Deep groove ball bearings	◕	◑	○	◑	○	◔	◑	●	●	○	◕	◑	●	◕	◑
Angular contact ball bearings	◕	◕ ←	○	◑ a	○	○	◑	● c	◕	○	○	◕ a	◕	● a	◑ a
Angular contact ball bearings double row	◕	◕	○	◑	◑	○	◑	◑	◔	○	◕	◕	◑	◕	◑
Spindle bearings	◕	◕ ←	○	◑ a	○	○	●	● c	●	○	○	◕ a	●	● a	◑ a
Four-point bearings	◔	◕	○	○	◑	○	◔	◔	◔	○	○	◑	◕ e	◕	○
Self-aligning ball bearings	◕	◔	○	◑	○	●	○	◕	◔	● d	◕	◔	●	◑	◑
Cylindrical roller bearings NU, N	●	○	●	○	●	◔	◕	●	◑	◑	○	◕	◕ f	○	●
NJ	●	◑ ←	◔	○	●	◔	◑	◕ b	◔	○	○	◕	◕ b	◑	◑
NUP, NJ + HJ	●	◑	○	◑	●	◔	◑	◕ b	◔	○	○	◕	◕ b	◕	◔
NN	●	○	●	○	●	○	●	●	◑	●	○	●	●	○	●
NCF, NJ23VH	●	◑ ←	◔	◑	◑	◔	○	○	○	○	○	●	○	◑	◑
NNC, NNF	●	◑	○	◑	○	○	○	○	○	○	◕	●	○	◑	◑

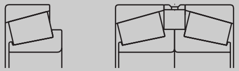
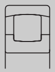
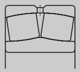
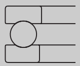
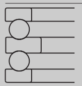
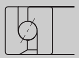
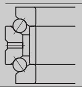
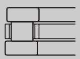
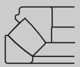
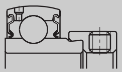
← Single bearings and bearings in tandem arrangement in single direction   a) for paired mounting   b) for low axial load   c) limited suitability for paired mounting   d) also with adapter or withdrawal sleeves   e) axial load only   f) very good for narrow series

Bearing Types
Synoptic table: Bearing types and their characteristics

Suitability

- ● very good
- ◕ good
- ◑ normal/possible
- ◔ limited
- ○ not suitable/not applicable

Characteristics:

Bearing type		Radial loadability	Axial loadability in both directions	Length compensation within the bearing	Length compensation with sliding fit	Separable bearings	Compensation of misalignment	Increased precision	Suitability for high speeds	Quiet running	Tapered bore	Seal at one or both sides	High rigidity	Low friction	Locating bearings	Floating bearings
Tapered roller bearings		●	● ←	○	◑ a	●	◔	◕	◑ c	◔	○	○	● a	◕	● a	◔ a
Barrel roller bearings		●	◑	○	◑	○	●	○	◑	◔	● d	○	◕	◑	◕	◑
Spherical roller bearings		●	◕	○	◑	○	●	○	◑	◔	● d	◑	◕	◕	◕	◑
Thrust ball bearings		○	◕ ←	○	○	●	◑ g	◕	◑	◔	○	○	◑	◕	◑	○
		○	◕	○	○	●	◑ g	○	◔	○	○	○	◑	◔	◑	○
Angular contact thrust ball bearings		◔	◕ ←	○	○	○	◔	●	◕ c	◔	○	○	◕ a	◑	● a	○
		○	◕	○	○	●	○	●	●	◔	○	○	●	◑	●	○
Cylindrical roller thrust bearings		○	● ←	○	○	●	○	◑	◔	○	○	○	◕	○	◕	○
Spherical roller thrust bearings		◔	● ←	○	○	●	●	○	◔	○	○	○	◕	○	◕	○
S-type bearings		◕	◑	◔	◔	○	◑ g	○	◔	○	○	●	◑	○	◑	○

← Single bearings and bearings in tandem arrangement in single direction

a) for paired mounting

c) limited suitability for paired mounting

d) also with adapter or withdrawal sleeves

g) S-type bearings and thrust ball bearings with seating washer compensate for misalignment during mounting

Selection of bearing arrangement

In order to guide and support a rotating shaft, at least two bearings are required which are arranged at a certain distance from each other. A bearing arrangement with locating and floating bearings, with adjusted bearings or with floating bearings can be selected, depending on the case.

Locating-floating bearing arrangement

Due to machining tolerances the centre distances between the shaft seats and the housing seats are often not exactly the same if a shaft is supported by two radial bearings. Warming-up during operation also causes the distances to change. These differences in distance are compensated for in the floating bearing.

Cylindrical roller bearings of N and NU designs are ideal floating bearings. These bearings allow the roller and cage assembly to shift on the raceway of the lipless bearing ring.

All other bearing types, e.g. deep groove ball bearings and spherical roller bearings only function as floating bearings when one bearing ring is provided with a loose fit. The ring under point load (see table on page 104) is therefore given a loose fit; this is generally the outer ring.

The locating bearing, on the other hand, guides the shaft axially and transmits external axial forces. For shafts with more than two bearings, only one bearing is designed as a locating bearing in order to avoid detrimental axial preload.

The bearing to be designed as a locating bearing depends on how high the axial load is and how accurately the shaft must be axially guided.

Closer axial guidance is achieved, for example, with a double row angular contact ball bearing than with a deep groove ball bearing or a spherical roller bearing. A pair of symmetrically arranged angular contact ball bearings or tapered roller bearings provides extremely close axial guidance when designed as locating bearings.

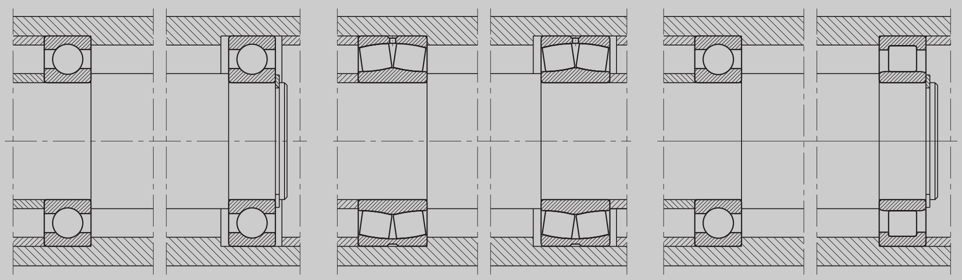
Angular contact ball bearings of universal design are especially advantageous. The bearings can be paired at will without shims in O or X arrangement. Angular contact ball bearings of the universal design are finished in such a way that when mounting in an X or O arrangement, they have a low axial clearance (UA design), a zero clearance (UO) or a light preload (UL).

Spindle bearings of the universal design UL have a light preload when mounted in an X or O arrangement (designs with more preload available upon request).

Mounting is also facilitated by matched tapered roller bearings as a locating bearing (design N11). They are paired with an axial clearance in such a way that neither setting nor adjusting jobs are required. In the case of transmissions, a four-point bearing is sometimes mounted directly next to a cylindrical roller bearing in such a way that a locating bearing results. A four-point bearing whose outer ring is not supported radially can only transfer axial forces. The cylindrical roller bearing takes on the radial load.

A cylindrical roller bearing of the NUP design can also be used as a locating bearing when the axial force is low.

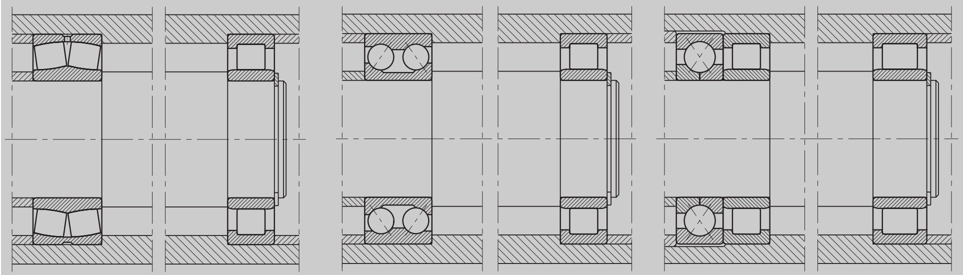
▼ Examples of a locating-floating bearing arrangement



a. Locating bearing: Deep groove ball bearing — Floating bearing: Deep groove ball bearing

b. Locating bearing: Spherical roller bearing — Floating bearing: Spherical roller bearing

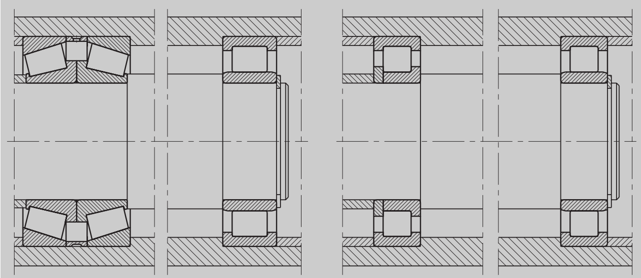
c. Locating bearing: Deep groove ball bearing — Floating bearing: Cylindrical roller bearing NU



d. Locating bearing: Spherical roller bearing — Floating bearing: Cylindrical roller bearing NU

e. Locating bearing: Double row angular contact ball bearing — Floating bearing: Cylindrical roller bearing NU

f. Locating bearing: Four-point bearing and cylindrical roller bearing NU — Floating bearing: Cylindrical roller bearing NU

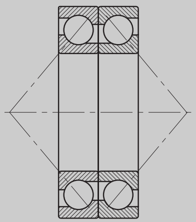


g. Locating bearing: Two tapered roller bearings — Floating bearing: Cylindrical roller bearing NU

h. Locating bearing: Cylindrical roller bearing NUP — Floating bearing: Cylindrical roller bearing NU

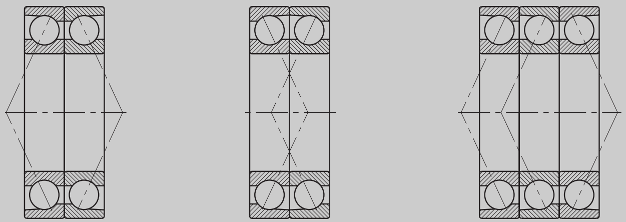
Bearing Arrangement
Locating-floating bearing arrangement

▼ Angular contact ball bearing pair of universal design as locating bearing
a = O arrangement, b = X arrangement



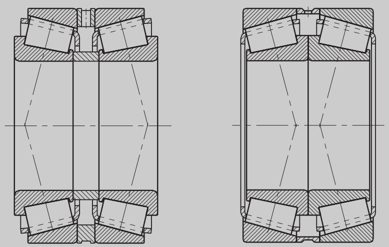
a b

▼ Spindle bearings of universal design as locating bearing
a = O arrangement, b = X arrangement, c = tandem-O arrangement



a b c

▼ Tapered roller bearing pair as locating bearing
a = O arrangement, b = X arrangement



a b

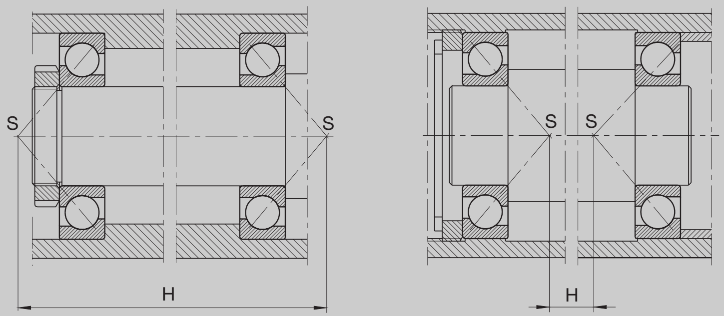
Bearing Arrangement
Adjusted bearing arrangement

Adjusted bearing arrangement

As a rule, an adjusted bearing arrangement consists of two symmetrically-arranged angular contact ball bearings or tapered roller bearings. During mounting, a bearing ring is displaced on its seat until the bearing arrangement has the appropriate clearance or the required preload. This means that the adjusted bearing arrangement is particularly suitable for those cases in which a close guidance is required, for example, for pinion bearing arrangements with spiral toothed bevel gears and spindle bearing arrangements in machine tools. In principle, bearings either in an O arrangement or an X arrangement may be selected.

In the O arrangement, the apexes S of the cone formed by the contact lines point outward while those of the X arrangement point inward. The spread H, i.e. the distances between the pressure cone apexes, is larger in the O arrangement than in the X arrangement. The O arrangement provides a smaller tilting clearance.

▼ Adjusted bearing arrangement with angular contact ball bearings in O arrangement (a)
Adjusted bearing arrangement with angular contact ball bearings in X arrangement (b)



a b

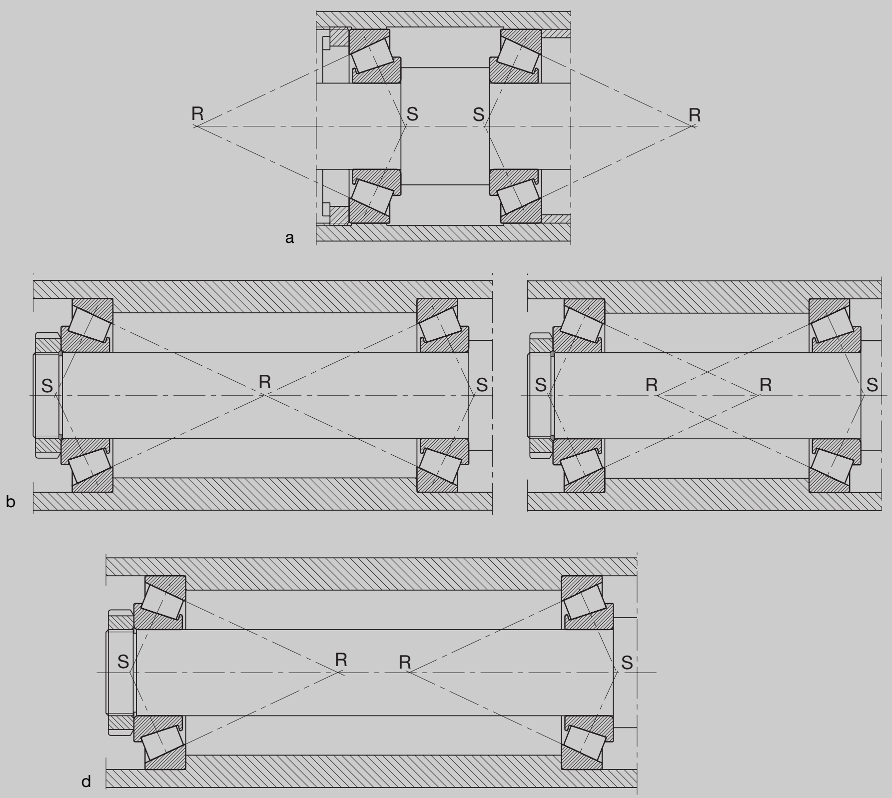
Bearing Arrangement
Adjusted bearing arrangement

Thermal expansion must be taken into consideration when setting the axial clearance. In the X arrangement (a) a temperature gradient running from the shaft to the housing always leads to a reduction of clearance (conditions: same material for shaft and housing, same temperature of inner rings and entire shaft, same temperature of outer rings and entire housing).

In the O arrangement, on the other hand, a distinction is made between three cases. If the roller cone apexes (R), i.e. the points where the bearing centre line intersects the projection of the inclined outer ring raceway, coincide at one point (b), the adjusted bearing clearance is maintained under the above-mentioned conditions.

If the roller cones (c) overlap when bearing distance is short the axial clearance decreases as a result of heat expansion. If they do not come in contact when the distance is great (d), the axial clearance increases as a result of heat expansion.

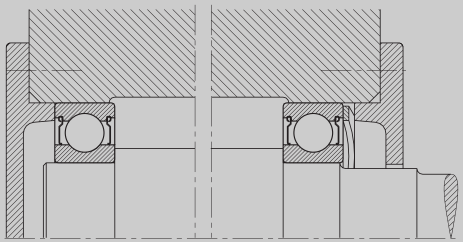
▼ Adjusted bearing arrangement with tapered roller bearings in X arrangement (a) and their roller cone apexes.
Adjusted bearing arrangement with tapered roller bearings in O arrangement,
when the roller cone apexes coincide (b),
when the roller cone apexes overlap (c),
when the roller cone apexes do not overlap (d).



Bearing Arrangement
Adjusted bearing arrangement · Floating bearing arrangement

Adjusted bearing arrangements are also possible by preloading with springs. This elastic type of adjustment compensates for heat expansion. They are also used when bearings are in danger of vibrations when stationary.

▼ Adjusted deep groove ball bearings preloaded with spring washer



Floating bearing arrangement

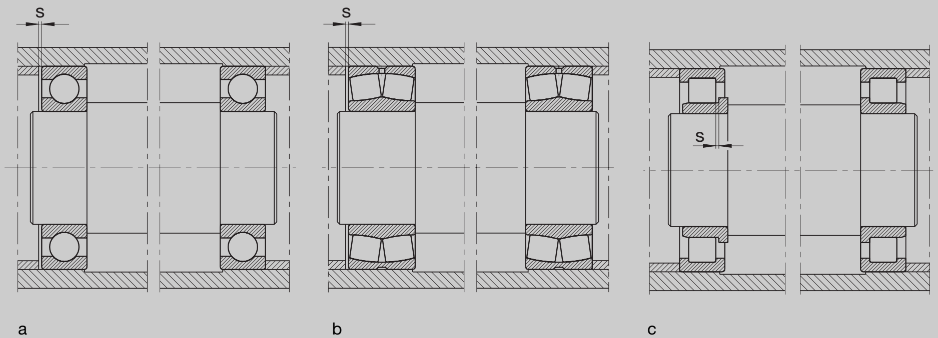
The floating bearing arrangement is an economical solution where a close axial guidance of the shaft is not required. Its design is similar to that of the adjusted bearing arrangement. In a floating bearing arrangement, the shaft, however, can shift by the axial clearance s relative to the housing. The value s is determined depending on the guiding accuracy so that detrimental axial preloading of the bearings is prevented even under unfavourable thermal conditions.

Deep groove ball bearings, self-aligning ball bearings and spherical roller bearings, for example, are bearing types which are suitable for the floating bearing arrangement. One ring of both bearings - generally the outer ring - is fitted to allow displacement.

In floating bearing arrangements with NJ cylindrical roller bearings, length is compensated for in the bearings. Inner and outer rings can be given a tight fit.

Tapered roller bearings and angular contact ball bearings are not suitable for a floating bearing arrangement because they must be adjusted for flawless running.

▼ Examples of a floating bearing arrangement
a = two deep groove ball bearings, b = two spherical roller bearings, c = two cylindrical roller bearings NJ,
s = axial clearance



Dimensioning

In numerous cases, the bore diameter of the bearings is already specified by the whole construction of the machine or device. Whether requirements on life, static safety and cost efficiency have been fulfilled should be checked by means of a dimensioning calculation prior to finally determining the remaining main dimensions and bearing type. This calculation involves the comparison of a bearing's load with its load carrying capacity.

A differentiation is made between dynamic and static stress in rolling bearing engineering.

Static stress implies that there is no relative movement or a very slow one between the rings ($n < 10\ \text{min}^{-1}$). For these conditions the safety against excessive plastic deformations of the raceways and rolling elements is checked.

Most bearings are dynamically stressed. Their rings turn relatively to each other. The dimensioning calculation checks the safety against premature material fatigue of the raceways and rolling elements.

Only in rare cases does the nominal life calculation according to DIN ISO 281 state the life which is actually attainable. Cost-effective constructions, however, demand that the bearing's capacity is utilized as much as possible. The greater the utilization the more important a careful bearing dimensioning. The FAG calculation method for the attainable life, which takes the operating and environmental effects into consideration, has proven effective. The method is based on DIN ISO 281 and on the findings published by FAG in 1981 on the endurance strength of rolling bearings. Since then the calculation method has been refined to such an extent that bearings can be designed for reliable operation even in the case of contaminated lubricant.

The dynamic and static load ratings given in this catalogue apply to rolling bearings of chromium steel, which were subjected to standard heat-treatment, only in the usual operating temperature range of up to 100 °C. The minimum hardness of raceways and rolling elements is then 58 HRC.

Higher operating temperatures reduce the material hardness resulting in a drastic loss of the load carrying capacity of the bearing. Please consult the FAG Application Engineering in such cases.

Statically stressed bearings

The calculation of the index of static stressing f_s serves to ascertain that a bearing with adequate load rating has been selected.

$$f_s = \frac{C_0}{P_0}$$

where

f_s index of static stressing
C_0 static load rating [kN]
P_0 equivalent static load [kN]

The index of static stressing f_s is a safety factor against permanent deformations of the contact areas of the rolling elements. A high f_s value is required for bearings which must run smoothly and particularly quietly. Smaller values suffice when a moderate degree of running quietness is required. The following values are generally recommended:

$f_s = 1.5 \ldots 2.5$ for a high degree
$f_s = 1.0 \ldots 1.5$ for a normal degree
$f_s = 0.7 \ldots 1.0$ for a moderate degree

Values recommended for spherical roller thrust bearings and precision bearings are shown in the tables.

The static load rating C_0 [kN] according to DIN ISO 76 - 1988, is indicated in the tables for every bearing. This load (a radial one for radial bearings, an axial and centrical one for axial bearings) at the centre of the most heavily loaded contact area between rolling element and raceway causes a theoretical contact pressure p_0 of

- 4600 N/mm² for self-aligning ball bearings
- 4200 N/mm² for all other ball bearings
- 4000 N/mm² for all roller bearings

Under the C_0 load (corresponding to $f_s = 1$) a plastic total deformation of rolling element and raceway of about $^1/_{10\,000}$ of the rolling element diameter at the most heavily loaded contact area arises.

The equivalent static load P_0 [kN] is a theoretical value. It is a radial load for radial bearings and an axial and centrical load for thrust bearings. P_0 causes the same stress at the centre of the most heavily loaded contact area of rolling element/raceway as the actual load combination.

$$P_0 = X_0 \cdot F_r + Y_0 \cdot F_a \quad \text{[kN]}$$

where

P_0 equivalent static load [kN]
F_r radial load [kN]
F_a axial load [kN]
X_0 radial factor
Y_0 thrust factor

The values for X_0 and Y_0 as well as information on the calculation of the equivalent static load for the various bearing types can be found in the bearing tables or their preceding texts.

Dynamically stressed bearings

The standardized calculation method (DIN ISO 281) for dynamically stressed rolling bearings is based on material fatigue (formation of pitting) as the cause of failure. The life formula is:

$$L_{10} = L = \left(\frac{C}{P}\right)^p \quad [10^6 \text{ revolutions}]$$

where

$L_{10} = L$ nominal rating life [10^6 revolutions]
C dynamic load rating [kN]
P equivalent dynamic load [kN]
p life exponent

L_{10} is the nominal rating life in millions of revolutions, which is reached or exceeded by at least 90 percent of a large group of identical bearings.

The dynamic load rating C [kN] according to DIN ISO 281 - 1993, is indicated in the tables for every bearing. With this load an L_{10} rating life of 10^6 revolutions is reached.

The equivalent dynamic load P [kN] is a theoretical value. It is a radial load for radial bearings or axial load for axial bearings, which is constant in size and direction. P yields the same life as the actual load combination.

$$P = X \cdot F_r + Y \cdot F_a \quad \text{[kN]}$$

where

P equivalent dynamic load [kN]
F_r radial load [kN]
F_a axial load [kN]
X radial factor
Y thrust factor

The values for X and Y as well as information on the calculation of the equivalent dynamic load for the various bearing types can be found in the bearing tables or their preceding texts.

The life exponent p differs for ball bearings and roller bearings.

$p = 3$ for ball bearings

$p = \frac{10}{3}$ for roller bearings

When the bearing speed is constant, the life can be expressed in hours:

$$L_{h10} = L_h = \frac{L \cdot 10^6}{n \cdot 60} \,[\text{h}]$$

where

$L_{h10} = L_h$ nominal rating life [h]
L nominal rating life [10^6 revolutions]
n speed (revolutions per minute) [min^{-1}].

On converting the equation we obtain:

$$L_h = \frac{L \cdot 500 \cdot 33\tfrac{1}{3} \cdot 60}{n \cdot 60}$$

$$\frac{L_h}{500} = \left(\frac{C}{P}\right)^p \cdot \left(\frac{33\tfrac{1}{3}}{n}\right)$$

or $$\sqrt[p]{\frac{L_h}{500}} = \sqrt[p]{\frac{33\tfrac{1}{3}}{n}} \cdot \frac{C}{P}$$

where

$$f_L = \sqrt[p]{\frac{L_h}{500}} \text{ index of dynamic stressing,}$$

i.e. $f_L = 1$ for a life of 500 hours,

$$f_n = \sqrt[p]{\frac{33\,^1\!/_3}{n}} \text{ speed factor,}$$

i.e. $f_n = 1$ for a speed of 33 $^1/_3$ min^{-1}.

See page 34 for f_n values for ball bearings and page 35 for roller bearings.

The life equation is therefore given the simplified form:

$$f_L = \frac{C}{P} \cdot f_n$$

where

f_L	index of dynamic stressing	
C	dynamic load rating	[kN]
P	equivalent dynamic load	[kN]
f_n	speed factor	

Index of dynamic stressing f_L

The f_L value is an empirical value obtained from field-proven identical or similar bearing mountings. The f_L values help to select the right bearing size. The tables on pages 36 to 39 list the f_L values to be aimed at for various bearing applications. In addition to an adequate fatigue life, the f_L values take into account other requirements such as rigidity, low weight for lightweight constructions, adaptation to given mating parts, higher-than-usual peak loads, etc. (see also FAG publications on special applications). The f_L values conform with the latest standards resulting from technical progress.

For comparison with a field-proven bearing mounting the calculation of stressing must, of course, be based on the same former method. The usual data for the calculation are listed in the tables as well as the f_L values. Where supplementary factors are required, the pertinent f_z values are indicated. $f_z \cdot P$ is used for the calculation instead of P. The nominal rating life L_h is assessed with the help of the f_L value.

To change f_L to L_h see table on page 34 for ball bearings and on page 35 for roller bearings.

With the f_L and L_h values dimensioning parameters are obtained only for those cases in which a comparison with field-proven bearings is possible. For a more precise assessment of the attainable life also the effects of lubrication, temperature, and cleanliness must be taken into account (see page 40 et seq.).

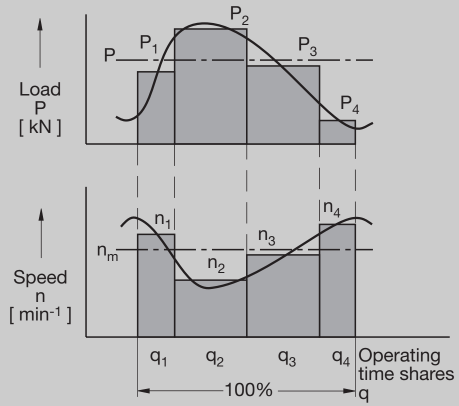
Variable load and speed

If the load and speed for dynamically stressed bearings change in time, corresponding consideration must be given when calculating the equivalent load. The curve is approximated by a series of individual loads and speeds of a certain duration q [%]. In this case, the equivalent dynamic load P is obtained from:

$$P = \sqrt[3]{P_1^3 \cdot \frac{n_1}{n_m} \cdot \frac{q_1}{100} + P_2^3 \cdot \frac{n_2}{n_m} \cdot \frac{q_2}{100} + \ldots} \; [\text{kN}]$$

and the mean rotational speed n_m from:

$$n_m = n_1 \cdot \frac{q_1}{100} + n_2 \cdot \frac{q_2}{100} + \ldots \; [\text{min}^{-1}]$$



Dimensioning
Dynamically stressed bearings

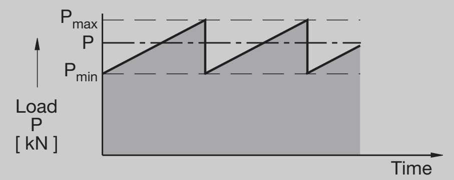
For the sake of simplicity, exponent 3 is indicated in the formulas for ball bearings and roller bearings.

If the load is variable but the speed constant:

$$P = \sqrt[3]{P_1^3 \cdot \frac{q_1}{100} + P_2^3 \cdot \frac{q_2}{100} + \ldots} \; [\text{kN}]$$

If the load grows linearly from a minimum value P_{min} to a maximum value P_{max} at a constant speed:

$$P = \frac{P_{min} + 2P_{max}}{3} \; [\text{kN}]$$



The mean value of the equivalent dynamic load may not be used for the adjusted life calculation (see page 40). The general loading of a bearing consists of various load types. The times during which the same load type acts on a bearing must be summed up and the individual subsums entered in the L_{hna} calculation. The attainable life can then be calculated with the formula on page 49.

Minimum rolling bearing load, avoiding overdimensioning

At too low loading - e.g. at high speeds during the test run - slippage may occur and lead to bearing damage if lubrication is inadequate. We recommend the following minimum loads for radial bearings:

For caged ball bearings:	P/C = 0.01,
for caged roller bearings:	P/C = 0.02,
for full-complement bearings:	P/C = 0.04

(P is the equivalent dynamic load, C is the dynamic load rating).

The minimum loads for axial bearings can be taken from the introduction prior to the tables.

Please consult our Technical Service if you have questions on the minimum rolling bearing load.

Overdimensioning of bearings may lead to a shorter service life: Overdimensioned bearings are exposed to the risk of slippage and increased lubricant stressing with for-life grease lubrication. Slippage may destroy the functional surfaces by smearing or micro pitting. In order to obtain a cost-effective and operationally reliable bearing arrangement the load carrying capacity must be fully utilized. To this end, in addition to the load rating capacity further influencing parameters must be taken into account as is the case with the adjusted life calculation.

Remarks

The above calculation methods and symbols conform to the specifications in DIN ISO 76 and 281. For simplification reasons, C and C_0 are used for the dynamic and static load ratings for radial and axial bearings in formulas and tables as are P and P_0 for the equivalent dynamic load and equivalent static load respectively. The standard makes a differentiation between:

C_r	dynamic radial load rating
C_a	dynamic axial load rating
C_{0r}	static radial load rating
C_{0a}	static axial load rating
P_r	equivalent dynamic radial load
P_a	equivalent dynamic axial load
P_{0r}	equivalent static radial load
P_{0a}	equivalent static axial load

For reasons of simplification, the indices r and a are not used with C and P in this catalogue. The relation of the load ratings and equivalent loads to radial and axial bearings is unequivocal in practice.

DIN ISO 281 only mentions the rating life L_{10} and the adjusted rating life L_{na} in 10^6 revolutions. The life values L_h and L_{hna} expressed in hours can thus be calculated (see also pages 31 and 40). In practice, L_h and L_{hna} and, especially the index of dynamic stressing, f_L, are commonly used. Therefore, recommended values for the index of dynamic stressing f_L and life formulas L_h and L_{hna} are included in this catalogue as a supplement to the standard.

Dimensioning

Rating life L_h and speed factor f_n for ball bearings

▼ f_L values for ball bearings

$$f_L = \sqrt[3]{\frac{L_h}{500}}$$

L_h	f_L	L_h	f_L	L_h	f_L	L_h	f_L	L_h	f_L
h		h		h		h		h	
100	0.585	420	0.944	1700	1.5	6500	2.35	28000	3.83
110	0.604	440	0.958	1800	1.53	7000	2.41	30000	3.91
120	0.621	460	0.973	1900	1.56	7500	2.47	32000	4
130	0.638	480	0.986	2000	1.59	8000	2.52	34000	4.08
140	0.654	500	1	2200	1.64	8500	2.57	36000	4.16
150	0.669	550	1.03	2400	1.69	9000	2.62	38000	4.24
160	0.684	600	1.06	2600	1.73	9500	2.67	40000	4.31
170	0.698	650	1.09	2800	1.78	10000	2.71	42000	4.38
180	0.711	700	1.12	3000	1.82	11000	2.8	44000	4.45
190	0.724	750	1.14	3200	1.86	12000	2.88	46000	4.51
200	0.737	800	1.17	3400	1.89	13000	2.96	48000	4.58
220	0.761	850	1.19	3600	1.93	14000	3.04	50000	4.64
240	0.783	900	1.22	3800	1.97	15000	3.11	55000	4.79
260	0.804	950	1.24	4000	2	16000	3.17	60000	4.93
280	0.824	1000	1.26	4200	2.03	17000	3.24	65000	5.07
300	0.843	1100	1.3	4400	2.06	18000	3.3	70000	5.19
320	0.862	1200	1.34	4600	2.1	19000	3.36	75000	5.31
340	0.879	1300	1.38	4800	2.13	20000	3.42	80000	5.43
360	0.896	1400	1.41	5000	2.15	22000	3.53	85000	5.54
380	0.913	1500	1.44	5500	2.22	24000	3.63	90000	5.65
400	0.928	1600	1.47	6000	2.29	26000	3.73	100000	5.85

▼ f_n values for ball bearings

$$f_n = \sqrt[3]{\frac{33\frac{1}{3}}{n}}$$

n	f_n	n	f_n	n	f_n	n	f_n	n	f_n
min^{-1}		min^{-1}		min^{-1}		min^{-1}		min^{-1}	
10	1.49	55	0.846	340	0.461	1800	0.265	9500	0.152
11	1.45	60	0.822	360	0.452	1900	0.26	10000	0.149
12	1.41	65	0.8	380	0.444	2000	0.255	11000	0.145
13	1.37	70	0.781	400	0.437	2200	0.247	12000	0.141
14	1.34	75	0.763	420	0.43	2400	0.24	13000	0.137
15	1.3	80	0.747	440	0.423	2600	0.234	14000	0.134
16	1.28	85	0.732	460	0.417	2800	0.228	15000	0.131
17	1.25	90	0.718	480	0.411	3000	0.223	16000	0.128
18	1.23	95	0.705	500	0.405	3200	0.218	17000	0.125
19	1.21	100	0.693	550	0.393	3400	0.214	18000	0.123
20	1.19	110	0.672	600	0.382	3600	0.21	19000	0.121
22	1.15	120	0.652	650	0.372	3800	0.206	20000	0.119
24	1.12	130	0.635	700	0.362	4000	0.203	22000	0.115
26	1.09	140	0.62	750	0.354	4200	0.199	24000	0.112
28	1.06	150	0.606	800	0.347	4400	0.196	26000	0.109
30	1.04	160	0.593	850	0.34	4600	0.194	28000	0.106
32	1.01	170	0.581	900	0.333	4800	0.191	30000	0.104
34	0.993	180	0.57	950	0.327	5000	0.188	32000	0.101
36	0.975	190	0.56	1000	0.322	5500	0.182	34000	0.0993
38	0.957	200	0.55	1100	0.312	6000	0.177	36000	0.0975
40	0.941	220	0.533	1200	0.303	6500	0.172	38000	0.0957
42	0.926	240	0.518	1300	0.295	7000	0.168	40000	0.0941
44	0.912	260	0.504	1400	0.288	7500	0.164	42000	0.0926
46	0.898	280	0.492	1500	0.281	8000	0.161	44000	0.0912
48	0.886	300	0.481	1600	0.275	8500	0.158	46000	0.0898
50	0.874	320	0.471	1700	0.27	9000	0.155	50000	0.0874

Dimensioning

Rating life L_h and speed factor f_n for roller bearings

▼ f_L values for roller bearings

$$f_L = \sqrt[\frac{10}{3}]{\frac{L_h}{500}}$$

L_h	f_L	L_h	f_L	L_h	f_L	L_h	f_L	L_h	f_L
h		h		h		h		h	
100	0.617	420	0.949	1700	1.44	6500	2.16	28000	3.35
110	0.635	440	0.962	1800	1.47	7000	2.21	30000	3.42
120	0.652	460	0.975	1900	1.49	7500	2.25	32000	3.48
130	0.668	480	0.988	2000	1.52	8000	2.3	34000	3.55
140	0.683	500	1	2200	1.56	8500	2.34	36000	3.61
150	0.697	550	1.03	2400	1.6	9000	2.38	38000	3.67
160	0.71	600	1.06	2600	1.64	9500	2.42	40000	3.72
170	0.724	650	1.08	2800	1.68	10000	2.46	42000	3.78
180	0.736	700	1.11	3000	1.71	11000	2.53	44000	3.83
190	0.748	750	1.13	3200	1.75	12000	2.59	46000	3.88
200	0.76	800	1.15	3400	1.78	13000	2.66	48000	3.93
220	0.782	850	1.17	3600	1.81	14000	2.72	50000	3.98
240	0.802	900	1.19	3800	1.84	15000	2.77	55000	4.1
260	0.822	950	1.21	4000	1.87	16000	2.83	60000	4.2
280	0.84	1000	1.23	4200	1.89	17000	2.88	65000	4.31
300	0.858	1100	1.27	4400	1.92	18000	2.93	70000	4.4
320	0.875	1200	1.3	4600	1.95	19000	2.98	80000	4.58
340	0.891	1300	1.33	4800	1.97	20000	3.02	90000	4.75
360	0.906	1400	1.36	5000	2	22000	3.11	100000	4.9
380	0.921	1500	1.39	5500	2.05	24000	3.19	150000	5.54
400	0.935	1600	1.42	6000	2.11	26000	3.27	200000	6.03

▼ f_n values for roller bearings

$$f_n = \sqrt[\frac{10}{3}]{\frac{33\frac{1}{3}}{n}}$$

n	f_n	n	f_n	n	f_n	n	f_n	n	f_n
min^{-1}		min^{-1}		min^{-1}		min^{-1}		min^{-1}	
10	1.44	55	0.861	340	0.498	1800	0.302	9500	0.183
11	1.39	60	0.838	360	0.49	1900	0.297	10000	0.181
12	1.36	65	0.818	380	0.482	2000	0.293	11000	0.176
13	1.33	70	0.8	400	0.475	2200	0.285	12000	0.171
14	1.3	75	0.784	420	0.468	2400	0.277	13000	0.167
15	1.27	80	0.769	440	0.461	2600	0.270	14000	0.163
16	1.25	85	0.755	460	0.455	2800	0.265	15000	0.16
17	1.22	90	0.742	480	0.449	3000	0.259	16000	0.157
18	1.2	95	0.73	500	0.444	3200	0.254	17000	0.154
19	1.18	100	0.719	550	0.431	3400	0.25	18000	0.151
20	1.17	110	0.699	600	0.42	3600	0.245	19000	0.149
22	1.13	120	0.681	650	0.41	3800	0.242	20000	0.147
24	1.1	130	0.665	700	0.401	4000	0.238	22000	0.143
26	1.08	140	0.65	750	0.393	4200	0.234	24000	0.139
28	1.05	150	0.637	800	0.385	4400	0.231	26000	0.136
30	1.03	160	0.625	850	0.378	4600	0.228	28000	0.133
32	1.01	170	0.613	900	0.372	4800	0.225	30000	0.13
34	0.994	180	0.603	950	0.366	5000	0.222	32000	0.127
36	0.977	190	0.593	1000	0.36	5500	0.216	34000	0.125
38	0.961	200	0.584	1100	0.35	6000	0.211	36000	0.123
40	0.947	220	0.568	1200	0.341	6500	0.206	38000	0.121
42	0.933	240	0.553	1300	0.333	7000	0.201	40000	0.119
44	0.92	260	0.54	1400	0.326	7500	0.197	42000	0.117
46	0.908	280	0.528	1500	0.319	8000	0.193	44000	0.116
48	0.896	300	0.517	1600	0.313	8500	0.19	46000	0.114
50	0.885	320	0.507	1700	0.307	9000	0.186	50000	0.111

Dimensioning

Recommended f_L values and general stress conditions

Application	Index of dynamic stressing f_L	Stress conditions
Power-driven vehicles		**Drive**
motor cycles	0.9 ... 1.6	Max. engine torque and corresponding rotational speed taking into consideration the transmissible torque. Mean f_L value from f_{L1}, f_{L2}, f_{L3}...of the speed gears and the corresponding time shares q_1, q_2, q_3 ... (%)
cars:drive	1 .... 1.3	
dirt-protected bearings (gearboxes)	0.7 ... 1	
cars: wheel bearings	1.4 ... 2.2	
light trucks	1.6 ... 2	
medium trucks	1.8 ... 2.2	
heavy trucks	2 ... 2.6	
busses	1.8 ... 2.8	

$$f_L = \sqrt[3]{\frac{100}{\frac{q_1}{f_{L1}^3}+\frac{q_2}{f_{L2}^3}+\frac{q_3}{f_{L3}^3}+\dots}}$$

Wheel bearings, example of collective driving loads

Static axle load K_{stat} at mean speed
Mean f_L value (see above) from three driving conditions: driving straight, good road with static load K_{stat}
driving straight, bad road with $K_{stat} \cdot f_z$
driving in bends with $K_{stat} \cdot f_z \cdot m$

Vehicle type	Supplementary factor f_z
car, bus, motor cycle	1.3
station wagon, truck, towing vehicle	1.5
cross-country truck, agricultural tractor	1.5 ... 1. 7

m is the coefficient of road grip

wheel type	m
steerable wheels	0.6
non-steerable wheels	0.35

Application	Index of dynamic stressing f_L	Stress conditions
internal combustion engines	1.2 ... 2	maximum forces (gas pressure, inertia forces) at top dead centre and at full load with f_z; maximum rotational speed

Factor f_z:

process	Otto engine	Diesel engine
two-stroke	0.35	0.5
four-stroke	0.3	0.4

Application	Index of dynamic stressing f_L	Stress conditions
Rail vehicles		
axle box roller bearings for		static axle load with factor f_z (depending on top speed, vehicle type and superstructure of the track)
haulage cars	2.5 ... 3.5	
trams	3.5 ... 4	
passenger coaches	3 ... 3.5	
goods wagons	3 ... 3.5	
mineral wagons	3 ... 3.5	
rail cars	3.5 ... 4	
locomotives/outer bearings	3.5 ... 4	
locomotives/inner bearings	4.5 ... 5	

vehicle type	f_z
mineral wagons, haulage cars, steel works vehicles	1.2 ... 1.4
goods wagons, passenger coaches, rail cars, trams	1.2 ... 1.5
locomotives	1.3 ... 1.8

Application	Index of dynamic stressing f_L	Stress conditions
transmission gears for rail vehicles	3 ... 4.5	collective loads with corresponding mean speeds; mean f_L values (see motor vehicle drives)

Dimensioning

Recommended f_L values and general stress conditions

Application	Index of dynamic stressing f_L	Stress conditions
Shipbuilding		
ship's propeller thrust blocks	3 ... 4	max. propeller thrust; nominal propeller speed
ship's propeller shaft bearings	4 ... 6	proportional shaft weight; nominal rotational speed; f_z = 2
large marine gears	2.5 ... 3.7	nominal power; nominal speed
small marine gears	2 ... 3	nominal power; nominal speed
propulsion units	1.5 ... 2.5	nominal power; nominal speed
		Rudder bearings
		statically loaded by rudder pressure, weight, drive power
Agricultural machinery		
agricultural tractors self-propelled	1.5 ... 2	same as motor vehicles
cultivating machines	1.5 ... 2	same as motor vehicles
seasonal machines	1 ... 1.5	maximum output; nominal speed
Construction machinery		
crawler tractors, loaders	2 ... 2.5	same as motor vehicles
excavators/travelling gears	1 ... 1.5	mean torque of the hydrostatic drive;
excavators/slewing gears	1.5 ... 2	mean rotational speed
vibrating road rollers,		
vibrators	1.5 ... 2.5	centrifugal force · f_z (supplementary factor f_z = 1.1 ... 1.3)
vibrating pokers	1 ... 1.5	
Electric motors		
electric motors for		
household appliances	1.5 ... 2	rotor weight · f_z · nominal speed
standard motors	3.5 ... 4.5	supplementary factor f_z = 1.5 ... 2 for stationary machinery
large motors	4 ... 5	f_z = 1.5 ... 2.5 for traction motors
traction motors	3 ... 3.5	for pinion drives: varying load conditions and their time shares
Rolling mills, metal production plants		
roll stands	1 ... 3	mean rolling load; rolling speed ($f_L$value according to roll stand and rolling programme)
rolling mill gears	3 ... 4	nominal or maximum torque; nominal speed
roller tables	2.5 ... 3.5	weight of material, shocks; rolling speed
centrifugal casting machines	3.5 ... 4.5	weight, imbalance; nominal speed
		BOF applications
		statically loaded by maximum weight
Machine tools		
lathe spindles, milling spindles	3 ... 4.5	cutting power, driving power,
boring spindles	3 ... 4	preload, workpiece weight;
grinding spindles	2.5 ... 3.5	operating speed
headstock spindles of grinding machines	3.5 ... 5	
machine tool gears	3 ... 4	nominal power; nominal speed
presses/flywheel	3.5 ... 4	flywheel weight; nominal speed
presses/eccentric shaft	3 ... 3.5	press load, corresponding time share, nominal speed
electric tools and pneumatic tools	2 ... 3	cutting and driving power; nominal speed

Dimensioning
Recommended f_L values and general stress conditions

Application	Index of dynamic stressing f_L	Stress conditions
Woodworking machines		
milling cutters and cutter shafts	3 ... 4	cutting and driving power; nominal speed
frame saws/main bearings	3.5 ... 4	inertia forces; nominal speed
frame saws/connecting rod bearings	2.5 ... 3	inertia forces; nominal speed
circular saws	2 ... 3	cutting and driving power, nominal speed
Gears for machinery construction		
universal gears	2 ... 3	nominal power; nominal speed
gear motors	2 ... 3	nominal power; nominal speed
large-size gears, stationary	3 ... 4.5	nominal power; nominal speed
Materials handling		
belt drives/open-cast mining	4.5 ... 5.5	nominal power; nominal speed
belt conveyor idlers/ open-cast mining	4.5 ... 5	weight of belt and material conveyed; operating speed
belt conveyor idlers/general	2.5 ... 3.5	weight of belt and material conveyed; operating speed
belt pulleys	4 ... 4.5	belt pull, weight of belt and material conveyed; operating speed
bucket wheel excavators/drive	2.5 ... 3.5	nominal power; nominal speed
bucket wheel excavators/ bucket wheel	4.5 ... 6	digging pressure, weight; operating speed
bucket wheel excavators/ bucket wheel drive	4.5 ... 5.5	nominal power; nominal speed
winding cable sheaves	4 ... 4.5	load on cable; nominal speed (DIN 22 410)
rope pulleys	2.5 ... 3.5	load on rope; nominal speed
Pumps, blowers, compressors		
ventilating fans	3.5 ... 4.5	axial or radial load, rotor weight, imbalance
high-capacity blowers	4 ... 5	imbalance = rotor weight · f_z; nominal speed supplementary factor f_z = 0.5 for fresh-air blowers f_z = 0.8 ... 1 for exhaustors
piston pumps	3.5 ... 4.5	nominal pressure, nominal speed
centrifugal pumps	3 ... 4.5	axial load, rotor weight; nominal speed
hydraulic axial piston pumps and hydraulic radial piston pumps	1 ... 2.5	nominal pressure, nominal speed
gear pumps	1 ... 2.5	operating pressure, nominal speed
compressors	2 ... 3.5	operating pressure, inertia forces, nominal speed
Centrifuges, stirrers		
centrifuges	2.5 ... 3	weight, imbalance; nominal speed
large stirrers	3.5 ... 4	weight, driving force; nominal speed
Crushers, mills, screens, etc.		
jaw crushers	3 ... 3.5	drive power, radius of eccentricity; nominal speed
cone crushers, roll crushers	3 ... 3.5	crushing force; nominal speed
beater mills, hammer mills, impact mills	4 ... 5	rotor weight · f_z ; nominal speed; f_z = 2 ... 2.5
tube mills	4 ... 5	total weight · f_z; nominal speed; f_z = 1.5 ... 2.5
vibrating mills	2 ... 3	centrifugal force · f_z; nominal speed; f_z = 1.2 ... 1.3
pulverising mills	4 ... 5	contact load · f_z; nominal speed; f_z = 1.5 ... 3
vibrating screens	2.5 ... 3	centrifugal force · f_z; nominal speed; f_z = 1.2
briquette presses	3.5 ... 4	pressure; nominal speed
rotary kiln support rollers	4 ... 5	roller load · f_z; nominal speed; factor for eccentric loading f_z = 1.2 ... 1.3; at higher load check static load carrying capacity

Dimensioning
Recommended f_L values and general stress conditions

Application	Index of dynamic stressing f_L	Stress conditions
Paper machines, printing machines		
paper machines/ wet section	5 ... 5.5	screen pull, felt draw, roll or cylinder weight, contact pressure; nominal speed
paper machines/ dryer section	5.5 ... 6.5	
paper machines/ refiners	5 ... 5.5	
paper machines/ calenders	4.5 ... 5	
printing machines	4 ... 4.5	roll or cylinder weight, contact pressure; nominal speed
Textile machinery		
spinning machines/ spindles	3.5 ... 4.5	imbalance loads; nominal speed
power looms, knitting and hosiery machines	3 ... 4	drive power, imbalance load, inertia forces; nominal speed
Plastics processing machinery		
screw extruders for plastic materials	3 ... 3.5	maximum injection pressure; operating speed; with injection moulding machines check static load carrying capacity
rubber and plastics sheeting calenders	3.5 ... 4.5	mean rolling load; mean speed; (temperature)
Belt and rope drives		circumferential force · f_z (due to preload and shock loads)
chain drives		f_z = 1.5
V-belts		f_z = 2 ... 2.5
fabric belts		f_z = 2 ... 3
leather belts		f_z = 2.5 ... 3.5
steel bands		f_z = 3 ... 4
toothed belts		f_z = 1.5 ... 2

Adjusted rating life calculation

The nominal life L or L_h deviates more or less from the really attainable life of rolling bearings. The equation $L = (C/P)^p$ considers only the load out of the scope of operating conditions. The really attainable life, however, depends on a variety of other influences, e.g. the lubricant film thickness, the cleanliness in the lubricating gap, the lubricant additives, and the bearing type.

Therefore, the standard DIN ISO 281 introduced the "modified life" in addition to the nominal life, but it did so far not give figures for the factor which takes the operating conditions into account. With the FAG calculation process for the attainable life, however, operating conditions can be expressed in terms of figures by the factor a_{23}. The stress index f_{s*} is also considered as a criterion for dimensioning. It is a measure of the maximum compressive stresses in the areas of rolling contact.

Attainable (modified) life

The attainable (modified) life L_{na} is calculated with the following formula according to DIN ISO 281:

$$L_{na} = a_1 \cdot a_2 \cdot a_3 \cdot L \; [10^6 \text{ revolutions}]$$

or when expressed in hours

$$L_{hna} = a_1 \cdot a_2 \cdot a_3 \cdot L_h \; [h]$$

where

L_{na} attainable (modified) life [10^6 revolutions]
L_{hna} attainable life [h]
a_1 factor for failure probability
a_2 factor for material
a_3 factor for operating conditions
L, L_h nominal rating life [10^6 revolutions], [h]

Life adjustment factor a_1 for failure probability

Rolling bearing failures due to fatigue are subject to statistical laws, which is why the failure probability must be taken into account when calculating the fatigue life. Generally 10% failure probability is taken. The L_{10} life is the nominal rating life. The factor a_1 is also used for failure probabilities between 10 % and 1%, see the following table.

▼ Factor a_1

Failure probability %	10	5	4	3	2	1
Fatigue life	L_{10}	L_5	L_4	L_3	L_2	L_1
Factor a_1	1	0.62	0.53	0.44	0.33	0.21

Life adjustment factor a_2 for material

Factor a_2 takes into consideration the characteristics of the material and its heat treatment. The standard permits factors $a_2 > 1$ for bearings of particularly clean steel.

Life adjustment factor a_3 for operating conditions

Factor a_3 takes into consideration the operating conditions, especially the lubrication condition under operating speed and operating temperature. The standard does not yet include figures for this factor.

FAG method of calculating the adjusted life

Diverse and systematic laboratory investigations and the feedback from practical experience, allow us today to quantify the effect of various operating conditions on the attainable life of rolling bearings.

The method of calculating the attainable life is based on DIN ISO 281. It takes into the account the effects of the magnitude of load, lubricating film thickness, lubricant doping, contaminants in the lubricating gap, and the bearing type.

Should life-influencing parameters change during the operating time, the L_{hna} value must be calculated for each individual period under constant conditions. The attainable life can then be calculated with the formula on page 49.

This calculation method also shows that rolling bearings are fail-safe under the following conditions:

- utmost cleanliness in the lubricating gap corresponding to V = 0.3 (see page 46)
- full separation of the surfaces in rolling contact by the lubricating film
- load corresponding to $f_{s*} \geqq 8$

  $f_{s*} = C_0/P_{0*}$

  C_0 static load rating [kN]
  P_{0*} equivalent bearing load [kN]

  determined by the formula

  $P_{0*} = X_0 \cdot F_r + Y_0 \cdot F_a$ [kN]

  where X_0 and Y_0 are factors from the bearing tables

  and

  F_r dynamic radial force [kN]
  F_a dynamic axial force [kN]

  With stress index f_{s*} a connection is established between the bearing stressing and equivalent stresses usually employed for dimensioning in General Mechanical Engineering.

Attainable life L_{na}, L_{hna}

$$L_{na} = a_1 \cdot a_{23} \cdot L \; [10^6 \text{ revolutions}]$$
and
$$L_{hna} = a_1 \cdot a_{23}. L_h \; [h]$$

where

a_1 factor for failure probability (see page 40)
a_{23} factor for material and operating conditions. Due to their interdependence FAG combined the factors a_2 and a_3 indicated in DIN ISO 281 in the factor a_{23},

$a_{23} = a_2 \cdot a_3$

L nominal life [10^6 revolutions]
L_h nominal life [h]

Factor a_{23}

The a_{23} factor for determining the attainable life L_{na} or L_{hna} (see preceding section) is obtained from the formula

$a_{23} = a_{23II} \cdot s$

where

a_{23II} basic value (diagram on page 45)
s cleanliness factor (diagrams on page 47)

The factor a_{23} takes into account effects of material, bearing type, load, lubrication and cleanliness, see graph on page 42.

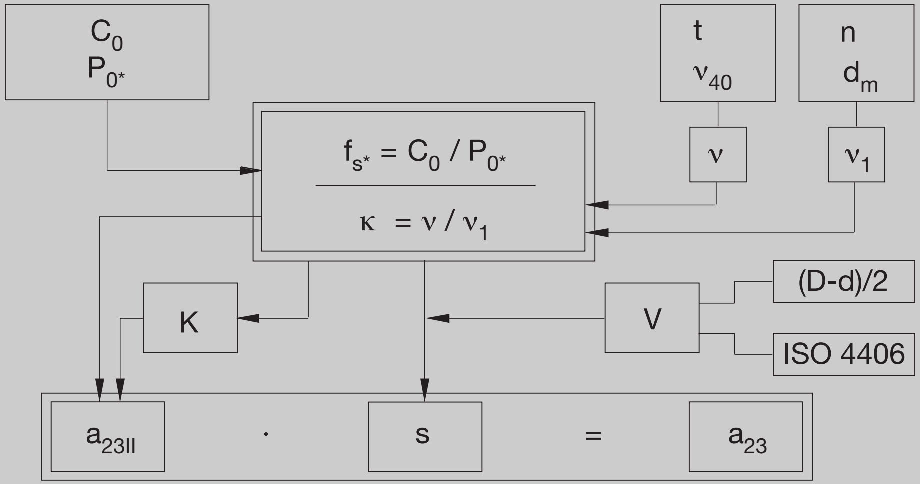
The diagram on page 45 is the basis for the determination of the a_{23} factor. Zone II of the diagram, which is the most important zone in practical operation applies to good cleanliness standards (basic value a_{23II} for s = 1).

At higher or lower cleanliness standards, s > 1 or s < 1.

▼ Graph for determining a_{23}

C_0 static load rating
P_{0*} equivalent load (page 41)
f_{s*} stress index (page 41)
K $= K_1 + K_2$ (diagrams on page 44)
a_{23II} basic value (diagram on page 45)
s cleanliness factor (diagrams on page 47)

t operating temperature
ν_{40} nominal viscosity
ν operating viscosity (lower diagram on page 43)
n operating speed
d_m mean diameter
ν_1 rated viscosity (upper diagram on page 43)
κ viscosity ratio
V contamination factor (table on page 46)



Viscosity ratio κ

The viscosity ratio κ as the measure of the lubricating film formation is shown on the abscissa of the diagram on page 45.

$\kappa = \nu/\nu_1$

ν operating viscosity of the lubricant in the rolling contact area

ν_1 rated viscosity depending on diameter and speed

The **rated viscosity** ν_1 is determined from the upper diagram on page 43 with the help of the mean diameter (D + d)/2 and the operating speed n.

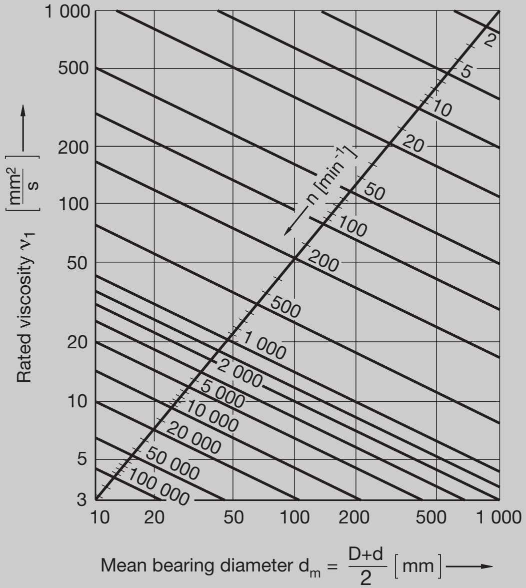
The **operating viscosity** ν of a lubricating oil is obtained from the viscosity-temperature (V-T) diagram (lower diagram on page 43) as a function of the operating temperature t and the (nominal) viscosity of the oil at 40 °C.

In the case of lubricating greases ν is the operating viscosity of the base oil.

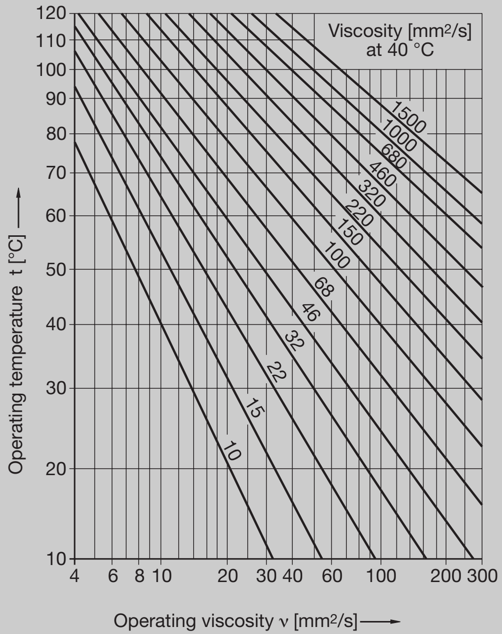
Recommendations on oil viscosity and oil selection are given on page 131.

In heavily loaded bearings with a high percentage of sliding ($f_{s*} < 4$), the temperature in the contact area of the rolling elements is up to 20 K higher than the temperature measurable at the stationary ring (without the effect of external heat). The difference can be approached by using half the operating viscosity ν read off the V-T diagram for the formula $\kappa = \nu/\nu_1$.

▼ Rated viscosity ν_1



▼ V-T diagram for mineral oils



Basic a_{23II} factor

The **value** $K = K_1 + K_2$ is required for locating the basic a_{23II} factor in the diagram on page 45.

K_1 can be read off the upper diagram on this page as a function of the bearing type and the stress index f_{s*}.

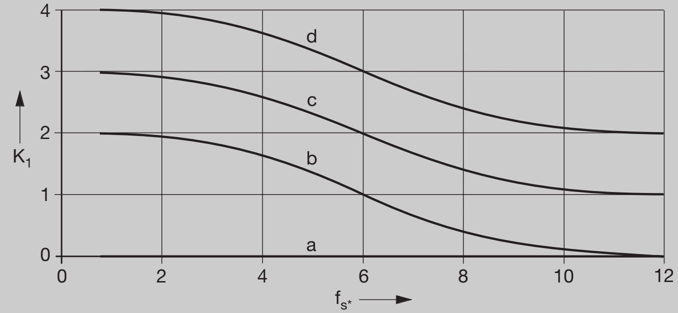
K_2 depends on the viscosity ratio κ and the index f_{s*}. The values in the lower diagram on this page apply to lubricants without additives or lubricants with additives whose special effect in rolling bearings was not tested. K_2 equals 0 for lubricants with additives with a corresponding suitability proof.

With K = 0 to 6, a_{23II} is found on one of the curves in zone II of the diagram on page 45.

With K > 6, a_{23II} must be expected to be in zone III. In such a case a smaller K value and thus zone II should be aimed at by improving the conditions.

If adequate quantities of an appropriate grease are used for lubrication, the same K_2 values can be assumed as for a suitably doped oil. The selection of the right grease is very important for bearings with a higher sliding motion share and for large, heavily stressed bearings. If the suitability of a lubricating grease is not exactly known, an a_{23II} factor from the lower limit of zone II should be chosen to be on the safe side. This is specially recommended in cases where the given lubricating interval cannot be maintained.

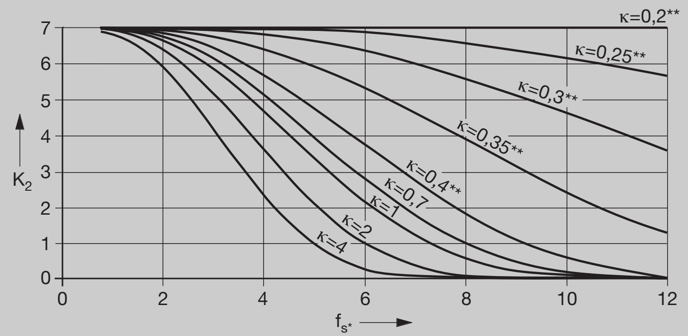
▼ Value K_1 depending on the index f_{s*} and the bearing type



a ball bearings
b tapered roller bearings
cylindrical roller bearings
c spherical roller bearings
spherical roller thrust bearings [3]
cylindrical roller thrust bearings [1), 3)]
d full complement cylindrical roller bearings [1), 2)]

1) Attainable only with lubricant filtering corresponding to V < 1, otherwise $K_1 \geq 6$ must be assumed.
2) To be observed for the determination of ν: the friction is at least twice the value in caged bearings. This results in higher bearing temperature.
3) Minimum load must be observed (page 500)

▼ Value K_2 depending on the index f_{s*} for lubricants without additives and lubricants with additives whose effect in rolling bearings was not tested

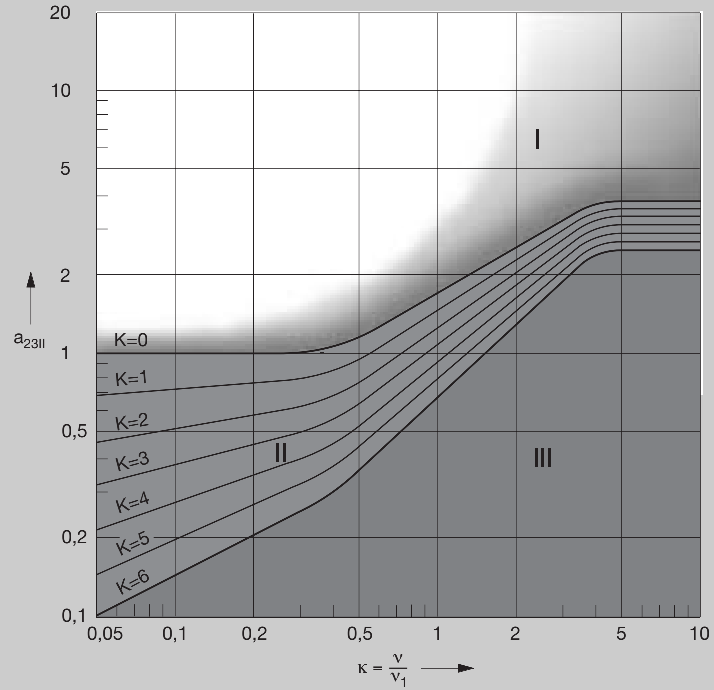


K_2 equals 0 for lubricants with additives with a corresponding suitability proof.

** With $\kappa \leqq 0.4$ wear dominates unless eliminated by suitable additives.

▼ Basic a_{23II} factor for determining the a_{23} factor

$\kappa = \nu/\nu_1$	viscosity ratio
ν	operating viscosity of lubricant, see page 42
ν_1	rated viscosity, see page 42
$K = K_1 + K_2$	values for determining the basic a_{23II} factor, see page 44



Zones

I: Transition to the endurance strength
Precondition: Utmost cleanliness in the lubricating gap and loads which are not too high, suitable lubricant

II: Normal degree of cleanliness in the lubricating gap (with effective additives tested in rolling bearings, a_{23} factors > 1 are possible even with κ < 0.4)

III: Unfavourable lubricating conditions
Heavily contaminated lubricant
Unsuitable lubricants

Limits of the life calculation

As in the case of the former life calculation, only material fatigue is taken into consideration as a cause of failure for the adjusted life calculation as well. The calculated life can only correspond to the actual service life of the bearing when the lubricant service life or the life limited by wear is not shorter than the fatigue life.

Cleanliness factor s

Factor s quantifies the effect of contamination on the life. Contamination factor V (see table below) is required to obtain s.

s = 1 always applies to **normal cleanliness (V = 1)**, i.e. $a_{23II} = a_{23}$.

With **improved cleanliness (V = 0.5)** and **utmost cleanliness (V = 0.3)** a cleanliness factor s ≧ 1 is obtained from the right diagram (a) of page 47, based on the index f_{s^*} (see page 41) and depending on the viscosity ratio κ.

s = 1 applies to κ ≦ 0.4.

With **V = 2 (moderately contaminated lubricant)** and **V = 3 (heavily contaminated lubricant)**, s < 1 is obtained from diagram b on page 47. The effect of a reduction of the factor s due to high V values is the greater the lower is the load acting on a bearing.

Contamination factor V for quantifying the cleanliness

Contamination factor V depends on the bearing cross section, the type of contact between the mating surfaces, and the cleanliness class of the oil.

If hard particles from a defined size on are cycled in the most heavily stressed contact area of a rolling bearing, the resulting indentations in the contact surfaces lead to premature material fatigue. The smaller the contact area, the more damaging the effect of a particle of a defined size.

At the same contamination level, small bearings react, therefore, more sensitively than larger ones and bearings with point contact (ball bearings) are more vulnerable than bearings with line contact (roller bearings).

The **necessary oil cleanliness class** according to ISO 4406 is an objectively measurable level of the contamination of a lubricant. It is determined by the standardized particle-counting method. The numbers of all particles > 5 µm and all particles > 15 µm are allocated to a certain oil cleanliness class. An oil cleanliness 15/12 according to ISO 4406 means that between 16000 and 32000 particles > 5 µm and betweeen 2000 and 4000 particles > 15 µm are present per 100 ml of a fluid. The step from one class to the next is by doubling or halving the particle number.

Specially particles with a hardness > 50 HRC reduce the life of rolling bearings. These are particles of hardened steel, sand and abrasive particles. Abrasive particles are particulary harmful.

If the major part of foreign particles in the oil samples is in the life-reducing hardness range, which is the case in many technical applications, the cleanliness class determined with a particle counter can be compared directly with the values of the table on page 46. If, however, the filtered out contaminants are found, after counting, to be almost exclusively mineral matter as, for example, the particularly harmful moulding sand or abrasive grains, the measured values must be increased by one to two cleanliness classes before determining the contamination factor V. On the other hand, if the greater part of the particles found in the lubricant are soft materials such as wood, fibres or paint, the measured value of the particle counter should be reduced correspondingly.

▼ Guide values for the contamination factor V

(D-d)/2 mm	V	Point contact: required oil cleanliness class according to ISO 4406[1])	Point contact: guide values for a suitable filtration ratio according to ISO 4572	Line contact: required oil cleanliness class according to ISO 4406[1])	Line contact: guide values for a suitable filtration ratio according to ISO 4572
≦ 12.5	0.3	11/8	$\beta_3 \geqq 200$	12/9	$\beta_3 \geqq 200$
	0.5	12/9	$\beta_3 \geqq 200$	13/10	$\beta_3 \geqq 75$
	1	14/11	$\beta_6 \geqq 75$	15/12	$\beta_6 \geqq 75$
	2	15/12	$\beta_6 \geqq 75$	16/13	$\beta_{12} \geqq 75$
	3	16/13	$\beta_{12} \geqq 75$	17/14	$\beta_{25} \geqq 75$
> 12.5 ... 20	0.3	12/9	$\beta_3 \geqq 200$	13/10	$\beta_3 \geqq 75$
	0.5	13/10	$\beta_3 \geqq 75$	14/11	$\beta_6 \geqq 75$
	1	15/12	$\beta_6 \geqq 75$	16/13	$\beta_{12} \geqq 75$
	2	16/13	$\beta_{12} \geqq 75$	17/14	$\beta_{25} \geqq 75$
	3	18/14	$\beta_{25} \geqq 75$	19/15	$\beta_{25} \geqq 75$
> 20 ... 35	0.3	13/10	$\beta_3 \geqq 75$	14/11	$\beta_6 \geqq 75$
	0.5	14/11	$\beta_6 \geqq 75$	15/12	$\beta_6 \geqq 75$
	1	16/13	$\beta_{12} \geqq 75$	17/14	$\beta_{12} \geqq 75$
	2	17/14	$\beta_{25} \geqq 75$	18/15	$\beta_{25} \geqq 75$
	3	19/15	$\beta_{25} \geqq 75$	20/16	$\beta_{25} \geqq 75$
> 35	0.3	14/11	$\beta_6 \geqq 75$	14/11	$\beta_6 \geqq 75$
	0.5	15/12	$\beta_6 \geqq 75$	15/12	$\beta_{12} \geqq 75$
	1	17/14	$\beta_{12} \geqq 75$	18/14	$\beta_{25} \geqq 75$
	2	18/15	$\beta_{25} \geqq 75$	19/16	$\beta_{25} \geqq 75$
	3	20/16	$\beta_{25} \geqq 75$	21/17	$\beta_{25} \geqq 75$

The oil cleanliness class can be determined by means of oil samples by filter manufacturers and institutes. It is a measure of the probability of life-reducing particles being cycled in a bearing. Suitable sampling should be observed (see e.g. DIN 51750). Today, on-line measuring instruments are available. The cleanliness classes are reached if the entire oil volume flows through the filter within a few minutes. To ensure a high degree of cleanliness flushing is required **prior** to bearing operation.

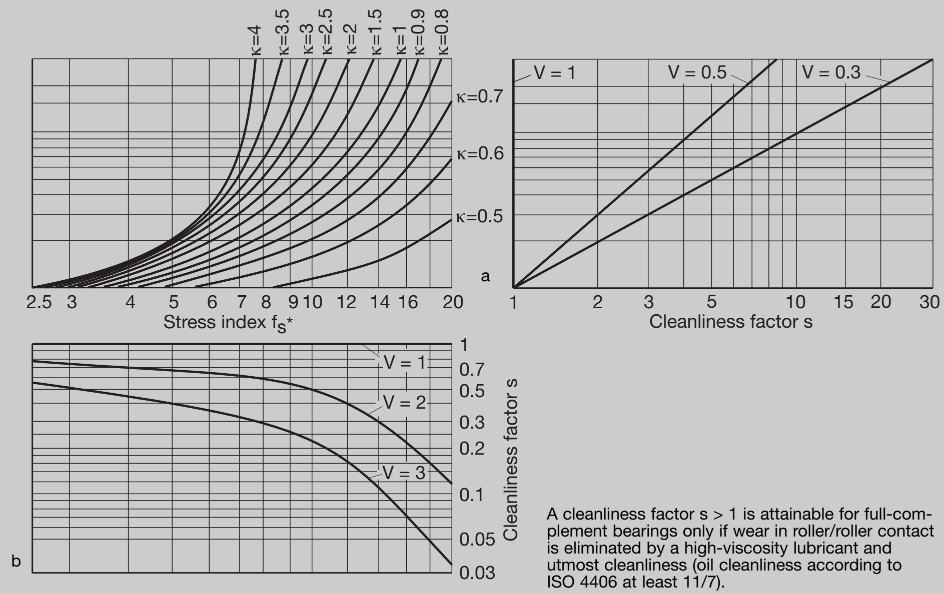
For example, filtration ratio $\beta_3 \geqq 200$ (ISO 4572) means that in the so-called multi-pass test only one of 200 particles ≧ 3 µm passes through the filter. Filters with coarser filtration ratios than $\beta_{25} \geqq 75$ should not be used due to the ill effect on the other components within the circulation system.

[1]) Only particles with a hardness > 50 HRC have to be taken into account.

▼ Oil cleanliness classes according to ISO 4406 (excerpt)

Number of particles per 100 ml over 5 µm: more than	and up to	over 15 µm: more than	and up to	Code
500000	1000000	64000	130000	20/17
250000	500000	32000	64000	19/16
130000	250000	16000	32000	18/15
64000	130000	8000	16000	17/14
32000	64000	4000	8000	16/13
16000	32000	2000	4000	15/12
8000	16000	1000	2000	14/11
4000	8000	500	1000	13/10
2000	4000	250	500	12/9
1000	2000	130	250	11/8
1000	2000	64	130	11/7
500	1000	32	64	10/6
250	500	32	64	9/6

▼ Diagram for determining the cleanliness factor s
a) Diagram for improved (V = 0.5) and utmost (V = 0.3) cleanliness
b) Diagram for moderately contaminated lubricant (V = 2) and heavily contaminated lubricant (V = 3)



A cleanliness factor s > 1 is attainable for full-complement bearings only if wear in roller/roller contact is eliminated by a high-viscosity lubricant and utmost cleanliness (oil cleanliness according to ISO 4406 at least 11/7).

A defined **filtration ratio** should exist in order to reach the oil cleanliness required. The filtration ratio is a measure of the separation capability of a filter at defined particle sizes. Filtration ratio $ß_x$ is the ratio of all particles > x µm before passing through the filter and the particles > x µm which have passed through the filter. See the graph below.

Filtration ratio $ß_3 \geqq 200$, for example, means that in the so-called multi-pass test (ISO 4572) only one of 200 particles $\geqq$ 3 µm may pass through the filter.

A filter of a certain filtration ratio is not automatically indicative of an oil cleanliness class.

Evaluation of cleanliness

According to today's knowledge the following cleanliness scale is useful (the three most important are in boldface):

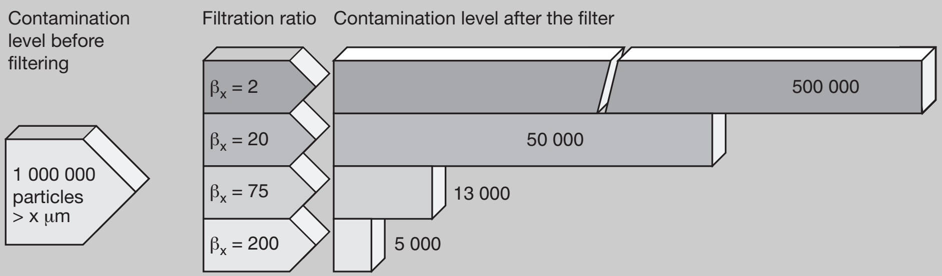
V = 0.3 utmost cleanliness
V = 0.5 improved cleanliness
V = 1 normal cleanliness
V = 2 moderately contaminated lubricant
V = 3 heavily contaminated lubricant

Utmost cleanliness

In practice, cleanliness is utmost in

– bearings which are greased and protected by seals or shields against dust by the manufacturer. The life of fail-safe types is usually limited by the service life of the lubricant.
– grease lubrication by the user who oberserves that the cleanliness level of the newly supplied bearing will be maintained throughout the entire operating time by fitting the bearing under top cleanliness conditions into a clean housing, lubricates it with clean grease and takes care that dirt cannot enter the bearing during operation.
– bearings with oil circulating systems if the circulating system is flushed prior to the first operation of the cleanly fitted bearings (fresh oil to be filled in via superfine filters) and oil cleanliness classes according to V = 0.3 are ensured during the entire operating time, see table on page 46.

▼ Filtration ratio $ß_x$



Normal cleanliness

Normal cleanliness is assumed for frequently occurring conditions:

– Good sealing adapted to the environment
– Cleanliness during mounting
– Oil cleanliness according to V = 1
– Observing the recommended oil change intervals.

Heavily contaminated lubricant

In this area a_{23} factors for dirt particles according to contamination factor V = 3 (table on page 46) may be obtained. Operating conditions should be improved!

Possible causes of heavy contamination:

– The cast housing was inadequately or not at all cleaned (foundry sand, particles from machining left in the housing).
– Abraded particles from components which are subject to wear enter the circulating oil system of the machine.
– Foreign matter penetrates into the bearing due to an unsatisfactory seal.
– Water which entered the bearing, also condensation water causes standstill corrosion or deterioration of the lubricant properties.

These conditions describe the basic parameters of the contamination factor V, and, as a rule, must be taken into account in the calculation. The intermediate values V = 0.5 (improved cleanliness) and V = 2 (moderately contaminated lubricant) must only be used if the user has the necessary experience to judge the cleanliness conditions accurately.

Worn particles also cause wear. FAG selected the heat treatment of the bearing parts in such a way that, in the case of V = 0.3, bearings with low sliding motion percentages (e.g. radial ball bearings and radial cylindrical roller bearings) show hardly any wear also during very long periods of time.

Cylindrical roller thrust bearings, full-complement cylindrical roller bearings and other bearings with high sliding motion shares react strongly to small hard contaminants. In such cases, superfine filtration of the lubricant can prevent critical wear.

Attainable life under changeable operating conditions

Should life-influencing paramaters (e.g. load, speed, temperature, cleanliness, type and quality of lubricant) change, the attainable (adjusted) life (L_{hna1}, L_{hna2}, ...) must be calculated separately for each individual period of operation q [%] under constant conditions. The attainable life is calculated for the total operating time by the formula:

$$L_{hna} = \frac{100}{\dfrac{q_1}{L_{hna1}} + \dfrac{q_2}{L_{hna2}} + \dfrac{q_3}{L_{hna3}} + \ldots}$$

Limits of the life calculation

As in the case of the former life calculation, only material fatigue is taken into consideration as a cause of failure for the adjusted life calculation as well. The calculated life can only correspond to the actual service life of the bearing when the lubricant service life or the life limited by wear is not shorter than the fatigue life.

Bearing computation at the PC

The version 1.1 of the **electronic** FAG **rolling bearing catalogue** is based on this printed catalogue. The programme on CD-ROM is even more efficient and advantageous for the user. The user is led to the best solution reliably and quickly in dialogue and saves a lot of work and time otherwise required for searching, selecting and calculating rolling bearings. Any background information can be fetched on-line in the form of texts, photos, drawings, diagrams, tables or animated pictures.

A CD-ROM will be available with which bearings can be selected for a bearing position, a shaft, and a shaft system.

Bearing data

All influences listed in the specification must be taken into consideration for the bearing arrangement. Not only the suitable bearing type and size have to be determined but also other characteristics and data on the bearing design, for example:

– Tolerances (see page 54)
– Bearing clearance (see page 74)
– Bearing material (see page 83)
– Cage design (see page 83)
– Sealing (see page 124)

Performance parameters such as suitability for high speeds (page 87) and suitability for high temperatures (page 86) are closely related to the bearing design.

Main dimensions, designation systems

Rolling bearings can be applied universally as ready-to-mount machine elements. This is especially due to the fact that the main dimensions of the popular bearings are standardized.

Dimensional plans according to ISO 15 apply to radial bearings (with the exception of tapered roller bearings and radial bearings with needle rollers), according to ISO 355 to metric tapered roller bearings and according to ISO 104 to thrust bearings. The dimensional plans of the ISO standards were taken over in DIN 616 and DIN ISO 355 (metric tapered roller bearings).

In the dimensional plans of DIN 616, each bearing bore has several outside diameters and widths. Popular diameter series are 8, 9, 0, 1, 2, 3, 4 (increasing outside diameters in this order). There are several width series in each diameter series e.g. 0, 1, 2, 3, 4 (the higher the figure the greater the width).

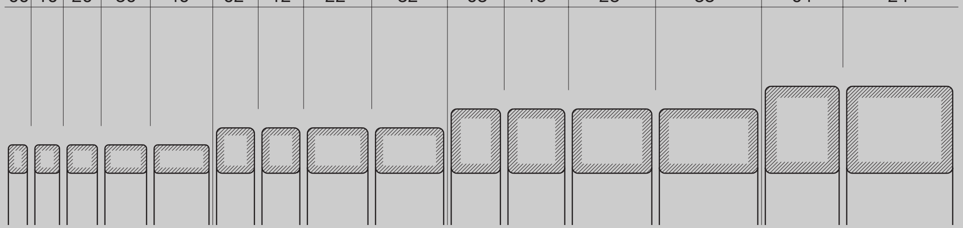
The first figure of the two-digit number for the dimension series indicates the width series (the height series for thrust bearings) and the second figure the diameter series.

The structure of the dimensional plan and the designation system for tapered roller bearings according to DIN ISO 355 differ from those according to DIN 616. In DIN ISO 355 a set figure (2, 3, 4, 5, 6) for the contact angle range is indicated. A larger figure means a larger contact angle. Two letters indicate the diameter and width series.

Deviations from the dimensional plans, e.g. for angular contact thrust ball bearings of series 2344 and 2347, are pointed out in the texts preceding the dimension tables.

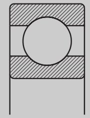
▼ Excerpt from the dimensional plan ISO 15 for radial bearings

Diameter series 0					Diameter series 2				Diameter series 3				Diameter series 4	
Width series					Width series				Width series				Width series	
0	1	2	3	4	0	1	2	3	0	1	2	3	0	2
Dimensional series					Dimensional series				Dimensional series				Dimensional series	
00	10	20	30	40	02	12	22	32	03	13	23	33	04	24

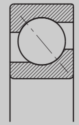


▼ Examples of basic codes for the designation of bearing series and bearing bores according to DIN 623

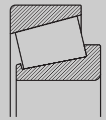
6206
Deep groove ball bearing
Width series 0
Diameter series 2
06 · 5 ≙ 30 mm bore



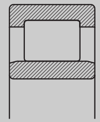
7305B
Angular contact ball bearing, single row
Width series 0
Diameter series 3
05 · 5 ≙ 25 mm bore



30209A
Tapered roller bearing
Width series 0
Diameter series 2
09 · 5 ≙ 45 mm bore



NU2314E
Cylindrical roller bearing
Lips on outer ring
Width series 2
Diameter series 3
14 · 5 ≙ 70 mm bore



▼ Designation for metric tapered roller bearings according to DIN ISO 355

Example: T 3 D B 045

T – Code letter for tapered roller bearings

3 – Contact angle range

Angle series	Contact angle range over	to
1	reserved	
2	10°	. . . 13°52’
3	13°52’	. . . 15°59’
4	15°59’	. . . 18°55’
5	18°55’	. . . 23°
6	23°	. . . 27°
7	27°	. . . 30°

D – Ratio of outside diameter to bore

Diameter series	$\frac{D}{d^{0.77}}$ over	to
A	reserved	
B	3.40	. . . 3.80
C	3.80	. . . 4.40
D	4.40	. . . 4.70
E	4.70	. . . 5.00
F	5.00	. . . 5.60
G	5.60	. . . 7.00

B – Ratio of bearing width to cross section height

Width series	$\frac{T}{(D-d)^{0.95}}$ over	to
A	reserved	
B	0.50	. . . 0.68
C	0.68	. . . 0.80
D	0.80	. . . 0.88
E	0.88	. . . 1.00

045 – Bearing bore in mm

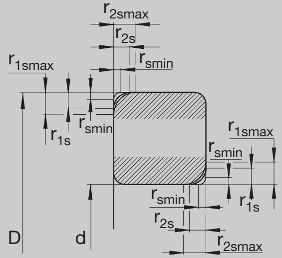
Limiting dimensions of corner

Symbols

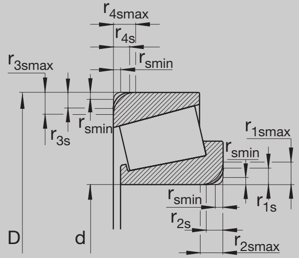
r_{1s}, r_{3s} single corner in radial direction
r_{2s}, r_{4s} single corner in axial direction

r_{smin} *) general symbol for the minimum corner r_{1smin}, r_{2smin}, r_{3smin}, r_{4smin}
r_{1smax}, r_{3smax} maximum corner in radial direction
r_{2smax}, r_{4smax} maximum corner in axial direction

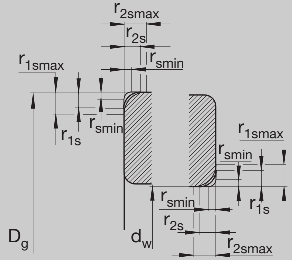
Radial bearings



Tapered roller bearings



Thrust bearings



Corner of radial bearings (except tapered roller bearings)

Dimensions in mm

r_{smin}	0.1	0.15	0.2	0.3		0.6		1		1.1		1.5	
Nominal bore diameter d over					40		40		50		120		120
to				40		40		50		120		120	
r_{1smax}	0.2	0.3	0.5	0.6	0.8	1	1.3	1.5	1.9	2	2.5	2.3	3
r_{2smax}	0.4	0.6	0.8	1	1	2	2	3	3	3.5	4	4	5

r_{smin}	2			2.1		2.5			3		4	5	6	7.5	9.5	12	15	19
Nominal bore diameter d over		80	220		280		100	280		280								
to	80	220		280		100	280		280									
r_{1smax}	3	3.5	3.8	4	4.5	3.8	4.5	5	5	5.5	6.5	8	10	12.5	15	18	21	25
r_{2smax}	4.5	5	6	6.5	7	6	6	7	8	8	9	10	13	17	19	24	30	38

Corner of tapered roller bearings
Cone

Dimensions in mm

r_{smin}	0.3		0.6		1		1.5			2		
Nominal bore diameter d over		40		40		50		120	250		120	250
to	40		40		50		120	250		120	250	
r_{1smax}	0.7	0.9	1.1	1.3	1.6	1.9	2.3	2.8	3.5	2.8	3.5	4
r_{2smax}	1.4	1.6	1.7	2	2.5	3	3	3.5	4	4	4.5	5

r_{smin}	2.5			3				4				5		6	
Nominal bore diameter d over		120	250		120	250	400		120	250	400		180		180
to	120	250		120	250	400		120	250	400		180		180	
r_{1smax}	3.5	4	4.5	4	4.5	5	5.5	5	5.5	6	6.5	6.5	7.5	7.5	9
r_{2smax}	5	5.5	6	5.5	6.5	7	7.5	7	7.5	8	8.5	8	9	10	11

Cup

Dimensions in mm

r_{smin}	0.3		0.6		1		1.5			2		
Nominal outside diameter D over		40		40		50		120	250		120	250
to	40		40		50		120	250		120	250	
r_{3smax}	0.7	0.9	1.1	1.3	1.6	1.9	2.3	2.8	3.5	2.8	3.5	4
r_{4smax}	1.4	1.6	1.7	2	2.5	3	3	3.5	4	4	4.5	5

r_{smin}	2.5			3				4				5		6	
Nominal outside diameter D over		120	250		120	250	400		120	250	400		180		180
to	120	250		120	250	400		120	250	400		180		180	
r_{3smax}	3.5	4	4.5	4	4.5	5	5.5	5	5.5	6	6.5	6.5	7.5	7.5	9
r_{4smax}	5	5.5	6	5.5	6.5	7	7.5	7	7.5	8	8.5	8	9	10	11

Corner of thrust bearings

Dimensions in mm

r_{smin}	0.1	0.15	0.2	0.3	0.6	1	1.1	1.5	2	2.1	3	4	5	6	7.5	9.5	12	15	19
r_{1smax}, r_{2smax}	0.2	0.3	0.5	0.8	1.5	2.2	2.7	3.5	4	4.5	5.5	6.5	8	10	12.5	15	18	21	25

Tapered roller bearings in inch dimensions (ISO 1123)

Cone

Dimensions in mm

Nominal bore diameter d	over / to	–/50.8	50.8/101.6	101.6/254
r_{smin} (see dimension tables)		Tolerance in mm		
r_{1smax}		r_{smin} +0.4	r_{smin} +0.5	r_{smin} +0.65
r_{2smax}		r_{smin} +0.9	r_{smin} +1.25	r_{smin} +1.8

Cup

Dimensions in mm

Nominal outside diameter D	over / to	–/101.6	101.6/168.3	168.3/266.7	266.7/355.6
r_{smin} (see dimension tables)		Tolerance in mm			
r_{3smax}		r_{smin} +0.6	r_{smin} +0.65	r_{smin} +0.85	r_{smin} +1.7
r_{4smax}		r_{smin} +1.05	r_{smin} +1.15	r_{smin} +1.35	r_{smin} +1.7

*) The lower limit value r_{smin} for the corner or chamfer according to ISO 582 and DIN 620 T6 is listed in the dimension tables.
The fillet radii at the shaft and housing shoulders are based on this value.

Tolerances

The dimensional and running tolerances of rolling bearings are stated in DIN 620. The tables (pages 56 to 73) also contain tolerance values beyond the range set in DIN 620 T2 (edition 02.88) and DIN 620 T3 (edition 06.82).

See DIN ISO 1132 for definitions of dimensions and tolerances.

Bearings of tolerance class PN (normal tolerance) generally meet the requirements for typical bearing quality in machinery construction.

Very high demands are made on the working precision, speeds, and quietness of running of machine tools, measuring instruments, etc. For such cases the standard includes the closer tolerance classes P6, P6X, P5, P4, and P2.

In addition to the standardized tolerance classes FAG also produce bearings in tolerance classes P4S, SP (super precision), and UP (ultra precision).

Tolerance symbols

DIN ISO 1132, DIN 620

Bore diameter

d Nominal bore diameter (smallest theoretical diameter for tapered bore)

d_s Single bore diameter

d_{mp} 1. Mean bore diameter; arithmetical mean of the largest and smallest single bore diameters measured in one radial plane
2. Theoretical mean small end diameter of tapered bore; arithmetical mean of largest and smallest single bore diameters

d_{1mp} Theoretical mean large end diameter of tapered bore; arithmetical mean of the largest and smallest single bore diameters

$\Delta_{dmp} = d_{mp} - d$
Deviation of mean bore diameter from the nominal dimension

$\Delta_{ds} = d_s - d$
Deviation of single bore diameter from the nominal dimension

$\Delta_{d1mp} = d_{1mp} - d_1$
Deviation of the mean large end diameter of tapered bore from nominal dimension

V_{dp} Bore diameter variation; difference between the largest and smallest single bore diameters in one radial plane

$V_{dmp} = d_{mpmax} - d_{mpmin}$
Mean bore diameter variation; difference between the largest and smallest mean bore diameters

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 radial plane

$\Delta_{Dmp} = D_{mp} - D$
Deviation of mean outside diameter from nominal dimension

$\Delta_{Ds} = D_s - D$
Deviation of a single outside diameter from nominal dimension

V_{Dp} Outside diameter variation; difference between the largest and smallest single outside diameters in one radial plane

$V_{Dmp} = D_{mpmax} - D_{mpmin}$
Mean outside diameter variation; difference between the largest and smallest mean outside diameters

Width and height

B_s, C_s Single ring width (inner and outer rings)

$\Delta_{Bs} = B_s - B$, $\Delta_{Cs} = C_s - C$
Deviation of a single ring width (inner and outer rings) from nominal dimension

$V_{Bs} = B_{smax} - B_{smin}$, $V_{Cs} = C_{smax} - C_{smin}$
Variation of inner ring and outer ring widths; difference between the largest and smallest single ring widths

T_s Single overall width of a tapered roller bearing

T_{1s} Single overall width of a tapered roller bearing with cone and master cup

T_{2s} Single overall width of a tapered roller bearing with master cone and cup

$\Delta_{Ts} = T_s - T$, $\Delta_{T1s} = T_{1s} - T_1$, $\Delta_{T2s} = T_{2s} - T_2$
Deviation of a single overall width of a tapered roller bearing from nominal dimension

*) H_s, H_{1s}, H_{2s}, H_{3s}, H_{4s}
Single overall thrust bearing height

*) $\Delta_{Hs}=H_s-H$, $\Delta_{H1s}=H_{1s}-H_1$, $\Delta_{H2s}=H_{2s}-H_2$, ...
Deviation of a single overall thrust bearing height from nominal dimension

Running accuracy

K_{ia} Radial runout of assembled bearing inner ring

K_{ea} Radial runout of assembled bearing outer ring

S_d Side face runout of inner ring with reference to bore

S_D Variation in inclination of outside cylindrical surface to outer ring side face

S_{ia} Assembled bearing inner ring face runout with raceway (axial runout)

S_{ea} Assembled bearing outer ring face runout with raceway (axial runout)

S_i Shaft washer thickness variation from raceway middle to back face (axial runout of thrust bearings)

S_e Housing washer thickness variation from raceway middle to back face (axial runout of thrust bearings)

*) The overall height of the thrust bearing is designated with T in the standard.

Tolerances of radial bearings (except tapered roller bearings)

Inner ring

Dimensions in mm

Tolerance class PN (normal tolerance)

Tolerance in microns (0.001 mm)

Nominal bore diameter	over to	2.5 10	10 18	18 30	30 50	50 80	80 120	120 180	180 250	250 315	315 400	400 500	500 630	630 800	800 1000	1000 1250	1250 1600	1600 2000
Bore, cylindrical Deviation	Δ_{dmp}	0 −8	0 −8	0 −10	0 −12	0 −15	0 −20	0 −25	0 −30	0 −35	0 −40	0 −45	0 −50	0 −75	0 −100	0 −125	0 −160	0 −200
Variation V_{dp}	diameter series 7·8·9	10	10	13	15	19	25	31	38	44	50	56	63					
	0·1	8	8	10	12	19	25	31	38	44	50	56	63					
	2·3·4	6	6	8	9	11	15	19	23	26	30	34	38					
Variation	V_{dmp}	6	6	8	9	11	15	19	23	26	30	34	38					
Bore, taper 1:12 Deviation	Δ_{dmp}	+15 0	+18 0	+21 0	+25 0	+30 0	+35 0	+40 0	+46 0	+52 0	+57 0	+63 0	+70 0	+80 0	+90 0	+105 0	+125 0	+150 0
Deviation	Δ_{d1mp} – Δ_{dmp}	+15 0	+18 0	+21 0	+25 0	+30 0	+35 0	+40 0	+46 0	+52 0	+57 0	+63 0	+70 0	+80 0	+90 0	+105 0	+125 0	+150 0
Variation	V_{dp}	10	10	13	15	19	25	31	38	44	50	56						
Bore, taper 1:30 Deviation	Δ_{dmp}					+15 0	+20 0	+25 0	+30 0	+35 0	+40 0	+45 0	+50 0	+75 0	+100 0	+125 0	+160 0	+200 0
Deviation	Δ_{d1mp} – Δ_{dmp}					+35 0	+40 0	+50 0	+55 0	+60 0	+65 0	+75 0	+85 0	+100 0	+100 0	+115 0	+125 0	+150 0
Variation	V_{dp}					19	25	31	38	44	50	56	63					
Width deviation	Δ_{Bs}	0 −120	0 −120	0 −120	0 −120	0 −150	0 −200	0 −250	0 −300	0 −35 0	0 −400	0 −450	0 −500	0 −750	0 −1000	0 −1250	0 −1600	0 −2000
Width variation	V_{Bs}	15	20	20	20	25	25	30	30	35	40	50	60	70	80	100	120	140
Radial runout	K_{ia}	10	10	13	15	20	25	30	40	50	60	65	70	80	90	100	120	140

Tolerance class P6

Nominal bore diameter	over to	2.5 10	10 18	18 30	30 50	50 80	80 120	120 180	180 250	250 315	315 400	400 500	500 630	630 800	800 1000	1000 1250	1250 1600	1600 2000
Deviation	Δ_{dmp}	0 −7	0 −7	0 −8	0 −10	0 −12	0 −15	0 −18	0 −22	0 −25	0 −30	0 −35	0 −40	0 −50	0 −65	0 −80	0 −100	0 −130
Variation V_{dp}	diameter series 7·8·9	9	9	10	13	15	19	23	28	31	38	44	50					
	0·1	7	7	8	10	15	19	23	28	31	38	44	50					
	2·3·4	5	5	6	8	9	11	14	17	19	23	26	30					
Variation	V_{dmp}	5	5	6	8	9	11	14	17	19	23	26	30					
Width deviation	Δ_{Bs}	0 −120	0 −120	0 −120	0 −120	0 −150	0 −200	0 −250	0 −300	0 −350	0 −400	0 −450	0 −500	0 −750	0 −1000	0 −1250	0 −1600	0 −2000
Width variation	V_{Bs}	15	20	20	20	25	25	30	30	35	40	45	50	55	60	70	70	80
Radial runout	K_{ia}	6	7	8	10	10	13	18	20	25	30	35	40	50	60	80	80	100

See page 181 for the width tolerances Δ_{Bs} for angular contact ball bearings of universal design.

Outer ring

Dimensions in mm

Tolerance class PN (normal tolerance)

Tolerances in microns (0.001 mm)

Nominal outside diameter	over to	6 18	18 30	30 50	50 80	80 120	120 150	150 180	180 250	250 315	315 400	400 500	500 630	630 800	800 1000	1000 1250	1250 1600	1600 2000	2000 2500
Deviation	Δ_{Dmp}	0 −8	0 −9	0 −11	0 −13	0 −15	0 −18	0 −25	0 −30	0 −35	0 −40	0 −45	0 −50	0 −75	0 −100	0 −125	0 −160	0 −200	0 −250
Variation V_{Dp}	diameter series 7·8·9	10	12	14	16	19	23	31	38	44	50	56	63	94	125				
	0·1	8	9	11	13	19	23	31	38	44	50	56	63	94	125				
	2·3·4	6	7	8	10	11	14	19	23	26	30	34	38	55	75				
	sealed bear-ings 2·3·4	10	12	16	20	26	30	38											
Variation	V_{Dmp}	6	7	8	10	11	14	19	23	26	30	34	38	55	75				
Radial runout	K_{ea}	15	15	20	25	35	40	45	50	60	70	80	100	120	140	160	190	220	250

The width tolerances Δ_{Cs} and V_{Cs} are identical to Δ_{Bs} and V_{Bs} for the inner ring.

Tolerance class P6

Nominal outside diameter	over to	6 18	18 30	30 50	50 80	80 120	120 150	150 180	180 250	250 315	315 400	400 500	500 630	630 800	800 1000	1000 1250	1250 1600	1600 2000	2000 2500
Deviation	Δ_{Dmp}	0 −7	0 −8	0 −9	0 −11	0 −13	0 −15	0 −18	0 −20	0 −25	0 −28	0 −33	0 −38	0 −45	0 −60	0 −80	0 −100	0 −140	0 −180
Variation V_{Dp}	diameter series 7·8·9	9	10	11	14	16	19	23	25	31	35	41	48	56	75				
	0·1	7	8	9	11	16	19	23	25	31	35	41	48	56	75				
	2·3·4	5	6	7	8	10	11	14	15	19	21	25	29	34	45				
	sealed bearings 0·1·2·3·4	9	10	13	16	20	25	30											
Variation	V_{Dmp}	5	6	7	8	10	11	14	15	19	21	25	29	34	45				
Runout	K_{ea}	8	9	10	13	18	20	23	25	30	35	40	50	60	75	100	100	100	120

The width tolerances Δ_{Cs} and V_{Cs} are identical to Δ_{Bs} and V_{Bs} for the inner ring.

Tolerances of radial bearings (except tapered roller bearings)

Inner ring

Dimensions in mm

Tolerance class P5

Tolerances in microns (0.001 mm)

Nominal bore diameter	over to	2.5 10	10 18	18 30	30 50	50 80	80 120	120 180	180 250	250 315	315 400	400 500	500 630	630 800
Deviation	Δ_{dmp}	0 –5	0 –5	0 –6	0 –8	0 –9	0 –10	0 –13	0 –15	0 –18	0 –23	0 –27	0 –33	0 –40
Variation V_{dp}	diameter series 7 · 8 · 9	5	5	6	8	9	10	13	15	18	23			
	0 · 1 · 2 · 3 · 4	4	4	5	6	7	8	10	12	14	18			
Variation	V_{dmp}	3	3	3	4	5	5	7	8	9	12			
Width deviation	Δ_{Bs}	0 –40	0 –80	0 –120	0 –120	0 –150	0 –200	0 –250	0 –300	0 –350	0 –400	0 –450	0 –500	0 –750
Width deviation	V_{Bs}	5	5	5	5	6	7	8	10	13	15	17	20	30
Radial runout	K_{ia}	4	4	4	5	5	6	8	10	13	15	17	20	25
Axial runout	S_d	7	7	8	8	8	9	10	11	13	15	17	20	30
Axial runout	S_{ia}	7	7	8	8	8	9	10	13	15	20	23	25	30

The axial runout values S_{ia} apply to ball bearings (except self-aligning ball bearings).

Tolerance class P4

Deviation	Δ_{dmp}, Δ_{ds}*)	0 –4	0 –4	0 –5	0 –6	0 –7	0 –8	0 –10	0 –12	0 –15	0 –19	0 –23	0 –26	0 –34
Variation V_{dp}	diameter series 7 · 8 · 9	4	4	5	6	7	8	10	12					
	0 · 1 · 2 · 3 · 4	3	3	4	5	5	6	8	9					
Variation	V_{dmp}	2	2	2.5	3	3.5	4	5	6					
Width deviation	Δ_{Bs}	0 –40	0 –80	0 –120	0 –120	0 –150	0 –200	0 –250	0 –300	0 –350	0 –400	0 –450	0 –500	0 –750
Width variation	V_{Bs}	2.5	2.5	2.5	3	4	4	5	6	7	8	9	10	15
Radial runout	K_{ia}	2.5	2.5	3	4	4	5	6	8	8	10	10	12	15
Axial runout	S_d	3	3	4	4	5	5	6	7	7	8	9	10	15
Axial runout	S_{ia}	3	3	4	4	5	5	7	8	10	12	13	15	20

The axial runout values S_{ia} apply to ball bearings (except self-aligning ball bearings).
*) These values Δ_{ds} and Δ_{Ds} apply only to diameter series 0 · 1 · 2 · 3 · 4.
See page 181 for the width tolerances Δ_{Bs} for angular contact ball bearings of universal design.

Outer ring

Dimensions in mm

Tolerance class P5

Tolerances in microns (0.001 mm)

Nominal outside diameter	over to	6 18	18 30	30 50	50 80	80 120	120 150	150 180	180 250	250 315	315 400	400 500	500 630	630 800	800 1000	1000 1250	1250 1600
Deviation	Δ_{Dmp}	0 –5	0 –6	0 –7	0 –9	0 –10	0 –11	0 –13	0 –15	0 –18	0 –20	0 –23	0 –28	0 –35	0 –40	0 –50	0 –65
Variation V_{Dp}	diameter series 7 · 8 · 9	5	6	7	9	10	11	13	15	18	20	23	28	35			
	0 · 1 · 2 · 3 · 4	4	5	5	7	8	8	10	11	14	15	17	21	26			
Variation	V_{Dmp}	3	3	4	5	5	6	7	8	9	10	12	14	18			
Width variation	V_{Cs}	5	5	5	6	8	8	8	10	11	13	15	18	20	25	30	40
Radial runout	K_{ea}	5	6	7	8	10	11	13	15	18	20	23	25	30	35	50	65
Variation of inclination	S_D	8	8	8	8	9	10	10	11	13	13	15	18	20	30	40	50
Axial runout	S_{ea}	8	8	8	10	11	13	14	15	18	20	23	25	30	40	55	70

The width tolerance Δ_{Cs} is identical to Δ_{Bs} for the inner ring.
The axial runout values S_{ea} apply to ball bearings (except self-aligning ball bearings).

Tolerance class P4

Deviation	Δ_{Dmp}, Δ_{Ds} *)	0 –4	0 –5	0 –6	0 –7	0 –8	0 –9	0 –10	0 –11	0 –13	0 –15	0 –20	0 –25	0 –28	0 –35	0 –40	0 –55
Variation V_{Dp}	diameter series 7 · 8 · 9	4	5	6	7	8	9	10	11	13	15						
	0 · 1 · 2 · 3 · 4	3	4	5	5	6	7	8	8	10	11						
Variation	V_{Dmp}	2	2.5	3	3.5	4	5	5	6	7	8						
Width variation	V_{Cs}	2.5	2.5	2.5	3	4	5	5	7	7	8	9	10	12	15	20	25
Radial runout	K_{ea}	3	4	5	5	6	7	8	10	11	13	14	17	20	25	30	40
Variation of inclination	S_D	4	4	4	4	5	5	5	7	8	10	10	12	14	20	25	30
Axial runout	S_{ea}	5	5	5	5	6	7	8	10	10	13	15	18	22	28	35	45

The width tolerance Δ_{Cs} is identical to Δ_{Bs} for the inner ring.
The axial runout values S_{ea} apply to ball bearings (except self-aligning ball bearings).

Tolerances of spindle bearings

Inner ring

Dimensions in mm

Nominal bore diameter	over / to	– / 10	10 / 18	18 / 30	30 / 50	50 / 80	80 / 120	120 / 150	150 / 180	180 / 250

Tolerance class P4S

Tolerances in microns (0.001 mm)

Deviation	Δ_{dmp}	0 / –4	0 / –4	0 / –5	0 / –6	0 / –7	0 / –8	0 / –10	0 / –10	0 / –12
Width deviation	Δ_{Bs}	0 / –40	0 / –80	0 / –120	0 / –120	0 / –150	0 / –200	0 / –250	0 / –250	0 / –300
Width variation	V_{Bs}	2.5	2.5	2.5	3	4	4	5	5	6
Radial runout	K_{ia}	1.5	1.5	2.5	2.5	2.5	2.5	2.5	5	5
Axial runout	S_d	1.5	1.5	1.5	1.5	1.5	2.5	2.5	4	5
Axial runout	S_{ia}	1.5	1.5	2.5	2.5	2.5	2.5	2.5	5	5

See page 202 for width tolerances Δ_{Bs} for spindle bearings of universal design.

Outer ring

Dimensions in mm

Nominal outside diameter	over / to	18 / 30	30 / 50	50 / 80	80 / 120	120 / 150	150 / 180	180 / 250	250 / 315	315 / 400

Tolerance class P4S

Tolerances in microns (0.001 mm)

Deviation	Δ_{Dmp}	0 / –5	0 / –6	0 / –7	0 / –8	0 / –9	0 / –10	0 / –11	0 / –13	0 / –15
Width variation	V_{Cs}	2.5	2.5	3	4	5	5	7	7	8
Radial runout	K_{ea}	2.5	2.5	4	5	5	5	7	7	8
Axial runout	S_D	1.5	1.5	1.5	2.5	2.5	2.5	4	5	7
Axial runout	S_{ea}	2.5	2.5	4	5	5	5	7	7	8

The width tolerance Δ_{Cs} is identical to Δ_{Bs} for the inner ring.

Bearing Data
Tolerances

Tolerances of radial bearings (except tapered roller bearings)

Inner ring

Dimensions in mm

Nominal bore diameter		over to	18 30	30 50	50 80	80 120	120 180	180 250	250 315	315 400	400 500	500 630	630 800	800 1000	1000 1250

Tolerance class SP (double row cylindrical roller bearings)

Tolerances in microns (0.001 mm)

		18–30	30–50	50–80	80–120	120–180	180–250	250–315	315–400	400–500	500–630	630–800	800–1000	1000–1250
Bore, cylindrical Deviation	Δ_{dmp}, Δ_{ds}	0 –6	0 –8	0 –9	0 –10	0 –13	0 –15	0 –18	0 –23	0 –27	0 –30	0 –40	0 –50	0 –65
Variation	V_{dp}	3	4	5	5	7	8	9	12	14				
Bore, tapered Deviation	Δ_{ds}	+10 0	+12 0	+15 0	+20 0	+25 0	+30 0	+35 0	+40 0	+45 0	+50 0	+65 0	+75 0	+90 0
Variation	V_{dp}	3	4	5	5	7	8	9	12	14				
Deviation	Δ_{d1mp}-Δ_{dmp}	+4 0	+6 0	+6 0	+8 0	+8 0	+10 0	+12 0	+12 0	+14 0				
Width deviation	Δ_{Bs}	0 –100	0 –120	0 –150	0 –200	0 –250	0 –300	0 –350	0 –400	0 –450	0 –500	0 –750	0 –1000	0 –1250
Width variation	V_{Bs}	5	5	6	7	8	10	13	15	17	20	30	33	40
Radial runout	K_{ia}	3	4	4	5	6	8	8	10	10	12	15	17	20
Axial runout	S_d	8	8	8	9	10	11	13	15	17	20	23	30	40
Axial runout	S_{ia}	8	8	8	9	10	13	15	20	23	25	30	40	50

Tolerance class UP (double row cylindrical roller bearings)

		18–30	30–50	50–80	80–120	120–180	180–250	250–315	315–400	400–500	500–630	630–800	800–1000	1000–1250
Bore, cylindrical Deviation	Δ_{dmp}, Δ_{ds}	0 –5	0 –6	0 –7	0 –8	0 –10	0 –12	0 –15	0 –19	0 –23	0 –26	0 –34	0 –40	0 –55
Variation	V_{dp}	2.5	3	3.5	4	5	6	8	10	12				
Bore, tapered Deviation	Δ_{ds}	+6 0	+7 0	+8 0	+10 0	+12 0	+14 0	+15 0	+17 0	+19 0	+20 0	+22 0	+25 0	+30 0
Variation	V_{dp}	2.5	3	3.5	4	5	6	8	10	12				
Deviation	Δ_{d1mp}-Δ_{dmp}	+2 0	+3 0	+3 0	+4 0	+4 0	+5 0	+6 0	+6 0	+7 0				
Width deviation	Δ_{Bs}	0 –25	0 –30	0 –40	0 –50	0 –60	0 –75	0 –100	0 –100	0 –100	0 –125	0 –125	0 –125	0 –125
Width variation	V_{Bs}	1.5	2	3	3	4	5	5	6	7	8	11	12	15
Radial runout	K_{ia}	1.5	2	2	3	3	4	4	5	5	6	7	9	10
Axial runout	S_d	3	3	4	4	5	6	6	7	8	9	11	12	15
Axial runout	S_{ia}	3	3	3	4	6	7	8	9	10	12	18	19	23

Outer ring

Dimensions in mm

Nominal outside diameter		over to	30 50	50 80	80 120	120 150	150 180	180 250	250 315	315 400	400 500	500 630	630 800	800 1000	1000 1250	1250 1600

Tolerance class SP (double row cylindrical roller bearings)

Tolerances in microns (0.001 mm)

		30–50	50–80	80–120	120–150	150–180	180–250	250–315	315–400	400–500	500–630	630–800	800–1000	1000–1250	1250–1600
Deviation	Δ_{Dmp}, Δ_{Ds}	0 –7	0 –9	0 –10	0 –11	0 –13	0 –15	0 –18	0 –20	0 –23	0 –28	0 –35	0 –40	0 –50	0 –65
Variation	V_{Dp}	4	5	5	6	7	8	9	10	12	14	18			
Radial runout	K_{ea}	5	5	6	7	8	10	11	13	15	17	20	25	30	30
Variation of inclination	S_D	8	8	9	10	10	11	13	13	15	18	20	30	40	50
Axial runout	S_{ea}	8	10	11	13	14	15	18	20	23	25	30	40	55	70

The width tolerances Δ_{Cs} and V_{Cs} are identical to Δ_{Bs} and V_{Bs} for the inner ring.

Tolerance class UP (double row cylindrical roller bearings)

		30–50	50–80	80–120	120–150	150–180	180–250	250–315	315–400	400–500	500–630	630–800	800–1000	1000–1250	1250–1600
Deviation	Δ_{Dmp},Δ_{Ds}	0 –5	0 –6	0 –7	0 –8	0 –9	0 –10	0 –12	0 –14	0 –17	0 –20	0 –25	0 –30	0 –36	0 –48
Variation	V_{Dp}	3	3	4	4	5	5	6	7	9	10	13			
Radial runout	K_{ea}	3	3	3	4	4	5	6	7	8	9	11	12	15	19
Variation of inclination	S_D	2	2	3	3	3	4	4	5	5	6	7	10	12	15
Axial runout	S_{ea}	4	4	5	6	7	9	9	12	12	14	17	21	26	34

The width tolerances Δ_{Cs} and V_{Cs} are identical to Δ_{Bs} and V_{Bs} for the inner ring.

Tolerances of tapered roller bearings in metric dimensions

Cone

Dimensions in mm

Nominal bore diameter	over to	10 18	18 30	30 50	50 80	80 120	120 180	180 250	250 315	315 400	400 500	500 630	630 800	800 1000

Tolerance class PN (normal tolerance)

Tolerances in microns (0.001 mm)

Deviation	Δ_{dmp}	0 –12	0 –12	0 –12	0 –15	0 –20	0 –25	0 –30	0 –35	0 –40	0 –45	0 –50	0 –75	0 –100
Variation	V_{dp}	12	12	12	15	20	25	30	35	40	45	50	75	100
	V_{dmp}	9	9	9	11	15	19	23	26	30				
Width deviation	Δ_{Bs}	0 –120	0 –120	0 –120	0 –150	0 –200	0 –250	0 –300	0 –350	0 –400	0 –450	0 –500	0 –750	0 –1000
Radial runout	K_{ia}	15	18	20	25	30	35	50	60	70	70	85	100	120
Width deviation	Δ_{Ts}	+200 0	+200 0	+200 0	+200 0	+200 –200	+350 –250	+350 –250	+350 –250	+400 –400	+400 –400	+500 –500	+600 –600	+750 –750
	Δ_{T1s}	+100 0	+100 0	+100 0	+100 0	+100 –100	+150 –150	+150 –150	+150 –150	+200 –200				
	Δ_{T2s}	+100 0	+100 0	+100 0	+100 0	+100 –100	+200 –100	+200 –100	+200 –100	+200 –200				

Tolerance class P6X

Deviation	Δ_{dmp}	0 –12	0 –12	0 –12	0 –15	0 –20	0 –25	0 –30	0 –35	0 –40
Variation	V_{dp}	12	12	12	15	20	25	30	35	40
	V_{dmp}	9	9	9	11	15	19	23	26	30
Width deviation	Δ_{Bs}	0 –50	0 –50	0 –50	0 –50	0 –50	0 –50	0 –50	0 –50	0 –50
Radial runout	K_{ia}	15	18	20	25	30	35	50	60	70
Width deviation	Δ_{Ts}	+100 0	+100 0	+100 0	+100 0	+100 0	+150 0	+150 0	+200 0	+200 0
	Δ_{T1s}	+50 0	+50 0	+50 0	+50 0	+50 0	+50 0	+50 0	+100 0	+100 0
	Δ_{T2s}	+50 0	+50 0	+50 0	+50 0	+50 0	+100 0	+100 0	+100 0	+100 0

Tapered roller bearings without flange of the series 320X, 329, 330, 331, 332 ($d \leqq 200$ mm) have the tolerance class P6X.

Cup

Dimensions in mm

Nominal outside diameter	over to	18 30	30 50	50 80	80 120	120 150	150 180	180 250	250 315	315 400	400 500	500 630	630 800	800 1000	1000 1250	1250 1600

Tolerance class PN (normal tolerance)

Tolerances in microns (0.001 mm)

Deviation	Δ_{Dmp}	0 –12	0 –14	0 –16	0 –18	0 –20	0 –25	0 –30	0 –35	0 –40	0 –45	0 –50	0 –75	0 –100	0 –125	0 –160
Variation	V_{Dp}	12	14	16	18	20	25	30	35	40	45	50	75	100	125	160
	V_{Dmp}	9	11	12	14	15	19	23	26	30	34	38				
Width deviation	Δ_{Cs}	The width tolerance Δ_{Cs} is identical to Δ_{Bs} for the cone.														
Radial runout	K_{ea}	18	20	25	35	40	45	50	60	70	80	100	120	120	120	120

Tolerance class P6X

Deviation	Δ_{Dmp}	0 –12	0 –14	0 –16	0 –18	0 –20	0 –25	0 –30	0 –35	0 –40	0 –45	0 –50
Variation	V_{Dp}	12	14	16	18	20	25	30	35	40	45	50
	V_{Dmp}	9	11	12	14	15	19	23	26	30	34	38
Width deviation	Δ_{Cs}	0 –100	0 –100	0 –100	0 –100	0 –100	0 –100	0 –100	0 –100	0 –100	0 –100	0 –100
Radial runout	K_{ea}	18	20	25	35	40	45	50	60	70	80	100

Tolerances of tapered roller bearings in metric dimensions

Cone

Dimensions in mm

Nominal bore diameter	over to	**10** **18**	**18** **30**	**30** **50**	**50** **80**	**80** **120**	**120** **180**	**180** **250**	**250** **315**	**315** **400**	**400** **500**	**500** **630**	**630** **800**
Tolerance class P5		Tolerances in microns (0.001 mm)											
Deviation	Δ_{dmp}	0 –7	0 –8	0 –10	0 –12	0 –15	0 –18	0 –22	0 –25	0 –30	0 –35	0 –40	0 –75
Variation	V_{dp}	5	6	8	9	11	14	17					
	V_{dmp}	5	5	5	6	8	9	11					
Width deviation	Δ_{Bs}	0 –200	0 –200	0 –240	0 –300	0 –400	0 –500	0 –600					
Radial runout	K_{ia}	5	5	6	7	8	11	13					
Axial runout	S_d	7	8	8	8	9	10	11	13	15	17	20	30
Width deviation	Δ_{Ts}	+200 –200	+200 –200	+200 –200	+200 –200	+200 –200	+350 –250	+350 –250	+350 –250	+400 –400	+400 –400	+500 –500	+600 –600

Tolerance class P4		10–18	18–30	30–50	50–80	80–120	120–180	180–250
Deviation	Δ_{dmp}, Δ_{ds}	0 –5	0 –6	0 –8	0 –9	0 –10	0 –13	0 –15
Variation	V_{dp}	4	5	6	7	8	10	11
	V_{dmp}	4	4	5	5	5	7	8
Width deviation	Δ_{Bs}	0 –200	0 –200	0 –240	0 –300	0 –400	0 –500	0 –600
Radial runout	K_{ia}	3	3	4	4	5	6	8
Axial runout	S_d	3	4	4	5	5	6	7
Axial runout	S_{ia}	3	4	4	4	5	7	8
Width deviation	Δ_{Ts}	+200 –200	+200 –200	+200 –200	+200 –200	+200 –200	+350 –250	+350 –250

Cup

Dimensions in mm

Nominal outside diameter	over to	**18** **30**	**30** **50**	**50** **80**	**80** **120**	**120** **150**	**150** **180**	**180** **250**	**250** **315**	**315** **400**	**400** **500**	**500** **630**	**630** **800**	**800** **1000**
Tolerance class P5		Tolerances in microns (0.001 mm)												
Deviation	Δ_{Dmp}	0 –8	0 –9	0 –11	0 –13	0 –15	0 –18	0 –20	0 –25	0 –28	0 –33	0 –38	0 –45	0 –60
Variation	V_{Dp}	6	7	8	10	11	14	15	19	22				
	V_{Dmp}	5	5	6	7	8	9	10	13	14				
Width deviation	Δ_{Cs}	The width tolerance Δ_{Cs} is identical to Δ_{Bs} for the cone.												
Radial runout	K_{ea}	6	7	8	10	11	13	15	18	20	23	25	30	35
Variation of inclination	S_D	8	8	8	9	10	10	11	13	13	15	18	20	30

Tolerance class P4		18–30	30–50	50–80	80–120	120–150	150–180	180–250	250–315	315–400
Deviation	Δ_{Dmp}, Δ_{Ds}	0 –6	0 –7	0 –9	0 –10	0 –11	0 –13	0 –15	0 –18	0 –20
Variation	V_{Dp}	5	5	7	8	8	10	11	14	15
	V_{Dmp}	4	5	5	5	6	7	8	9	10
Width deviation	Δ_{Cs}	The width tolerance Δ_{Cs} is identical to Δ_{Bs} for the cone.								
Radial runout	K_{ea}	4	5	5	6	7	8	10	11	13
Variation of inclination	S_D	4	4	4	5	5	5	7	8	10
Axial runout	S_{ea}	5	5	5	6	7	8	10	10	13

Tolerances of tapered roller bearings in inch dimensions

Cone

Dimensions in mm

Nominal bore diameter		over		81	102	127	305	508	610	915	1220
		to	81	102	127	305	508	610	915	1220	

Normal tolerance

Tolerances in microns (0.001 mm)

Deviation Δ_{dmp}	+13 0	+25 0	+25 0	+25 0	+50 0	+50 0	+75 0	+100 0	+125 0
Width deviation Δ_{Bs}	Normal tolerance of metric tapered roller bearings								
Radial runout K_{ia}	Normal tolerance of metric tapered roller bearings								
Single row bearings Width deviation Δ_{Ts}	+200 0	+200 0	+350 –250	+350 –250	+375 –375	+375 –375	+375 –375	+375 –375	+375 –375

Dimensions in mm

Nominal bore diameter		over		150	250	315	500
		to	150	250	315	500	710

Tolerance class Q3

Tolerances in microns (0.001 mm)

Deviation Δ_{dmp}	+11 0	+13 0	+13 0	+20 0	+25 0
Width deviation Δ_{Bs}	0 –250	0 –300	0 –350	0 –400	0 –600
Width variation V_{Bs}	2	3	5	7	10
Radial runout K_{ia}	4	4	4	7	9
Axial runout S_d	4	6	7	8	10
Axial runout S_{ia}	4	6	8	10	13
Single row bearings Width deviation Δ_{Ts}	+200 –200	+200 –200	+200 –200	+200 –200	+380 –380

Cup

Dimensions in mm

Nominal outside diameter		over		305	610	915	1220
		to	305	610	915	1220	

Normal tolerance

Tolerances in microns (0.001 mm)

Deviation Δ_{Dmp}	+25 0	+50 0	+75 0	+100 0	+125 0
Radial runout K_{ea}	Normal tolerance of metric tapered roller bearings				

Dimensions in mm

Nominal outside diameter		over		150	250	315	500	630
		to	150	250	315	500	630	900

Tolerance class Q3

Tolerances in microns (0.001 mm)

Deviation Δ_{Dmp}	+11 0	+13 0	+13 0	+20 0	+25 0	+38 0
Width variation V_{Cs}	2	3	5	7	10	20
Radial runout K_{ea}	4	4	4	7	9	18
Variation of inclination S_D	4	6	7	8	10	20

Tolerances of thrust bearings

Shaft washer

Dimensions in mm

Nominal bore diameter															
	over		18	30	50	80	120	180	250	315	400	500	630	800	1000
	to	18	30	50	80	120	180	250	315	400	500	630	800	1000	1250

Tolerance class PN (normal tolerance)

Tolerances in microns (0.001 mm)

Deviation	Δ_{dmp}	0 –8	0 –10	0 –12	0 –15	0 –20	0 –25	0 –30	0 –35	0 –40	0 –45	0 –50	0 –75	0 –100	0 –125
Variation	V_{dp}	6	8	9	11	15	19	23	26	30	34	38			
Wall thickness variation	S_i	10	10	10	10	15	15	20	25	30	30	35	40	45	50
Seating washer deviation	Δ_{du}	+70 0	+70 0	+85 0	+100 0	+120 0	+140 0	+140 0	+160 0	+180 0	+180 0				

Tolerance class P6

Deviation	Δ_{dmp}	0 –8	0 –10	0 –12	0 –15	0 –20	0 –25	0 –30	0 –35	0 –40	0 –45	0 –50	0 –75	0 –100	0 –125
Variation	V_{dp}	6	8	9	11	15	19	23	26	30	34	38			
Wall thickness variation	S_i	5	5	6	7	8	9	10	13	15	18	21	25	30	35

Tolerance class P5

Deviation	Δ_{dmp}	0 –8	0 –10	0 –12	0 –15	0 –20	0 –25	0 –30	0 –35	0 –40	0 –45	0 –50	0 –75	0 –100	0 –125
Variation	V_{dp}	6	8	9	11	15	19	23	26	30	34	38			
Wall thickness variation	S_i	3	3	3	4	4	5	5	7	7	9	11	13	15	18

Tolerance class P4

Deviation	Δ_{dmp}	0 –7	0 –8	0 –10	0 –12	0 –15	0 –18	0 –22	0 –25	0 –30	0 –35	0 –40	0 –50	0 –70	0 –100
Variation	V_{dp}	5	6	8	9	11	14	17	19	23	26	30			
Wall thickness variation	S_i	2	2	2	3	3	4	4	5	5	6	7	8	8	9

Tolerance class SP (angular contact thrust ball bearings, series 2344 and 2347)

Deviation	Δ_{dmp}		0 –8	0 –10	0 –12	0 –15	0 –18	0 –22	0 –25	0 –30					
Variation	V_{dp}		6	8	9	11	14	17							
Wall thickness variation	S_i		3	3	4	4	5	5	7	7					
Height deviation	Δ_{Hs}		+50 –150	+75 –200	+100 –250	+125 –300	+150 –350	+175 –400	+200 –450	+250 –600					

Housing washer

Dimensions in mm

Nominal outside diameter															
	over	18	30	50	80	120	180	250	315	400	500	630	800	1000	1250
	to	30	50	80	120	180	250	315	400	500	630	800	1000	1250	1600

Tolerance class PN (normal tolerance)

Tolerances in microns (0.001 mm)

Deviation	Δ_{Dmp}	0 –13	0 –16	0 –19	0 –22	0 –25	0 –30	0 –35	0 –40	0 –45	0 –50	0 –75	0 –100	0 –125	0 –160
Variation	V_{Dp}	10	12	14	17	19	23	26	30	34	38	55	75		
Wall thickness variation	S_e	The wall thickness variation S_e of the housing washer is identical to S_i of the shaft washer.													
Seating washer deviation	Δ_{Du}	0 –30	0 –35	0 –45	0 –60	0 –75	0 –90	0 –105	0 –120	0 –135	0 –180				

Tolerance class P6

Deviation	Δ_{Dmp}	0 –13	0 –16	0 –19	0 –22	0 –25	0 –30	0 –35	0 –40	0 –45	0 –50	0 –75	0 –100	0 –125	0 –160
Variation	V_{Dp}	10	12	14	17	19	23	26	30	34	38	55	75		
Wall thickness variation	S_e	The wall thickness variation S_e of the housing washer is identical to S_i of the shaft washer.													

Tolerance class P5

Deviation	Δ_{Dmp}	0 –13	0 –16	0 –19	0 –22	0 –25	0 –30	0 –35	0 –40	0 –45	0 –50	0 –75	0 –100	0 –125	0 –160
Variation	V_{Dp}	10	12	14	17	19	23	26	30	34	38	55	75		
Wall thickness variation	S_e	The wall thickness variation S_e of the housing washer is identical to S_i of the shaft washer.													

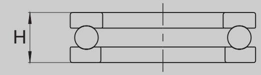
Tolerance class P4

Deviation	Δ_{Dmp}	0 –8	0 –9	0 –11	0 –13	0 –15	0 –20	0 –25	0 –28	0 –33	0 –38	0 –45	0 –70	0 –90	0 –125
Variation	V_{Dp}	6	7	8	10	11	15	19	21	25	29	34			
Wall thickness variation	S_e	The wall thickness variation S_e of the housing washer is identical to S_i of the shaft washer.													

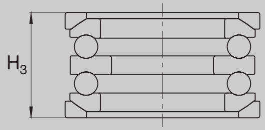
Tolerance class SP (angular contact thrust ball bearings, series 2344 and 2347)

Deviation	Δ_{Dmp}			–24 –43	–28 –50	–33 –58	–37 –66	–41 –73	–46 –82	–50 –90	–55 –99				
Variation	V_{Dp}			6	8	9	10	12							
Wall thickness variation	S_e	The wall thickness variation S_e of the housing washer is identical to S_i of the shaft washer.													

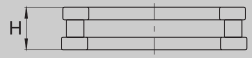
Section heights of thrust bearings



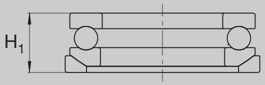
Thrust ball bearing



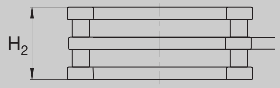
Thrust ball bearing
double direction
with seating washers



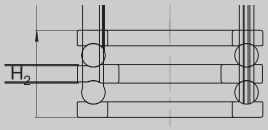
Cylindrical roller thrust bearing



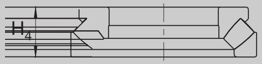
Thrust ball bearing
with seating washer



Cylindrical roller thrust bearing
double direction



Thrust ball bearing
double direction



Spherical roller thrust bearing

Section heights of thrust bearings

Dimensions in mm

Nominal bore diameter	over to	 **30**	**30** **50**	**50** **80**	**80** **120**	**120** **180**	**180** **250**	**250** **315**	**315** **400**	**400** **500**	**500** **630**	**630** **800**	**800** **1000**	**1000** **1250**

Tolerance classes PN to P4

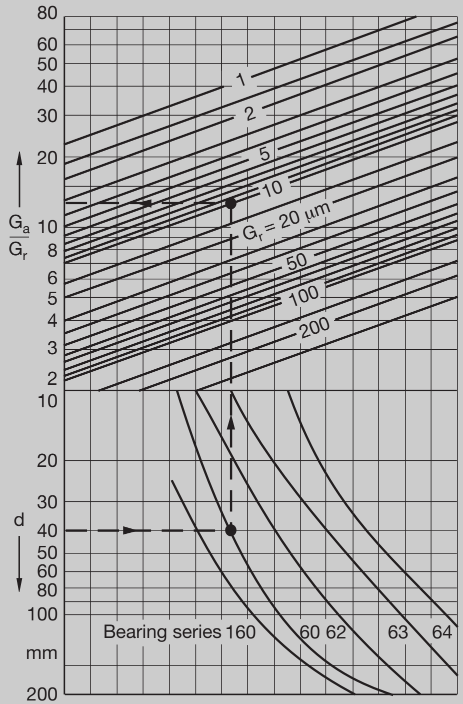
Tolerances in microns (0.001 mm)

Deviation	Δ_{Hs}	+20 –250	+20 –250	+20 –300	+25 –300	+25 –400	+30 –400	+40 –400	+40 –500	+50 –500	+60 –600	+70 –750	+80 –1000	+100 –1400
	Δ_{H1s}	+100 –250	+100 –250	+100 –300	+150 –300	+150 –400	+150 –400	+200 –400	+200 –500	+300 –500	+350 –600	+400 –750	+450 –1000	+500 –1400
	Δ_{H2s}	+150 –400	+150 –400	+150 –500	+200 –500	+200 –600	+250 –600	+350 –700	+350 –700	+400 –900	+500 –1100	+600 –1300	+700 –1500	+900 –1800
	Δ_{H3s}	+300 –400	+300 –400	+300 –500	+400 –500	+400 –600	+500 –600	+600 –700	+600 –700	+750 –900	+900 –1100	+1100 –1300	+1300 –1500	+1600 –1800
	Δ_{H4s}	+20 –300	+20 –300	+20 –400	+25 –400	+25 –500	+30 –500	+40 –700	+40 –700	+50 –900	+60 –1200	+70 –1400	+80 –1800	+100 –2400

Bearing clearance

The bearing clearance is the measurement by which one bearing ring can be displaced in relation to the other one either in the radial direction (radial clearance) or in the axial direction (axial clearance) from one end position to the other. In the case of some bearing types radial and axial clearances depend on each other, see table.

▼ Relation between radial and axial clearances with deep groove ball bearings



d = bearing bore [mm]
G_r = radial clearance [µm]
G_a = axial clearance [µm]

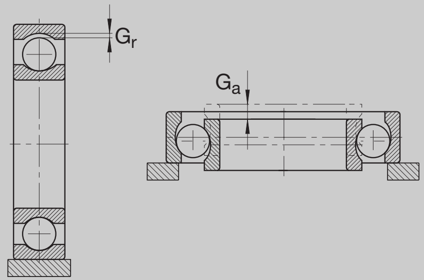
Example:
Deep groove ball bearing 6008.C3 with d = 40 mm
Radial clearance before mounting: 15 to 33 µm
Actual radial clearance: G_r = 24 µm

Mounting tolerances: Shaft k5
Housing J6
Radial clearance reduction during mounting: 14 µm
Radial clearance after mounting: 24 µm - 14 µm = 10 µm

According to this diagram G_a/G_r = 13

Axial clearance: G_a = 13 · 10 µm = 130 µm

▼ Bearing clearance
G_a = axial clearance, G_r = radial clearance



▼ Relation between radial and axial clearance with other bearing types

Bearing type	G_a/G_r
Angular contact ball bearings, single row, series 72B and 73B and arranged in pairs	1.2
Four-point bearings	1.4
Angular contact ball bearings, double row, series 32 and 33 series 32B and 33B	1.4 2
Self-aligning ball bearings	2.3 · Y_0*)
Tapered roller bearings, single row, arranged in pairs	4.6 · Y_0*)
Tapered roller bearing pairs adjusted (N11CA)	2.3 · Y_0*)
Spherical roller bearings	2.3 · Y_0*)

*) Y_0 value from bearing tables

Bearing Data
Bearing clearance

There is a distinction made between the clearance of the bearing prior to mounting and the clearance of the mounted bearing at operating temperature (operating clearance). The operating clearance should be as small as possible for the shaft to be guided perfectly.

The clearance of the non-mounted bearing is reduced during mounting due to tight fits of the bearing rings. As a rule, it therefore has to be larger than the operating clearance. The radial clearance is also reduced during operation when the inner ring becomes warmer than the outer ring, which is usually the case.

DIN 620 specifies standard values for the radial clearance of rolling bearings. The normal clearance (clearance group CN) is calculated in such a way that the bearing has an appropriate operating clearance under common mounting and operating conditions.

Normal fits are:

	Shaft	Housing
Ball bearings	j5...k5	H7...J7
Roller bearings	k5...m5	H7...M7

Mounting and service conditions which deviate, such as tight fits for both bearing rings or a temperature difference >10 K, make more radial clearance groups necessary. The suitable clearance group is calculated.

Suffixes for the clearance groups according to DIN 620:

C2 Radial clearance smaller than normal (CN)
C3 Radial clearance larger than normal (CN)
C4 Radial clearance larger than C3

Clearance values of non-mounted bearings are listed from pages 76 to 82 inclusive for the main bearing types. The tables also contain values which are beyond the range set in DIN 620 T4 (edition 08.87).

Reduction of the radial clearance by means of temperature differences

The reduction of the radial clearance Δ_{Grt} by means of temperature differences Δ_t [K] for non-adjusted bearings is approximately:

$$\Delta_{Grt} = \Delta_t \cdot \alpha \cdot (d + D)/2 \text{ [mm]},$$

where

$\alpha = 0.000011\ K^{-1}$	Linear thermal expansion coefficient of steel
d	Bearing bore [mm]
D	Bearing outside diameter [mm]

A greater change in radial clearance can be expected when the bearing position is exposed to the input or dissipation of heat. A smaller radial clearance results from heat input through the shaft or heat dissipation through the housing. A larger radial clearance results from heat input through the housing or heat dissipation through the shaft. Rapid run-up of the bearings to operating speed results in greater differences in temperature between the bearing rings than is the case in a steady state. Either the bearings should be run up slowly or a larger radial clearance than theoretically necessary for the bearing when under operating temperatures should be selected in order to prevent detrimental preload and bearing deformation.

Reduction of radial clearance by means of tight fits

The expansion of the inner ring raceway and the constriction of the outer ring raceway can be assumed to be approximately 80% and 70% of the interference respectively. (Preconditions: solid steel shaft, steel housing with normal wall thickness). Computation programmes are available for more exact calculations, see Section "FAG services programme" on page 685 et seq.

Radial clearance of FAG deep groove ball bearings with cylindrical bore

Dimensions in mm

Nominal bore diameter																	
	over	**2.5**	**6**	**10**	**18**	**24**	**30**	**40**	**50**	**65**	**80**	**100**	**120**	**140**	**160**	**180**	**200**
	to	**6**	**10**	**18**	**24**	**30**	**40**	**50**	**65**	**80**	**100**	**120**	**140**	**160**	**180**	**200**	**225**
Bearing clearance in microns																	
Clearance group C2	min	0	0	0	0	1	1	1	1	1	1	2	2	2	2	2	4
	max	7	7	9	10	11	11	11	15	15	18	20	23	23	25	30	32
Clearance group CN (normal)	min	2	2	3	5	5	6	6	8	10	12	15	18	18	20	25	28
	max	13	13	18	20	20	20	23	28	30	36	41	48	53	61	71	82
Clearance group C3	min	8	8	11	13	13	15	18	23	25	30	36	41	46	53	63	73
	max	23	23	25	28	28	33	36	43	51	58	66	81	91	102	117	132
Clearance group C4	min		14	18	20	23	28	30	38	46	53	61	71	81	91	107	120
	max		29	33	36	41	46	51	61	71	84	97	114	130	147	163	187

Nominal bore diameter																		
	over	**225**	**250**	**280**	**315**	**355**	**400**	**450**	**500**	**560**	**630**	**710**	**800**	**900**	**1000**	**1120**	**1250**	**1400**
	to	**250**	**280**	**315**	**355**	**400**	**450**	**500**	**560**	**630**	**710**	**800**	**900**	**1000**	**1120**	**1250**	**1400**	**1600**
Clearance group C2	min	4	4	8	8	8	10	10	20	20	30	30	30	40	40	40	60	60
	max	36	39	45	50	60	70	80	90	100	120	130	150	160	170	180	210	230
Clearance group CN (normal)	min	31	36	42	50	60	70	80	90	100	120	130	150	160	170	180	210	230
	max	92	97	110	120	140	160	180	200	220	250	280	310	340	370	400	440	480
Clearance group C3	min	87	97	110	120	140	160	180	200	220	250	280	310	340	370	400	440	480
	max	152	162	180	200	230	260	290	320	350	390	440	490	540	590	640	700	770
Clearance group C4	min	140	152	175	200	230	260	290	320	350	390	440	490	540	590	640	700	770
	max	217	237	260	290	330	370	410	460	510	560	620	690	760	840	910	1000	1100

Radial clearance of FAG self-aligning ball bearings

Dimensions in mm

Nominal bore diameter															
	over		**6**	**10**	**14**	**18**	**24**	**30**	**40**	**50**	**65**	**80**	**100**	**120**	**140**
	to	**6**	**10**	**14**	**18**	**24**	**30**	**40**	**50**	**65**	**80**	**100**	**120**	**140**	**160**

with cylindrical bore

Bearing clearance in microns

Clearance group C2	min	1	2	2	3	4	5	6	6	7	8	9	10	10	15
	max	8	9	10	12	14	16	18	19	21	24	27	31	38	44
Clearance group CN (normal)	min	5	6	6	8	10	11	13	14	16	18	22	25	30	35
	max	15	17	19	21	23	24	29	31	36	40	48	56	68	80
Clearance group C3	min	10	12	13	15	17	19	23	25	30	35	42	50	60	70
	max	20	25	26	28	30	35	40	44	50	60	70	83	100	120
Clearance group C4	min	15	19	21	23	25	29	34	37	45	54	64	75	90	110
	max	25	33	35	37	39	46	53	57	69	83	96	114	135	161

with tapered bore

Bearing clearance in microns

Clearance group C2	min					7	9	12	14	18	23	29	35	40	45
	max					17	20	24	27	32	39	47	56	68	74
Clearance group CN (normal)	min					13	15	19	22	27	35	42	50	60	65
	max					26	28	35	39	47	57	68	81	98	110
Clearance group C3	min					20	23	29	33	41	50	62	75	90	100
	max					33	39	46	52	61	75	90	108	130	150
Clearance group C4	min					28	33	40	45	56	69	84	100	120	140
	max					42	50	59	65	80	98	116	139	165	191

Axial clearance of FAG double row angular contact ball bearings of series 32, 32B, 33, 33B

Dimensions in mm

Nominal bore diameter												
	over	**6**	**10**	**18**	**24**	**30**	**40**	**50**	**65**	**80**	**100**	**120**
	to	**10**	**18**	**24**	**30**	**40**	**50**	**65**	**80**	**100**	**120**	**140**
Bearing clearance in microns												
Clearance group C2	min	1	1	2	2	2	2	3	3	3	4	4
	max	11	12	14	15	16	18	22	24	26	30	34
Clearance group CN (normal)	min	5	6	7	8	9	11	13	15	18	22	25
	max	21	23	25	27	29	33	36	40	46	53	59
Clearance group C3	min	12	13	16	18	21	23	26	30	35	42	48
	max	28	31	34	37	40	44	48	54	63	73	82
Clearance group C4	min	25	27	28	30	33	36	40	46	55	65	74
	max	45	47	48	50	54	58	63	71	83	96	108

Axial clearance of FAG double row angular contact ball bearings of series 33DA

Bearing clearance in microns

Clearance group C2	min	5	6	7	8	9	11	13	15	18	22	25
	max	22	24	25	27	29	33	36	40	46	53	59
Clearance group CN (normal)	min	11	13	14	16	18	22	25	29	35	42	48
	max	28	31	32	35	38	44	48	54	63	73	82
Clearance group C3	min	20	23	24	27	30	36	40	46	55	65	74
	max	37	41	42	46	50	58	63	71	83	96	108

Axial clearance of FAG four-point bearings

Dimensions in mm / Bearing clearance in microns

Nominal bore diameter																					
	over		18	40	60	80	100	140	180	220	260	300	355	400	450	500	560	630	710	800	900
	to	18	40	60	80	100	140	180	220	260	300	355	400	450	500	560	630	710	800	900	1000
Clearance group C2	min	20	30	40	50	60	70	80	100	120	140	160	180	200	220	240	260	280	300	330	360
	max	60	70	90	100	120	140	160	180	200	220	240	270	290	310	330	360	390	420	460	500
Clearance group CN (normal)	min	50	60	80	90	100	120	140	160	180	200	220	250	270	290	310	340	370	400	440	480
	max	90	110	130	140	160	180	200	220	240	280	300	330	360	390	420	450	490	540	590	630
Clearance group C3	min	80	100	120	130	140	160	180	200	220	260	280	310	340	370	400	430	470	520	570	620
	max	120	150	170	180	200	220	240	260	300	340	360	390	430	470	510	550	590	660	730	780

Radial clearance of single row and double row FAG cylindrical roller bearings

Dimensions in mm

with cylindrical bore

Bearing clearance in microns

Nominal bore diameter															
	over		24	30	40	50	65	80	100	120	140	160	180	200	225
	to	24	30	40	50	65	80	100	120	140	160	180	200	225	250
Clearance group C1NA[1]	min	5	5	5	5	5	10	10	10	10	10	10	15	15	15
	max	15	15	15	18	20	25	30	30	35	35	40	45	50	50
Clearance group C2	min	0	0	5	5	10	10	15	15	15	20	25	35	45	45
	max	25	25	30	35	40	45	50	55	60	70	75	90	105	110
Clearance group CN (normal)	min	20	20	25	30	40	40	50	50	60	70	75	90	105	110
	max	45	45	50	60	70	75	85	90	105	120	125	145	165	175
Clearance group C3	min	35	35	45	50	60	65	75	85	100	115	120	140	160	170
	max	60	60	70	80	90	100	110	125	145	165	170	195	220	235
Clearance group C4	min	50	50	60	70	80	90	105	125	145	165	170	195	220	235
	max	75	75	85	100	110	125	140	165	190	215	220	250	280	300

Nominal bore diameter																			
	over	250	280	315	355	400	450	500	560	630	710	800	900	1000	1120	1250	1400	1600	1800
	to	280	315	355	400	450	500	560	630	710	800	900	1000	1120	1250	1400	1600	1800	2000
Clearance group C1NA[1]	min	20	20	20	25	25	25	25	30	30	35	35	35	50	60	60	70	80	100
	max	55	60	65	75	85	95	100	110	130	140	160	180	200	220	240	270	300	320
Clearance group C2	min	55	55	65	100	110	110	120	140	145	150	180	200	220	230	270	330	380	400
	max	125	130	145	190	210	220	240	260	285	310	350	390	430	470	530	610	700	760
Clearance group CN (normal)	min	125	130	145	190	210	220	240	260	285	310	350	390	430	470	530	610	700	760
	max	195	205	225	280	310	330	360	380	425	470	520	580	640	710	790	890	1020	1120
Clearance group C3	min	190	200	225	280	310	330	360	380	425	470	520	580	640	710	790	890	1020	1120
	max	260	275	305	370	410	440	480	500	565	630	690	770	850	950	1050	1170	1340	1480
Clearance group C4	min	260	275	305	370	410	440	480	500	565	630	690	770	850	950	1050	1170	1340	1480
	max	330	350	385	460	510	550	600	620	705	790	860	960	1060	1190	1310	1450	1660	1840

with tapered bore

Bearing clearance in microns

Nominal bore diameter															
	over		24	30	40	50	65	80	100	120	140	160	180	200	225
	to	24	30	40	50	65	80	100	120	140	160	180	200	225	250
Clearance group C1NA[1]	min	10	15	15	17	20	25	35	40	45	50	55	60	60	65
	max	20	25	25	30	35	40	55	60	70	75	85	90	95	100
Clearance group C2	min	15	20	20	25	30	35	40	50	55	60	75	85	95	105
	max	40	45	45	55	60	70	75	90	100	110	125	140	155	170
Clearance group CN (normal)	min	30	35	40	45	50	60	70	90	100	110	125	140	155	170
	max	55	60	65	75	80	95	105	130	145	160	175	195	215	235
Clearance group C3	min	40	45	55	60	70	85	95	115	130	145	160	180	200	220
	max	65	70	80	90	100	120	130	155	175	195	210	235	260	285
Clearance group C4	min	50	55	70	75	90	110	120	140	160	180	195	220	245	270
	max	75	80	95	105	120	145	155	180	205	230	245	275	305	335

Nominal bore diameter																			
	over	250	280	315	355	400	450	500	560	630	710	800	900	1000	1120	1250	1400	1600	1800
	to	280	315	355	400	450	500	560	630	710	800	900	1000	1120	1250	1400	1600	1800	2000
Clearance group C1NA[1]	min	75	80	90	100	110	120	130	140	160	170	190	210	230	250	270	300	320	340
	max	110	120	135	150	170	190	210	230	260	290	330	360	400	440	460	500	530	560
Clearance group C2	min	115	130	145	165	185	205	230	260	295	325	370	410	455	490	550	640	700	760
	max	185	205	225	255	285	315	350	380	435	485	540	600	665	730	810	920	1020	1120
Clearance group CN (normal)	min	185	205	225	255	285	315	350	380	435	485	540	600	665	730	810	920	1020	1120
	max	255	280	305	345	385	425	470	500	575	645	710	790	875	970	1070	1200	1340	1480
Clearance group C3	min	240	265	290	330	370	410	455	500	565	630	700	780	865	960	1070	1200	1340	1480
	max	310	340	370	420	470	520	575	620	705	790	870	970	1075	1200	1330	1480	1660	1840
Clearance group C4	min	295	325	355	405	455	505	560	620	695	775	860	960	1065	1200	1330	1480	1660	1840
	max	365	400	435	495	555	615	680	740	835	935	1030	1150	1275	1440	1590	1760	1980	2200

[1]) Single and double row cylindrical roller bearings of the tolerance classes SP and UP have bearing clearance C1NA.

Radial clearance of FAG spherical roller bearings

Dimensions in mm

with cylindrical bore

Bearing clearance in microns

Nominal bore diameter	over	18	24	30	40	50	65	80	100	120	140	160	180	200	225
	to	24	30	40	50	65	80	100	120	140	160	180	200	225	250
Clearance group C2	min	10	15	15	20	20	30	35	40	50	60	65	70	80	90
	max	20	25	30	35	40	50	60	75	95	110	120	130	140	150
Clearance group CN (normal)	min	20	25	30	35	40	50	60	75	95	110	120	130	140	150
	max	35	40	45	55	65	80	100	120	145	170	180	200	220	240
Clearance group C3	min	35	40	45	55	65	80	100	120	145	170	180	200	220	240
	max	45	55	60	75	90	110	135	160	190	220	240	260	290	320
Clearance group C4	min	45	55	60	75	90	110	135	160	190	220	240	260	290	320
	max	60	75	80	100	120	145	180	210	240	280	310	340	380	420

Nominal bore diameter	over	250	280	315	355	400	450	500	560	630	710	800	900	1000	1120	1250	1400
	to	280	315	355	400	450	500	560	630	710	800	900	1000	1120	1250	1400	1600
Clearance group C2	min	100	110	120	130	140	140	150	170	190	210	230	260	290	320	350	380
	max	170	190	200	220	240	260	280	310	350	390	430	480	530	580	630	700
Clearance group CN (normal)	min	170	190	200	220	240	260	280	310	350	390	430	480	530	580	630	700
	max	260	280	310	340	370	410	440	480	530	580	650	710	770	840	910	1020
Clearance group C3	min	260	280	310	340	370	410	440	480	530	580	650	710	770	840	910	1020
	max	350	370	410	450	500	550	600	650	700	770	860	930	1050	1140	1240	1390
Clearance group C4	min	350	370	410	450	500	550	600	650	700	770	860	930	1050	1140	1240	1390
	max	460	500	550	600	660	720	780	850	920	1010	1120	1220	1430	1560	1700	1890

with tapered bore

Bearing clearance in microns

Nominal bore diameter	over	18	24	30	40	50	65	80	100	120	140	160	180	200	225
	to	24	30	40	50	65	80	100	120	140	160	180	200	225	250
Clearance group C2	min	15	20	25	30	40	50	55	65	80	90	100	110	120	140
	max	25	30	35	45	55	70	80	100	120	130	140	160	180	200
Clearance group CN (normal)	min	25	30	35	45	55	70	80	100	120	130	140	160	180	200
	max	35	40	50	60	75	95	110	135	160	180	200	220	250	270
Clearance group C3	min	35	40	50	60	75	95	110	135	160	180	200	220	250	270
	max	45	55	65	80	95	120	140	170	200	230	260	290	320	350
Clearance group C4	min	45	55	65	80	95	120	140	170	200	230	260	290	320	350
	max	60	75	85	100	120	150	180	220	260	300	340	370	410	450

Nominal bore diameter	over	250	280	315	355	400	450	500	560	630	710	800	900	1000	1120	1250	1400
	to	280	315	355	400	450	500	560	630	710	800	900	1000	1120	1250	1400	1600
Clearance group C2	min	150	170	190	210	230	260	290	320	350	390	440	490	540	600	660	740
	max	220	240	270	300	330	370	410	460	510	570	640	710	780	860	940	1060
Clearance group CN (normal)	min	220	240	270	300	330	370	410	460	510	570	640	710	780	860	940	1060
	max	300	330	360	400	440	490	540	600	670	750	840	930	1020	1120	1220	1380
Clearance group C3	min	300	330	360	400	440	490	540	600	670	750	840	930	1020	1120	1220	1380
	max	390	430	470	520	570	630	680	760	850	960	1070	1190	1300	1420	1550	1750
Clearance group C4	min	390	430	470	520	570	630	680	760	850	960	1070	1190	1300	1420	1550	1750
	max	490	540	590	650	720	790	870	980	1090	1220	1370	1520	1650	1800	1960	2200

Radial clearance of FAG barrel roller bearings

Dimensions in mm

Nominal bore diameter																	
	over		**30**	**40**	**50**	**65**	**80**	**100**	**120**	**140**	**160**	**180**	**225**	**250**	**280**	**315**
	to	**30**	**40**	**50**	**65**	**80**	**100**	**120**	**140**	**160**	**180**	**225**	**250**	**280**	**315**	**355**

with cylindrical bore

Bearing clearance in microns

Clearance group C2	min	2	3	3	4	5	7	10	15	20	25	30	35	40	40	45
	max	9	10	13	15	20	25	30	35	40	45	50	55	60	70	75
Clearance group CN (normal)	min	9	10	13	15	20	25	30	35	40	45	50	55	60	70	75
	max	17	20	23	27	35	45	50	55	65	70	75	80	85	100	105
Clearance group C3	min	17	20	23	27	35	45	50	55	65	70	75	80	85	100	105
	max	28	30	35	40	55	65	70	80	95	100	105	110	115	135	140
Clearance group C4	min	28	30	35	40	55	65	70	80	95	100	105	110	115	135	140
	max	40	45	50	55	75	90	95	110	125	130	135	140	145	170	175

with tapered bore

Bearing clearance in microns

Clearance group C2	min	9	10	13	15	20	25	30	35	40	45	50	55	60	70	75
	max	17	20	23	27	35	45	50	55	65	70	75	80	85	100	105
Clearance group CN (normal)	min	17	20	23	27	35	45	50	55	65	70	75	80	85	100	105
	max	28	30	35	40	55	65	70	80	95	100	105	110	115	135	140
Clearance group C3	min	28	30	35	40	55	65	70	80	95	100	105	110	115	135	140
	max	40	45	50	55	75	90	95	110	125	130	135	140	145	170	175
Clearance group C4	min	40	45	50	55	75	90	95	110	125	130	135	140	145	170	175
	max	55	60	65	75	95	120	125	140	155	160	165	170	175	205	210

Bearing materials

The performance of a rolling bearing is highly influenced by the material which is used.

The material of rings and rolling elements for FAG rolling bearings is normally a low-alloy, through-hardening chromium steel of a high degree of cleanliness. For bearings subject to heavy shock loads and reversed bending stresses also casehardening steel is used (supply on request).

In recent years, FAG have been able to increase the load ratings considerably particularly due to the improved quality of rolling bearing steels. Research results and practical experience confirm that bearings of today's standard steel reach the endurance strength under positive lubrication and cleanliness conditions and when loads are not too high.

The bearing rings and rolling elements of the FAG rolling bearings are heat-treated in such a way that they are dimensionally stable to 150 °C as a rule. For higher operating temperatures, special heat treatment is necessary (see section "High temperature suitability", page 86).

Applications in corrosive media require rolling bearing steels with increased resistance to corrosion. Standard bearings of "stainless steel" (according to DIN 17440) carry the prefix S and the suffix W203B (also see page 150: "Deep groove ball bearings of stainless steel"). They have the same main dimensions and load carrying capacity as the bearings of through-hardening rolling bearing steel. In order to maintain the increased resistance to corrosion, the surfaces must not be damaged during mounting or in operation (e.g. by contact corrosion). Please contact the FAG Technical Services for the selection of such bearings.

FAG produce balls of silicon nitride for ceramic hybrid spindle bearings. The ceramic balls are much lighter than steel balls. Centrifugal forces and friction are clearly lower. Hybrid bearings reach top speeds even at grease lubrication, have a long service life and a low operating temperature.

Cage design

Main functions of the cage:

- Separation of rolling elements to keep friction and heat development at a minimum.
- Keeping rolling elements at equal distances for uniform load distribution.
- Retaining rolling elements in separable bearings and in bearings which are swiveled out.
- Guiding rolling elements in the unloaded zone of the bearing.

Rolling bearing cages are subdivided into pressed cages and solid cages.

Pressed cages are usually made of sheet steel but some are made of sheet brass also. When compared with machined cages of metal they are advantageous in that they are lighter in weight. Since a pressed cage does not fill the gap between the inner and outer rings, lubricant easily enters the bearing. It is stored at the cage. As a rule, a pressed cage is only indicated in the bearing code when it is not considered part of the standard design of the bearing.

Solid cages are made of metal, textile laminated phenolic resin, and plastic material. They are indicated in the bearing code.

Machined cages of metal are used when requirements in cage strength are strict and temperatures are high.

Solid cages are also used when lip guidance is required. Lip riding cages for high-speed bearings are frequently made of light material such as light metal or textile laminated phenolic resin so that the forces of inertia remain small.

Solid cages of polyamide 66 are produced by injection moulding. As a rule, cage shapes can be produced by injection moulding, which have particularly high load carrying capacity. The positive effect of polyamide's elasticity and light weight can be seen with shock-type bearing stressing, high acceleration and deceleration rates, and with tilting of the bearing rings against each other. Polyamide cages have very good sliding and emergency running properties.

▼ Examples of rolling bearing cages
Pressed cages of steel: Lug cage (a) and rivet cage (b) for deep groove ball bearings, window-type cage (c) for spherical roller bearings.
Machined brass cages: Riveted machined cage (d) for deep groove ball bearings, brass window-type cage (e) for angular contact ball bearings and machined brass cage with integral crosspiece rivets (f) for cylindrical roller bearings.
Moulded cages made of glass-fibre reinforced polyamide: window-type cage (g) for single-row angular contact ball bearings and window-type cage (h) for cylindrical roller bearings.


a


b


c


d


e


f


g

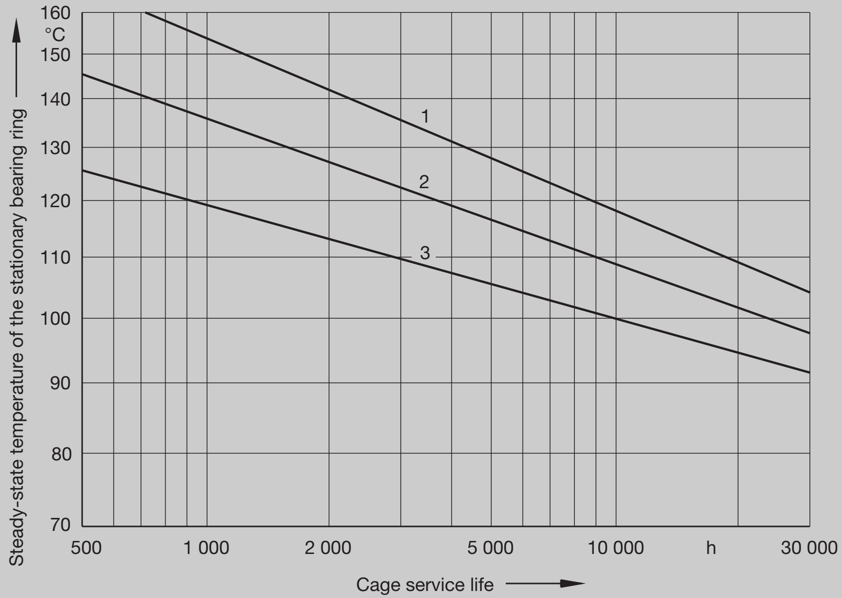

h

Bearing Data
Cages

Cages of glass-fibre reinforced polyamide 66 are suitable for steady-state operating temperatures up to 120 °C. With oil lubrication, additives contained in the oil may lead to a reduction of the cage service life. The diagram shows the relation between the cage service life, the steady-state temperature of the stationary bearing ring and the lubricant.

At higher temperatures, aged oil can also harm the cage service life and attention should be paid to the observance of the oil change intervals.

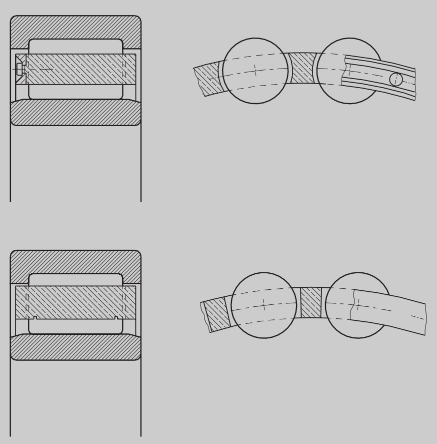
▼ Service life of window-type cages made of polyamide PA66-GF25. The curves apply to steady-state temperature.
If the high temperatures are temporary and not constant, the cage service life is longer.
1 = rolling bearing grease K according to DIN 51825, motor oil or machine lubricating oil, 2 = gear oil, 3 = hypoid oil



Another distinguishing feature of the cages is the **type of guidance**. Most cages are guided by the rolling elements and have no suffix for the type of guidance. Cages guided by the bearing outer ring are given the suffix A. Those guided by the inner ring have the suffix B.

When operating conditions are normal usually the cage design is taken which serves as the standard cage. The standard cages, which can differ within one bearing series according to the bearing size, are described in more detail in the text on the individual dimension tables. Only in the case of special operating conditions must a particularly suitable cage be selected.

▼ Rolling bearing cages are either rolling element riding (upper) or lip riding (lower)



High temperature suitability

FAG rolling bearings with an outside diameter of up to 240 mm are generally heat-treated to retain dimensional stability up to +150 °C. Operating temperatures over +150 °C require special heat treatment. Such bearings are identified by the suffixes S1 to S4 (DIN 623). Exceptions are indicated in the text preceding each tabular section.

Suffix	S1	S2	S3	S4
Maximum-operating temperature	200 °C	250 °C	300 °C	350 °C

FAG bearings with an outside diameter of more than 240 mm are generally dimensionally stable up to 200 °C.

Bearings with cages of glass-fibre reinforced polyamide 66 are suitable for steady-state operating temperatures up to 120 °C. With oil lubrication, additives contained in the oil may lead to a reduction of the cage service life. At higher temperatures, aged oil can also harm the cage service life and attention should be paid to the observance of the oil change intervals, also see page 85.

The permissible temperature of sealed bearings also depends on the requirements on the service life of the grease filling and on the efficiency of the rubbing seal.

Sealed bearings are lubricated with specially tested high-quality lithium soap base greases. These greases withstand +120 °C for a short period. At steady-state temperatures of 70 °C and higher, a reduction of the service life of standard lithium soap base greases must be expected.

Often sufficient service life values can only be attained with special greases. It must also be checked whether seals of heat-resistant materials should be used. + 110 °C is the limit of application of standard rubbing seals.

If high-temperature synthesis materials are used, it has to be taken into account that the very efficient fluorinated materials, when heated above +300 °C, can give off gasses and vapours which are detrimental to health. This has to be remembered especially if bearing parts are dismounted with a welding torch. FAG use fluorinated materials for seals made of fluorocaoutchouc (FKM, FPM, e.g. Viton®) or for fluorinated greases, e.g. Arcanol L79V, an FAG rolling bearing grease. Where high temperatures cannot be avoided the safety data sheet for the fluorinated material in question should be observed. The data sheet is available on request.

High speed suitability

Criteria for the attainable speed

Generally, the maximum attainable speed of rolling bearings is dictated by the permissible operating temperatures. The operating temperature depends on the frictional heat generated within the bearing, possible heat input or heat dissipation from the bearing. Bearing type and size, precision of the bearing and its surrounding parts, clearance, cage design, lubrication, and load influence the attainable speed.

The (**thermal) reference speed** is shown for most bearings in the dimension tables. It is determined by FAG according to the procedure for reference conditions indicated in DIN 732, part 1 (draft).

DIN 732, part 2 (draft), contains a method for determining the **thermally permissible operating speed** for cases where the operating conditions deviate from the reference conditions, e.g. in load, oil viscosity or permissible temperature. Calculations are facilitated by simple diagrams, prepared by FAG, see page 89.

The **limiting speed** which may be higher or lower than the reference speed takes into account only mechanical limits and must be considered as the maximum permissible operating speed.

It should be generally observed that the load is not too low at high speeds and high acceleration rates, see "Minimum rolling bearing load" on page 33.

Limiting speed

Decisive criteria for the limiting speed are mechanical limits e.g. the strength of the bearing parts or the sliding velocity of rubbing seals.

The bearing tables show the limiting speed also for bearings for which the standard does not define a reference speed, e.g. for bearings with rubbing seals. The limiting speed in such cases applies to a load corresponding to $P/C \approx 0.1$, an operating temperature of 70 °C, oil sump lubrication and current mounting conditions.

A limiting speed in the tables, which is lower than the reference speed is indicative of, for example, a limited cage strength. In such cases the higher value must not be used.

The limiting speed may only be exceeded on consultation with FAG.

Reference speed

The reference speed $n_{\Theta r}$ is defined in the draft of DIN 732, part 1, as the speed at which reference temperature is established. There is a balance between frictional energy generated within the bearing and the heat dissipated from the bearings.

The reference conditions are similar to the normal operating conditions of the current rolling bearings. They apply uniformly to all bearing types and sizes. Spindle bearings, four-point bearings, barrel roller bearings, and thrust ball bearings are not included. Reference conditions are selected in such a way that the same reference speeds are obtained for oil lubrication as well as for grease lubrication:

Thermally permissible operating speed

The thermally permissible operating speed n_{zul} is the speed at which the mean bearing temperature reaches the permissible value under realistic operating conditions. It is obtained by multiplying the reference speed $n_{\Theta r}$ with the speed ratio f_N.

$n_{zul} = n_{\Theta r} \cdot f_N$

The determination of f_N is described in DIN 732, part 2 (draft).

The FAG procedure is based on the draft of the standard. Instead of formulas, however, it uses diagrams for radial ball bearings, radial roller bearings, and roller thrust bearings thus facilitating the determination.

The speed ratio f_N is, by approximation, the product of a load parameter f_p, a temperature parameter f_t, and a lubrication parameter f_{v40}.

$f_N = f_p \cdot f_t \cdot f_{v40}$

It must always be checked whether the thermally permissible operating speed does not exceed the limiting speed (see Section "Limiting speed").

Reference conditions

- A reference temperature of 70 °C, measured at the outer ring; a reference ambient temperature of 20 °C
- A reference load of 5 % of the static load rating C_0; pure radial load for radial bearings, centrically acting axial load for thrust bearings
- Lubrication of radial bearings with lithium soap base grease with mineral base oil and no EP additives (base oil viscosity of 22 mm²/s at 70 °C); 30 % of the free bearing cavities filled with grease
- Oil lubrication of radial bearings with current mineral oil without EP additives; kinematic viscosity 12 mm²/s (at 70 °C); oil bath lubrication with oil level reaching up to the middle of the bottom rolling element
- Oil lubrication (oil circulation only) of thrust bearings with current mineral oil without EP additives; kinematic viscosity (at 70 °C) 48 mm²/s for cylindrial roller thrust bearings and 24 mm²/s for spherial roller thrust bearings
- Rolling bearings of normal design, i.e. normal precision, normal bearing clearance, without rubbing seals
- Bearing mounting with stationary outer ring, horizontal shaft, and with the current fits so that the bearings have a normal operating clearance
- Current stress distribution in the rolling bearing, i.e. no impairment by misalignment of the mating structures, by centrifugal forces of the rolling elements, by preload or large operating clearance
- Heat dissipation from the bearing via type-dependent standardized datum surfaces; it serves to calculate the rolling-bearing-specific reference heat flow density for the heat flow which is dissipated via the bearing seat. In the case of thrust bearings with oil circulation lubrication a heat flow is additionally dissipated through the lubricant. A rolling-bearing-specific heat flow density of 20 kW/m² is assumed for cylindrical roller thrust bearings and spherical roller thrust bearings.

Diagrams for load parameters f_p

Load parameters f_p are plotted as a function of the mean bearing diameter $d_m = (D+d)/2$ and P/C_0 values (equivalent dynamic load/static load rating).

Diagram 1 shows the curves for all radial ball bearings, diagram 3 for all radial roller bearings, and diagram 5 for thrust roller bearings.

Diagrams for temperature parameters f_t

The product of the temperature parameter f_t and the previously determined f_p value are obtained from diagrams 2, 4, and 6 (upper parts) for outer ring temperatures between 30 °C and 110 °C.

The diagrams are similar for all bearing types covered by the standard.

Diagrams for lubrication parameters f_{v40}

In the lower part of diagram 2 (radial ball bearings) and of diagram 4 (radial roller bearings) the speed ratio $f_N = f_p \cdot f_t \cdot f_{v40}$ is determined by means of the lubricating parameter f_{v40} for nominal viscosities ν_{40} from 10 to 1500 mm²/s.

Separate curves in the middle and the lower part of diagram 6 take into account that the standard indicates an operating viscosity of ν_{70} = 48 mm²/s (corresponding to a nominal viscosity of ν_{40} = 204 mm²/s) for cylindrical roller thrust bearings and an operating viscosity of ν_{70} = 24 mm²/s (corresponding to a nominal viscosity of ν_{40} = 84 mm²/s) for spherical roller thrust bearings.

In the case of grease lubrication, the base oil viscosity of the grease is used.

For more accurate calculations please use our rolling bearing catalogue on CD-ROM or contact our Technical Service.

Example of how to use the diagrams:

Rolling bearing

Deep groove ball bearing
6216 (80 x 140 x 26 mm)
$d_m = (D + d)/2 = 110$ mm
Reference speed 6300 min⁻¹
Limiting speed 11000 min⁻¹

Load ratio

$P/C_0 = 0.1$

Nominal viscosity

ν_{40} = 36 mm²/s.

Load parameter $f_p = 0.94$

(from diagram 1) with $P/C_0 = 0.1$ for deep groove ball bearings and d_m = 110 mm

Outer ring temperature

t = 90 °C

Product $f_p \cdot f_t = 1.4$

(from upper part of diagram 2) with $f_p = 0.94$ until the intersection with the 90 °C temperature curve

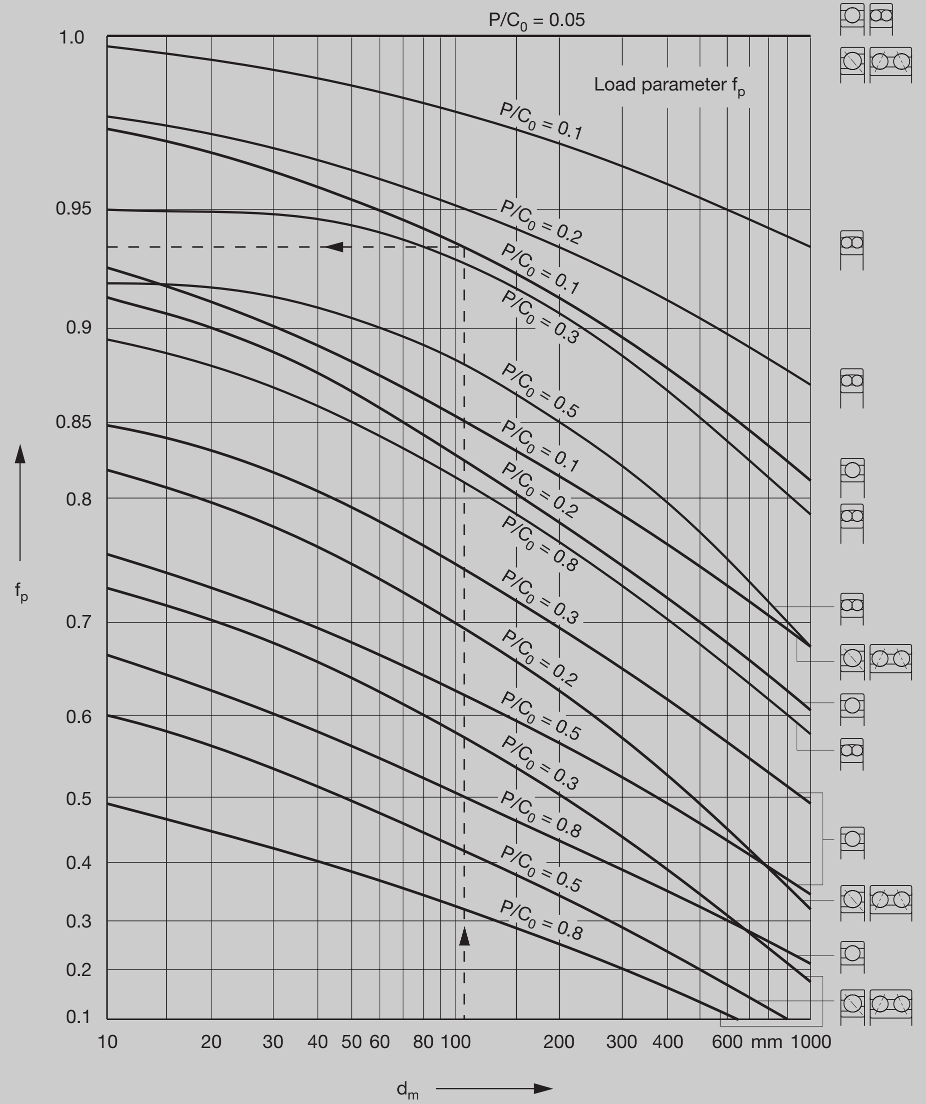
Speed ratio $f_N = 1.4$

(from the lower part of diagram 2) with $f_p \cdot f_t = 1.4$ until the intersection with the curve for lubrication parameter f_{v40} = 36 mm²/s.

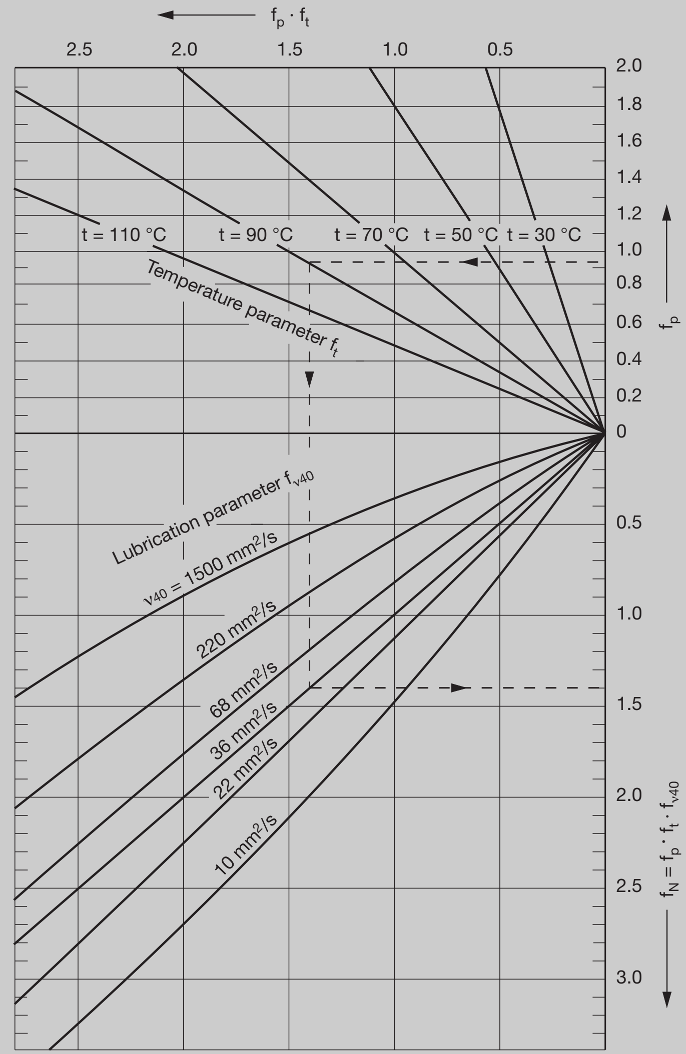
Thermally permissible operating speed

product from f_N and reference speed:
$1.4 \cdot 6300\ \text{min}^{-1} \approx 8800\ \text{min}^{-1}$
which is attainable because it is below the limiting speed (11000 min⁻¹)

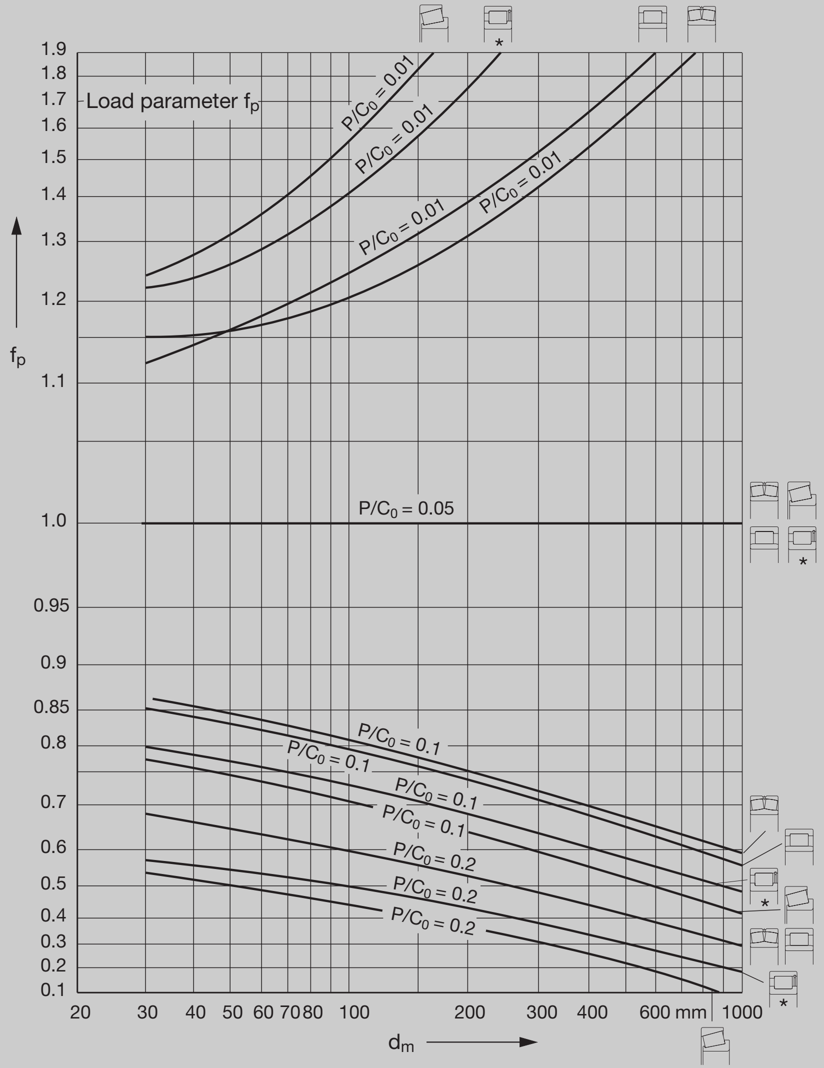
▼ Diagram 1: Load parameter f_p for radial ball bearings for determining the thermally permissible operating speed



▼ Diagram 2: Temperature parameter f_t (upper), lubrication parameter f_{v40} and speed ratio f_N for radial ball bearings for determining the thermally permissible operating speed



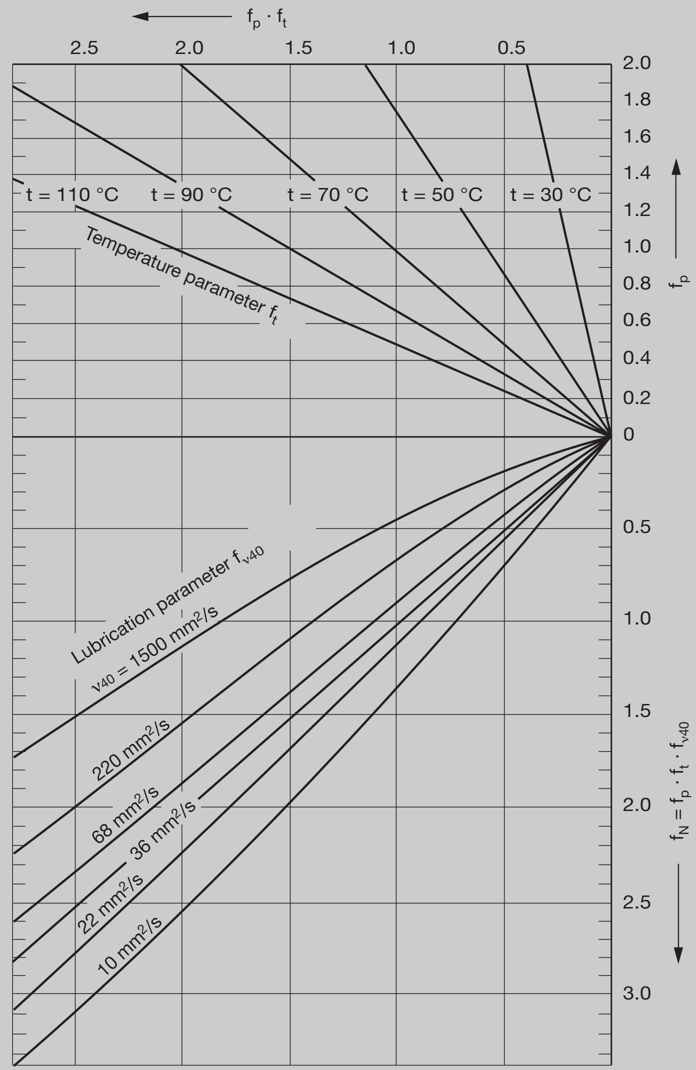
▼ Diagram 3: Load parameter f_p for radial roller bearings for determining the thermally permissible operating speed



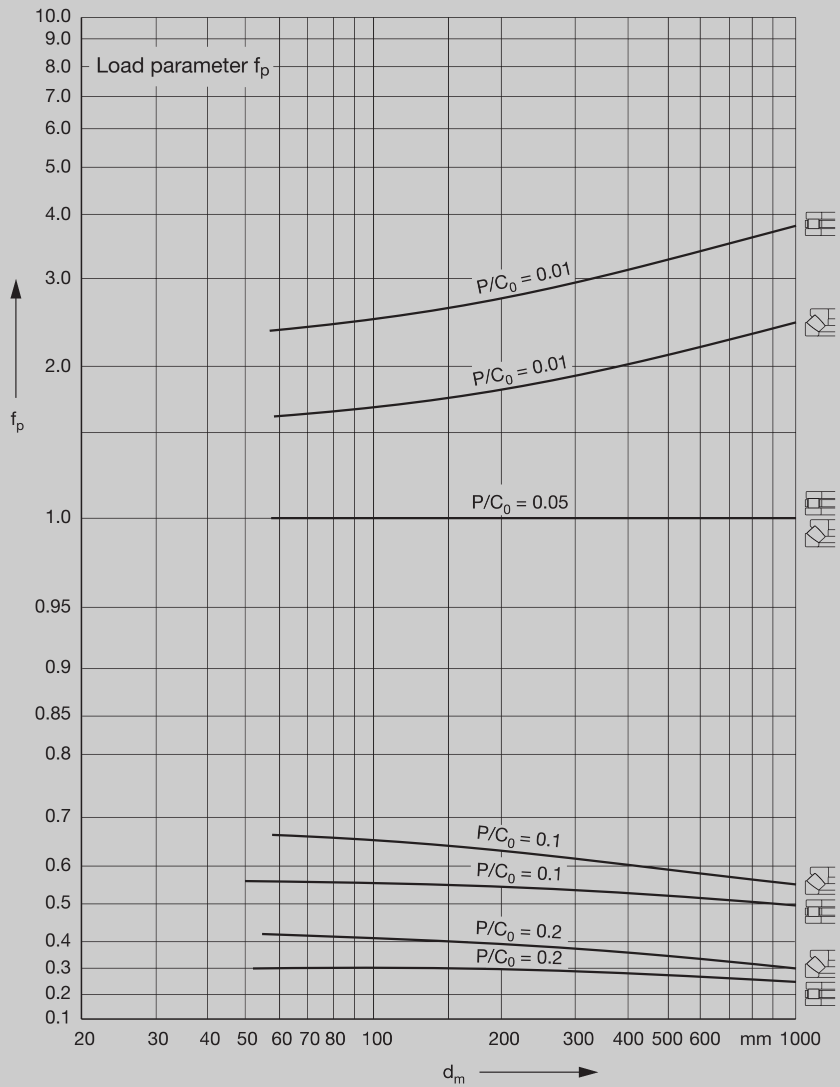
* full-complement cylindrical roller bearings

Bearing Data
High speed suitability

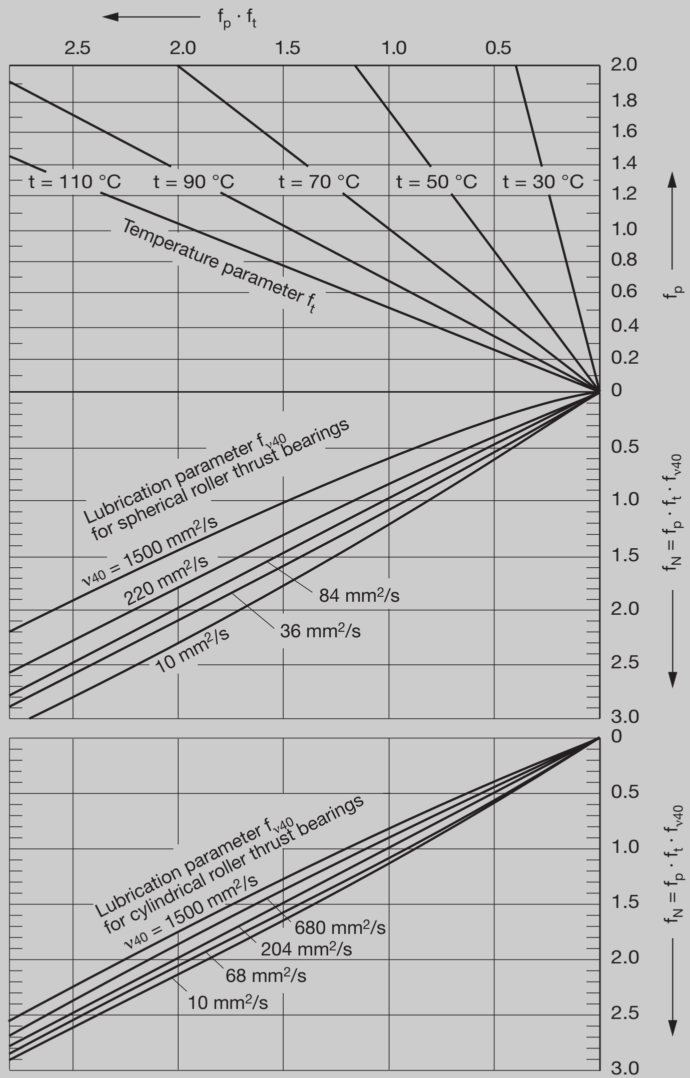
▼ Diagram 4: Temperature parameter f_t (upper), lubrication parameter f_{v40} and speed ratio f_N (lower) for radial roller bearings for determining the thermally permissible operating speed



▼ Diagram 5: Load parameter f_p for thrust roller bearings for determining the thermally permissible operating speed



▼ Diagram 6: Temperature parameter f_t for thrust roller bearings (upper), lubrication parameter $f_{\nu 40}$ and speed ratio f_N for spherical roller thrust bearings (middle) and for cylindrical roller thrust bearings (lower) for determining the thermally permissible operating speed



Friction

The friction in rolling bearings is low. The friction conditions vary, however, in the individual types, since besides the rolling contact friction, there are varying degrees of sliding friction. Lubricant friction is also present. Frictional heat affects the operating temperature of a bearing arrangement.

Rolling contact friction occurs when the rolling elements roll over the raceways; sliding friction occurs at the guiding surfaces of the rolling elements in the cage, at the lip guiding surfaces of the cage and, in roller bearings, at the roller faces and the raceway lips. Lubricant friction is the result of the internal friction of the lubricant between the working surfaces as well as its churning and working action.

▼ Coefficients of friction μ of various rolling bearings at P/C ≈ 0.1 for estimating the frictional moment M

Bearing type	Coefficient of friction μ
Deep groove ball bearings	0.0015
Angular contact ball bearings, single row	0.002
Angular contact ball bearings, double row	0.0024
Four-point bearings	0.0024
Self-aligning ball bearings	0.0013
Cylindrical roller bearings	0.0013
Cylindrical roller bearings, full complement	0.002
Tapered roller bearings	0.0018
Spherical roller bearings	0.002
Thrust ball bearings	0.0015
Cylindrical roller thrust bearings	0.004
Spherical roller thrust bearings	0.002

Frictional moment

The frictional moment M is the bearing's resistance to motion.

Estimation of the frictional moment

Under the conditions

- mean load ($P/C \approx 0.1$)
- viscosity ratio $\varkappa \approx 1$
- mean speed range
- mainly radial load in radial bearings, pure axial load in thrust bearings

the frictional moment M can be approximated by the formula

$$M = \mu \cdot F \cdot d/2$$

where

M	[N mm]	total frictional moment
μ		coefficient of friction (table)
F	[N]	resultant bearing load $F = \sqrt{F_r^2 + F_a^2}$
d	[mm]	bearing bore diameter

The constant coefficients of friction shown in the table cannot be applied to deviating operating conditions (magnitude of load, speed, viscosity). The frictional moment is then calculated as described in the following section.

Calculating the frictional moment

The frictional moment of a bearing depends on the load, the speed and the lubricant viscosity. The frictional moment comprises a load-independent component M_0 and a load-dependent component M_1. With high loads and low speeds a considerable amount of mixed friction can be added to M_0 and M_1. With a separating lubricating film, which develops under normal operating conditions, the entire frictional moment consists only of M_0 and M_1:

$$M = M_0 + M_1 \text{ [N mm]}$$

In calculating the frictional moment of axially loaded cylindrical roller bearings a mixed friction share must be taken into account, see formulas at the end of this section (page 98).

Bearings with a high sliding motion rate, for example full-complement cylindrical roller bearings, tapered roller bearings, spherical roller bearings and thrust bearings, run, after the run-in period, outside the mixed friction range if the following condition is fulfilled:

$$n \cdot \nu/(P/C)^{0.5} \geq 9000$$

n	[min^{-1}]	speed
ν	[mm^2/s]	operating viscosity of the oil or of the grease base oil

The **load-independent component of the frictional moment, M_0**, depends on the operating viscosity ν and the speed n. The operating viscosity is in turn influenced by the bearing friction through the bearing temperature. In addition, the bearing size (d_m) and especially the width of the rolling contact areas have an effect on M_0. M_0 is obtained from

$$M_0 = f_0 \cdot 10^{-7} \cdot (\nu \cdot n)^{2/3} \cdot d_m^3 \text{ [N mm]}$$

where

f_0		index for bearing type and lubrication type (see table)
ν	[mm^2/s]	operating viscosity of the oil or the grease base oil
n	[min^{-1}]	bearing speed
d_m	[mm]	(D + d)/2 mean bearing diameter

▼ Index f_0 for the calculation of M_0 (for oil bath lubrication)

Bearing type and series	Index f_0
Deep groove ball bearings	1.5...2
Single-row angular contact ball bearings	
72	2
73	3
Double-row angular contact ball bearings	
32	3.5
33	6
Four-point bearings	4
Self-aligning ball bearings	
12	1.5
13	2
22	2.5
23	3
Cylindrical roller bearings with cage	
2, 3, 4, 10	2
22	3
23	4
30	2.5
Cylindrical roller bearings, full-complement	
NCF29V	6
NCF30V	7
NNC49V	11
NJ23VH	12
NNF50V	13
Tapered roller bearings	
302, 303, 313	3
329, 320, 322, 323	4.5
330, 331, 332	6
Spherical roller bearings	
213, 222	3.5...4
223, 230, 239	4.5
231, 232	5.5...6
240, 241	6.5...7
Thrust ball bearings	
511, 512, 513, 514	1.5
522, 523	2
Cylindrical roller thrust bearings	
811	3
812	4
Spherical roller thrust bearings	
292E	2.5
293E	3
294E	3.3

The indices f_0 of the table apply to oil bath lubrication where the oil level in the stationary bearing reaches the centre of the bottommost rolling element. Wide series bearings of the same type have larger f_0 values. If radial bearings run on a vertical shaft under radial load, twice the value given in the table has to be assumed; the same applies to a large cooling-oil flow rate or an excessive amount of grease (i.e. more grease than can be displaced laterally).

In the starting phase, the f_0 values of freshly greased bearings resemble those of bearings with oil bath lubrication. After the grease is distributed within the bearing, half the f_0 value from the table has to be assumed. Then it is as the value obtained with oil throwaway lubrication. If the bearing is lubricated with a grease that is appropriate for the application, the frictional moment M_0 is obtained mainly from the internal frictional resistance of the base oil.

The **load-dependent frictional moment component, M_1**, results from the rolling contact friction and the sliding friction at the lips and the guiding areas of the cage. The calculation of M_1 using the index f_1 requires a separating lubricating film in the rolling contact areas ($\varkappa = \nu/\nu_1 \geq 1$).

M_1 is calculated as follows:

$M_1 = f_1 \cdot P_1 \cdot d_m$ [N mm]

where

f_1		index taking into account the magnitude of load, see table
P_1	[N]	load ruling M_1, see table
d_m	[mm]	$(D + d)/2$ mean bearing diameter

The larger the bearings, the smaller the rolling elements in relation to the mean bearing diameter d_m. With these formulas, large-size bearings, especially those with a thin cross-section, feature higher frictional moments M_1 than are actually found in field operation.

When determining the frictional moment of **cylindrical roller bearings which also have to accommodate axial loads**, the axial load-dependent friction moment component M_a has to be added to M_0 and M_1. Consequently,

$M = M_0 + M_1 + M_a$ [N mm]

and

$M_a = f_a \cdot 0.06 \cdot F_a \cdot d_m$ [N mm]

▼ Factors for calculating the load-dependent frictional moment M_1

Bearing type, series	f_1*)	$P_1$¹)
Deep groove ball bearings	$(0.0005...0.0009) \cdot (P_{0*}/C_0)^{0.5}$	F_r or $3.3\ F_a \cdot 0.1\ F_r$ ²)
Angular contact ball bearings		
single row, $\alpha = 15°$	$0.0008\ (P_{0*}/C_0)^{0.5}$	F_r or $3.3\ F_a \cdot 0.1\ F_r$ ²)
single row, $\alpha = 25°$	$0.0009\ (P_{0*}/C_0)^{0.5}$	F_r or $1.9\ F_a \cdot 0.1\ F_r$ ²)
single row, $\alpha = 40°$	$0.001\ (P_{0*}/C_0)^{0.33}$	F_r or $1.0\ F_a \cdot 0.1\ F_r$ ²)
double-row or single-row paired	$0.001\ (P_{0*}/C_0)^{0.33}$	F_r or $1.4\ F_a \cdot 0.1\ F_r$ ²)
Four-point bearings	$0.001\ (P_{0*}/C_0)^{0.33}$	$1.5\ F_a + 3.6\ F_r$
Self-aligning ball bearings	$0.0003\ (P_{0*}/C_0)^{0.4}$	F_r or $1.37\ F_a/e - 0.1\ F_r$ ²)
Cylindrical roller bearings		
with cage	0.0002...0.0004	F_r ³)
full-complement	0.00055	F_r ³)
Tapered roller bearings		
single-row	0.0004	$2\ Y\ F_a$ or F_r ²)
double-row or single-row paired	0.0004	$1.21\ F_a/e$ or F_r ²)
Spherical roller bearings		
Series 213, 222	$0.0005\ (P_{0*}/C_0)^{0.33}$	
Series 223	$0.0008\ (P_{0*}/C_0)^{0.33}$	$1.6\ F_a/e$, if $F_a/F_r > e$
Series 231, 240	$0.0012\ (P_{0*}/C_0)^{0.5}$	
Series 230, 239	$0.00075\ (P_{0*}/C_0)^{0.5}$	$F_r\ \{1 + 0.6\ [F_a/(e \cdot F_r)]^3\}$
Series 232	$0.0016\ (P_{0*}/C_0)^{0.5}$	if $F_a/F_r \leq e$
Series 241	$0.0022\ (P_{0*}/C_0)^{0.5}$	
Thrust ball bearings	$0.0012\ (F_a/C_0)^{0.33}$	F_a
Cylindrical roller thrust bearings	0.0015	F_a
Spherical roller thrust bearings	0.00023...0.00033	F_a (where $F_r \leq 0.55\ F_a$)

*) The higher value applies to the wider series
¹) Where $P_1 < F_r$, use the equation $P_1 = F_r$
²) The higher of the two values is used
³) Only radially loaded. For cylindrical roller bearings that are also subjected to axial loads, M_a has to be added to the frictional moment M_1: $M = M_0 + M_1 + M_a$

Symbols used

P_{0*}	[N]	equivalent load, determined from dynamic loads, see page 41
C_0	[N]	static load rating
F_a	[N]	axial component of the dynamic bearing load
F_r	[N]	radial component of the dynamic bearing load
Y, e		factors explained in the texts preceding the bearing tables

The coefficient f_a which depends on the axial load and the lubricating condition can be taken from the diagram (below).

Using these equations the frictional moment of a bearing can be assessed with adequate accuracy. In field applications, certain deviations are possible if the intended full fluid film lubrication cannot be maintained and mixed friction occurs. The most favourable lubricating condition is not always achieved in operation.

The **breakaway torque** of rolling bearings at machine start-up can be much more than the calculated values, especially at low temperatures and in bearings with rubbing seals.

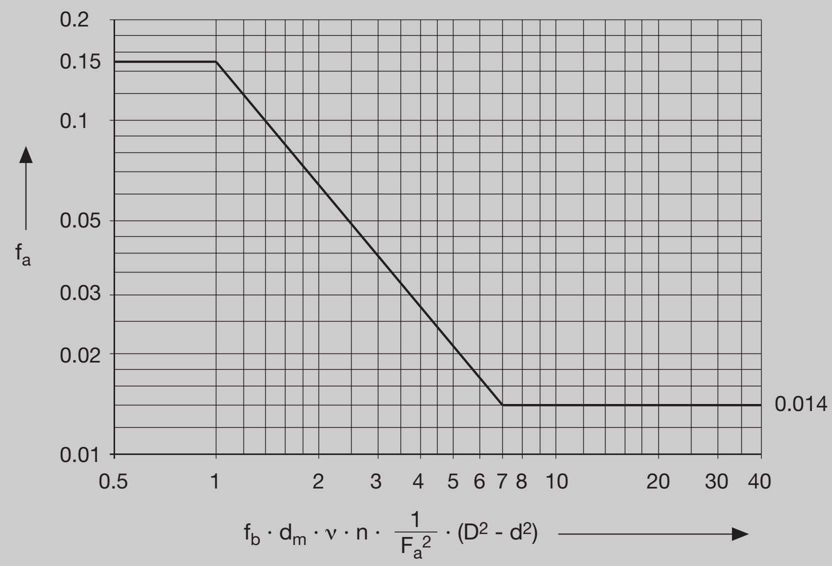
For **bearings with integrated rubbing seals** a considerable supplementary frictional moment component must be considered, in addition to the calculated frictional moment. For small grease-lubricated bearings the supplementary factor can be 8 (e.g. 6201.2RSR with a standard grease after grease distribution), for larger bearings the factor can be 3 (e.g. 6216.2RSR with a standard grease after grease distribution). The frictional moment of the seal also depends on the penetration class of the grease and on the speed.

The frictional moment and the **operating temperature** of rolling bearings can be quickly and easily assessed using the electronic FAG rolling bearing catalogue, also see the Section "FAG services programme". The calculation procedure is described in the FAG publication no. WL 81 115 "Rolling bearing lubrication".

▼ Coefficient of friction f_a for determining the axial load-dependent frictional moment M_a of axially loaded cylindrical roller bearings

The following parameters are required for determining M_a:

f_b	=	0.0048 for bearings with a cage
	=	0.0061 for full-complement bearings (without a cage)
d_m	[mm]	mean bearing diameter = $0.5 \cdot (D + d)$
ν	[mm²/s]	operating viscosity of the oil or grease base oil
n	[min⁻¹]	inner ring speed
F_a	[N]	axial load
D	[mm]	bearing O.D.
d	[mm]	bearing bore



Surrounding structure

Depending on their function rolling bearings must be fixed on the shaft and in the housing in radial, axial, and circumferential direction. Radial and circumferential location is achieved by frictional contact, i.e. the bearing rings are given tight fits. Axial location is usually achieved by positive contact, e.g. by nuts, housing covers, shaft end caps, spacers or snap rings.

Fits, bearing seats

The fit is derived from the ISO tolerances for shaft and housing (ISO 286) together with the tolerances for bore (Δ_{dmp}) and outside diameter (Δ_{Dmp}) of the bearing (DIN 620). The ISO tolerances are in the form of tolerance zones. They are determined by their position to the zero line (= tolerance position) and their size (= tolerance quality, see table on page 102). Tolerance positions are designated by letters (capitals for housings, small letters for shafts). See page 103 for a schematic display of the most commonly used rolling bearing fits.

The following aspects should be taken into account when selecting the fit:

- The bearing rings must be well supported on their circumference so that the load carrying capacity of the bearing is fully utilized.
- The rings should not move on their mating parts, otherwise the seats will be damaged.
- One of the floating bearing rings must adapt to length variations of shaft and housing, which means it is axially displaceable; only with cylindrical roller bearings N and NU does the displacement take place in the bearing.
- Easy mounting and dismounting of bearings must be possible.

With regard to the first two requirements, the inner rings and outer rings of radial bearings should always be given a tight fit. This, however, cannot be realized - at least for one ring - if the floating bearing (cf. "Bearing Arrangement", page 24) has to shift axially or non-separable bearings have to be mounted and dismounted. Whether the ring has point load or circumferential load is then a decisive factor. A loose fit is permissible (shaft to g, housing to G, H, or J) for the ring whose load is constantly directed at the same point (point load). The other ring, however, which rotates relative to the load direction (circumferential load), is generally given a tight fit. See page 104 for an illustration of the load and motion conditions.

Both rings of the cylindrical roller bearings N and NU can be given a tight fit because length compensation takes place in the bearing and because the rings can be mounted separately.

Higher loads, especially shock loads, require a larger interference and the compliance with close form tolerances.

The radial clearance of the bearing decreases with tight fits and a temperature gradient from the inner ring to the outer ring. This should be taken into consideration when selecting the radial clearance group (see "Bearing clearance", page 74).

Design of Surrounding Structure
Fits · Bearing seats

Recommendations for machining bearing seats

The degree of accuracy for the diameter tolerances of the bearing seats on the shaft and in the housing can be found in the tables "Recommendations for machining bearing seats", on page 103, and "ISO basic tolerances", on page 102.

The accuracy degrees for the cylindricity tolerance of the fitting surfaces (t_1 and t_3) and for the axial runout of the abutting shoulders (t_2 and t_4) should be tighter by one IT quality than the accuracy of the pertinent diameter tolerances.

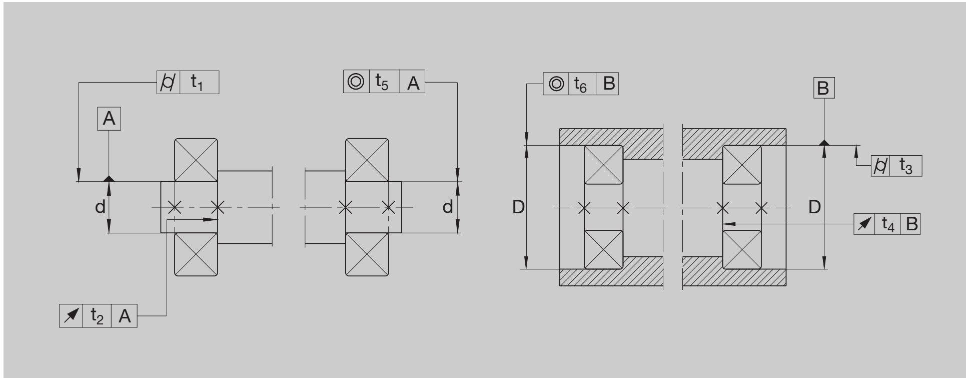
The tolerances of position, t_5 and t_6, for a second bearing seat on the shaft and in the housing - expressed by the coaxiality according to DIN ISO 1101 - must be guided by the angular aligning capability of the bearing (see texts preceding the bearing tables). Misalignment due to elastic deformation of shaft and housing must also be taken into consideration.

In order to attain the tolerances of cylindricity t_1 and t_3, we recommend for the measuring distance (width of bearing seat):

Straightness	$0.8 \cdot t_1$ and $0.8 \cdot t_3$
Circularity	$0.8 \cdot t_1$ and $0.8 \cdot t_3$
Parallelism	$1.6 \cdot t_1$ and $1.6 \cdot t_3$

Bearings with tapered bores are placed directly on the tapered shaft or on adapter or withdrawal sleeves. The tight fit of the inner ring is not determined by the shaft tolerance as with cylindrical bores but by the axial displacement on the tapered seat.

Larger diameter tolerances than for cylindrical seats are permissible for the seats of adapter and withdrawal sleeves; the form tolerances should be closer than the diameter tolerances.



▼ ISO basic tolerances (IT qualities) according to DIN ISO 286

Nominal dimensions in mm

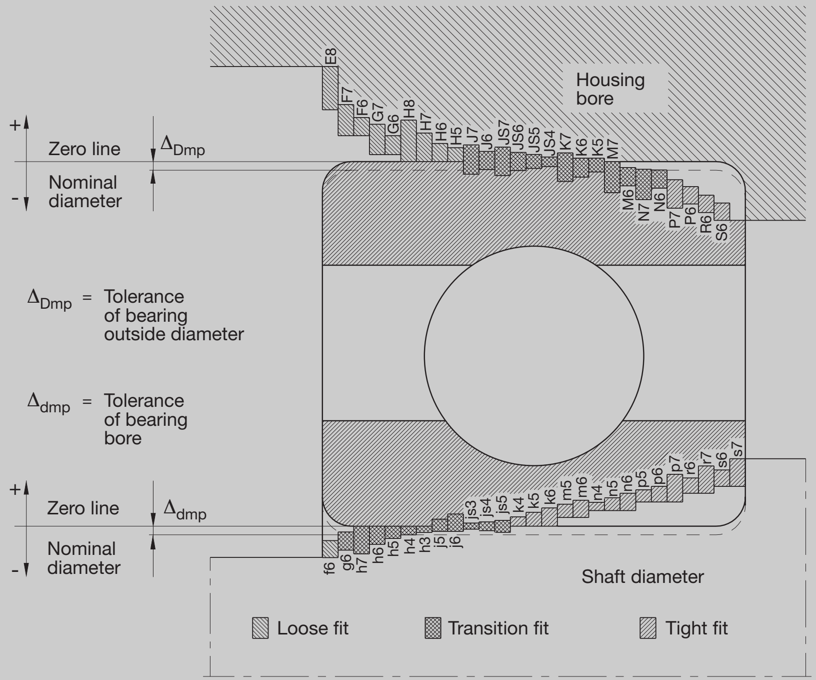
over to	1 3	3 6	6 10	10 18	18 30	30 50	50 80	80 120	120 180	180 250	250 315	315 400	400 500	500 630	630 800	800 1000	1000 1250	1250 1600	1600 2000	2000 2500	2500 3150
	Values in microns																				
IT0	0.5	0.6	0.6	0.8	1	1	1.2	1.5	2	3	4	5	6								
IT1	0.8	1	1	1.2	1.5	1.5	2	2.5	3.5	4.5	6	7	8								
IT2	1.2	1.5	1.5	2	2.5	2.5	3	4	5	7	8	9	10								
IT3	2	2.5	2.5	3	4	4	5	6	8	10	12	13	15								
IT4	3	4	4	5	6	7	8	10	12	14	16	18	20								
IT5	4	5	6	8	9	11	13	15	18	20	23	25	27	29	32	36	42	50	60	70	86
IT6	6	8	9	11	13	16	19	22	25	29	32	36	40	44	50	56	66	78	92	110	135
IT7	10	12	15	18	21	25	30	35	40	46	52	57	63	70	80	90	105	125	150	175	210
IT8	14	18	22	27	33	39	46	54	63	72	81	89	97	110	125	140	165	195	230	280	330
IT9	25	30	36	43	52	62	74	87	100	115	130	140	155	175	200	230	260	310	370	440	540
IT10	40	48	58	70	84	100	120	140	160	185	210	230	250	280	320	360	420	500	600	700	860
IT11	60	75	90	110	130	160	190	220	250	290	320	360	400	440	500	560	660	780	920	1100	1350
IT12	100	120	150	180	210	250	300	350	400	460	520	570	630	700	800	900	1050	1250	1500	1750	2100

Fits for thrust bearing washers

Thrust bearings which only accommodate axial loads, must not be radially guided (exception: cylindrical roller thrust bearings where there is a degree of freedom in the radial direction because of the even raceways). There is no degree of freedom in the case of thrust bearings with groove-shaped raceways, such as thrust ball bearings, and it must be created by means of a slide fit of the stationary washer. A tight fit is generally chosen for rotating washers. When thrust bearings accommodate radial loads as well as axial loads, for example spherical roller thrust bearings, fits are to be selected as for radial bearings.

The abutting surfaces of the mating parts have to be vertical to the rotary axis (axial runout tolerance according to IT5 or better), so that the load is distributed uniformly on all rolling elements.

▼ Principal fits for bearings



▼ Recommendations for machining bearing seats and their roughness

Tolerance class of the bearings	Bearing seat	Machining tolerance	Roughness class
Normal, P6X	Shaft	IT6 (IT5)	N5...N7
	Housing	IT7 (IT6)	N6...N8
P5	Shaft	IT5	N5...N7
	Housing	IT6	N6...N8
P4, P4S, SP	Shaft	IT4	N4...N6
	Housing	IT5	N5...N7
UP	Shaft	IT3	N3...N5
	Housing	IT4	N4...N6

The higher roughness classes are selected for larger diameters.

Roughness of the bearing seats

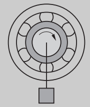
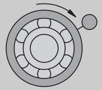
The roughness of the bearing seats must match the tolerance class of the bearings. The average roughness value R_a should not be too large so that the interference loss remains within limits. The recommended roughness values correspond to DIN 5425, edition 11.84.

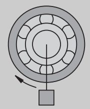
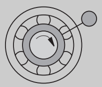
▼ Roughness classes according to DIN ISO 1302

Roughness class	N3	N4	N5	N6	N7	N8	N9	N10
	Values in microns							
Average roughness value R_a	0.1	0.2	0.4	0.8	1.6	3.2	6.3	12.5
Roughness depth $R_z \approx R_t$	1	1.6	2.5	6.3	10	25	40	63

Fits · Bearing seats

▼ Differences between circumferential load and point load

Bearing motions	Example	Illustration	Loading conditions	Fits
Rotating inner ring Stationary outer ring Constant load direction	Weight suspended by the shaft	 Weight	Circumferential load on inner ring and	Inner ring: tight fit mandatory
Stationary inner ring Rotating outer ring Direction of load rotating with outer ring	Large imbalance rotating with outer ring	 Imbalance	Point load on outer ring	Outer ring: slide fit permissible

Bearing motions	Example	Illustration	Loading conditions	Fits
Stationary inner ring Rotating outer ring Constant load direction	Automotive front wheel bearing (hub mounting) Conveyor idler	 Weight	Point load on inner ring and	Inner ring: slide fit permissible
Rotating inner ring Stationary outer ring Direction of load rotating with inner ring	Centrifuge Vibrating screen	 Imbalance	Circumferential load on outer ring	Outer ring: tight fit mandatory

Tables for tolerances and fits

Recommendations for the shaft and housing tolerances are shown on pages 105 and 114.

Figures for fits (tables see pages 106 to 120) apply to solid steel shafts and cast housings. At the top of the tables the normal tolerances for either the bore diameters or the outside diameters are just below the nominal diameters of the radial bearings (excluding tapered roller bearings). Below are the deviations of the chief tolerance zones for rolling bearing mountings.

There are five numbers in each box as follows:

Maximum material	+6	**18**	Interference or clearance when upper shaft deviations coincide with lower bore deviations
Shaft dia 40 j5		**10**	Probable interference or clearance
Minimum material	–5	5	Interference or clearance when lower shaft deviations coincide with upper bore deviations

Numbers **printed in boldface** identify interference.
Standard-type numbers in right column identify clearance.

The probable interference or clearance is assumed to be one third away from the maximum material end of the tolerance zone.

Shaft tolerances

Cylindrical bore radial bearings

Type of load	Bearing type	Shaft diameter	Axial displaceability Load	Tolerance
Point load on inner ring	Ball bearings Roller bearings	all sizes	Floating bearings with sliding inner ring	g6 (g5)
			Angular contact ball bearings and tapered roller bearings with adjusted inner ring	h6 (j6)
Circumferential load on inner ring or indeterminate load	Ball bearings	up to 40 mm	normal load	j6 (j5)
		up to 100 mm	low load	j6 (j5)
			normal and high load	k6 (k5)
		up to 200 mm	low load	k6 (k5)
			normal and high load	m6 (m5)
		over 200 mm	normal load	m6 (m5)
			high load, shocks	n6 (n5)
	Roller bearings	up to 60 mm	low load	j6 (j5)
			normal and high load	k6 (k5)
		up to 200 mm	low load	k6 (k5)
			normal load	m6 (m5)
			high load	n6 (n5)
		up to 500 mm	normal load	m6 (n6)
			high load, shocks	p6
		over 500 mm	normal load	n6 (p6)
			high load	p6

Thrust bearings

Type of load	Bearing type	Shaft diameter	Operating conditions	Tolerances
Axial load	Thrust ball bearings	all sizes		j6
	Thrust ball bearings double direction	all sizes		k6
	Cylindrical roller thrust bearings with shaft washer	all sizes		h6 (j6)
	Thrust cylindrical roller and cage assemblies	all sizes		h8
Combined load	Spherical roller thrust bearings	all sizes	Point load on shaft washer	j6
		up to 200 mm	Circumferential load on shaft washer	j6 (k6)
		over 200 mm		k6 (m6)

Dimensions in mm

Tolerance in microns (0.001 mm) (normal tolerance)

Shaft tolerance, interference or clearance in microns (0.001 mm)

Nominal shaft dimension over to	3 6		6 10		10 18		18 30		30 50		50 65	
Bearing bore diameter deviation Δ_{dmp}	0 −8		0 −8		0 −8		0 −10		0 −12		0 −15	
Diagram of fit Shaft Δ_{dmp} - 0 +												
f6	−10 −18	2 8 18	−13 −22	5 11 22	−16 −27	8 15 27	−20 −33	10 17 33	−25 −41	13 22 41	−30 −49	15 26 49
g5	−4 −9	**4** **0** 9	−5 −11	**3** 2 11	−6 −14	**2** 3 14	−7 −16	**3** 3 16	−9 −20	**3** 5 20	−10 −23	**5** 4 23
g6	−4 −12	**4** 1 12	−5 −14	**3** 3 14	−6 −17	**2** 4 17	−7 −20	**3** 5 20	−9 −25	**3** 6 25	−10 −29	**5** 6 29
h5	0 −5	**8** **4** 5	0 −6	**8** **3** 6	0 −8	**8** **3** 8	0 −9	**10** **4** 9	0 −11	**12** **4** 11	0 −13	**15** **6** 13
h6	0 −8	**8** **3** 8	0 −9	**8** **2** 9	0 −11	**8** **2** 11	0 −13	**10** **2** 13	0 −16	**12** **3** 16	0 −19	**15** **4** 19
j5	+3 −2	**11** **7** 2	+4 −2	**12** **7** 2	+5 −3	**13** **8** 3	+5 −4	**15** **9** 4	+6 −5	**18** **10** 5	+6 −7	**21** **12** 7
j6	+6 −2	**14** **8** 2	+7 −2	**15** **9** 2	+8 −3	**16** **10** 3	+9 −4	**19** **11** 4	+11 −5	**23** **14** 5	+12 −7	**27** **16** 7
js5	+2.5 −2.5	**11** **6** 3	+3 −3	**11** **6** 3	+4 −4	**12** **6** 4	+4.5 −4.5	**15** **9** 5	+5.5 −5.5	**18** **10** 6	+6.5 −6.5	**22** **13** 7
js6	+4 −4	**12** **7** 4	+4.5 −4.5	**13** **7** 5	+5.5 −5.5	**14** **8** 6	+6.5 −6.5	**17** **9** 7	+8 −8	**20** **11** 8	+9.5 −9.5	**25** **13** 10
k5	+6 +1	**14** **9** **1**	+7 +1	**15** **10** **1**	+9 +1	**17** **12** **1**	+11 +2	**21** **15** **2**	+13 +2	**25** **17** **2**	+15 +2	**30** **21** **2**
k6	+9 +1	**17** **11** **1**	+10 +1	**18** **12** **1**	+12 +1	**20** **14** **1**	+15 +2	**25** **17** **2**	+18 +2	**30** **21** **2**	+21 +2	**36** **25** **2**
m5	+9 +4	**17** **13** **4**	+12 +6	**20** **15** **6**	+15 +7	**23** **18** **7**	+17 +8	**27** **21** **8**	+20 +9	**32** **24** **9**	+24 +11	**39** **30** **11**
m6	+12 +4	**20** **15** **4**	+15 +6	**23** **17** **6**	+18 +7	**26** **20** **7**	+21 +8	**31** **23** **8**	+25 +9	**37** **27** **9**	+30 +11	**45** **34** **11**

Nominal shaft dimension over to	65 80		80 100		100 120		120 140		140 160		160 180		180 200		200 225		225 250	
Bearing bore diameter deviation Δ_{dmp}	0 −15		0 −20		0 −20		0 −25		0 −25		0 −25		0 −30		0 −30		0 −30	
f6	−30 −49	15 26 49	−36 −58	16 30 58	−36 −58	16 30 58	−43 −68	18 34 68	−43 −68	18 34 68	−43 −68	18 34 68	−50 −79	20 40 79	−50 −79	20 40 79	−50 −79	20 40 79
g5	−10 −23	**5** 4 23	−12 −27	**8** 4 27	−12 −27	**8** 4 27	−14 −32	**11** 3 32	−14 −32	**11** 3 32	−14 −32	**11** 3 32	−15 −35	**15** 2 35	−15 −35	**15** 2 35	−15 −35	**15** 2 35
g6	−10 −29	**5** 6 29	−12 −34	**8** 6 34	−12 −34	**8** 6 34	−14 −39	**11** 6 39	−14 −39	**11** 6 39	−14 −39	**11** 6 39	−15 −44	**15** 5 44	−15 −44	**15** 5 44	−15 −44	**15** 5 44
h5	0 −13	**15** **6** 13	0 −15	**20** **8** 15	0 −15	**20** **8** 15	0 −18	**25** **11** 18	0 −18	**25** **11** 18	0 −18	**25** **11** 18	0 −20	**30** **13** 20	0 −20	**30** **13** 20	0 −20	**30** **13** 20
h6	0 −19	**15** **4** 19	0 −22	**20** **6** 22	0 −22	**20** **6** 22	0 −25	**25** **8** 25	0 −25	**25** **8** 25	0 −25	**25** **8** 25	0 −29	**30** **10** 29	0 −29	**30** **10** 29	0 −29	**30** **10** 29
j5	+6 −7	**21** **12** 7	+6 −9	**26** **14** 9	+6 −9	**26** **14** 9	+7 −11	**32** **18** 11	+7 −11	**32** **18** 11	+7 −11	**32** **18** 11	+7 −13	**37** **20** 13	+7 −13	**37** **20** 13	+7 −13	**37** **20** 13
j6	+12 −7	**27** **16** 7	+13 −9	**33** **19** 9	+13 −9	**33** **19** 9	+14 −11	**39** **22** 11	+14 −11	**39** **22** 11	+14 −11	**39** **22** 11	+16 −13	**46** **26** 13	+16 −13	**46** **26** 13	+16 −13	**46** **26** 13
js5	+6.5 −6.5	**22** **13** 7	+7.5 −7.5	**28** **16** 8	+7.5 −7.5	**28** **16** 8	+9 −9	**34** **20** 9	+9 −9	**34** **20** 9	+9 −9	**34** **20** 9	+10 −10	**40** **23** 10	+10 −10	**40** **23** 10	+10 −10	**40** **23** 10
js6	+9.5 −9.5	**25** **13** 10	+11 −11	**31** **17** 11	+11 −11	**31** **17** 11	+12.5 −12.5	**38** **21** 13	+12.5 −12.5	**38** **21** 13	+12.5 −12.5	**38** **21** 13	+14.5 −14.5	**45** **25** 15	+14.5 −14.5	**45** **25** 15	+14.5 −14.5	**45** **25** 15
k5	+15 +2	**30** **21** **2**	+18 +3	**38** **26** **3**	+18 +3	**38** **26** **3**	+21 +3	**46** **32** **3**	+21 +3	**46** **32** **3**	+21 +3	**46** **32** **3**	+24 +4	**54** **37** **4**	+24 +4	**54** **37** **4**	+24 +4	**54** **37** **4**
k6	+21 +2	**36** **25** **2**	+25 +3	**45** **31** **3**	+25 +3	**45** **31** **3**	+28 +3	**53** **36** **3**	+28 +3	**53** **36** **3**	+28 +3	**53** **36** **3**	+33 +4	**63** **43** **4**	+33 +4	**63** **43** **4**	+33 +4	**63** **43** **4**
m5	+24 +11	**39** **30** **11**	+28 +13	**48** **36** **13**	+28 +13	**48** **36** **13**	+33 +15	**58** **44** **15**	+33 +15	**58** **44** **15**	+33 +15	**58** **44** **15**	+37 +17	**67** **50** **17**	+37 +17	**67** **50** **17**	+37 +17	**67** **50** **17**
m6	+30 +11	**45** **34** **11**	+35 +13	**55** **42** **13**	+35 +13	**55** **42** **13**	+40 +15	**65** **48** **15**	+40 +15	**65** **48** **15**	+40 +15	**65** **48** **15**	+46 +17	**76** **56** **17**	+46 +17	**76** **56** **17**	+46 +17	**76** **56** **17**

Example: Shaft dia 40 j5

Maximum material	+6	**18**	Interference or clearance when upper shaft deviations coincide with lower bore deviations
		10	Probable interference or clearance
Minimum material	−5	5	Interference or clearance when lower shaft deviations coincide with upper bore deviations

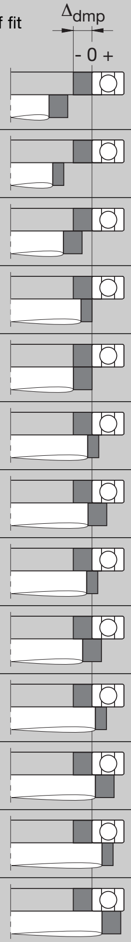
Numbers in **boldface print** identify interference.
Standard-type numbers in right column identify clearance.

Design of Surrounding Structure
Shaft fits

Dimensions in mm

Tolerance in microns (0.001 mm) (normal tolerance)

Shaft tolerance, interference or clearance in microns (0.001 mm)



Nominal shaft dimension over to	250 280		280 315		315 355		355 400		400 450		450 500	
Bearing bore diameter deviation Δ_{dmp}	0 −35		0 −35		0 −40		0 −40		0 −45		0 −45	
Diagram of fit Shaft												
f6	−56 −88	21 44 88	−56 −88	21 44 88	−62 −98	22 47 98	−62 −98	22 47 98	−68 −108	23 51 108	−68 −108	23 51 108
g5	−17 −40	**18** 1 40	−17 −40	**18** 1 40	−18 −43	**22** **0** 43	−18 −43	**22** **0** 43	−20 −47	**25** **1** 47	−20 −47	**25** **1** 47
g6	−17 −49	**18** 4 49	−17 −49	**18** 4 49	−18 −54	**22** 3 54	−18 −54	**22** 3 54	−20 −60	**25** 3 60	−20 −60	**25** 3 60
h5	0 −23	**35** **16** 23	0 −23	**35** **16** 23	0 −25	**40** **18** 25	0 −25	**40** **18** 25	0 −27	**45** **21** 27	0 −27	**45** **21** 27
h6	0 −32	**35** **13** 32	0 −32	**35** **13** 32	0 −36	**40** **15** 36	0 −36	**40** **15** 36	0 −40	**45** **17** 40	0 −40	**45** **17** 40
j5	+7 −16	**42** **23** 16	+7 −16	**42** **23** 16	+7 −18	**47** **25** 18	+7 −18	**47** **25** 18	+7 −20	**52** **28** 20	+7 −20	**52** **28** 20
j6	+16 −16	**51** **29** 16	+16 −16	**51** **29** 16	+18 −18	**58** **33** 18	+18 −18	**58** **33** 18	+20 −20	**65** **37** 20	+20 −20	**65** **37** 20
js5	+11.5 −11.5	**47** **27** 12	+11.5 −11.5	**47** **27** 12	+12.5 −12.5	**53** **32** 13	+12.5 −12.5	**53** **32** 13	+13.5 −13.5	**59** **35** 14	+13.5 −13.5	**59** **35** 14
js6	+16 −16	**51** **29** 16	+16 −16	**51** **29** 16	+18 −18	**58** **33** 18	+18 −18	**58** **33** 18	+20 −20	**65** **37** 20	+20 −20	**65** **37** 20
k5	+27 +4	**62** **43** **4**	+27 +4	**62** **43** **4**	+29 +4	**69** **47** **4**	+29 +4	**69** **47** **4**	+32 +5	**77** **53** **5**	+32 +5	**77** **53** **5**
k6	+36 +4	**71** **49** **4**	+36 +4	**71** **49** **4**	+40 +4	**80** **55** **4**	+40 +4	**80** **55** **4**	+45 +5	**90** **62** **5**	+45 +5	**90** **62** **5**
m5	+43 +20	**78** **59** **20**	+43 +20	**78** **59** **20**	+46 +21	**86** **64** **21**	+46 +21	**86** **64** **21**	+50 +23	**95** **71** **23**	+50 +23	**95** **71** **23**
m6	+52 +20	**87** **65** **20**	+52 +20	**87** **65** **20**	+57 +21	**97** **72** **21**	+57 +21	**97** **72** **21**	+63 +23	**108** **80** **23**	+63 +23	**108** **80** **23**

Nominal shaft dimension over to	500 560		560 630		630 710		710 800		800 900		900 1000		1000 1120		1120 1250		1250 1600	
Bearing bore diameter deviation Δ_{dmp}	0 −50		0 −50		0 −75		0 −75		0 −100		0 −100		0 −125		0 −125		0 −160	
f6	−76 −120	26 58 120	−76 −120	26 58 120	−80 −130	5 47 130	−80 −130	5 47 130	−86 −146	**14** 39 146	−86 −146	**14** 39 146	−98 −164	**27** 38 164	−98 −164	**27** 38 164	−110 −188	**50** 29 188
g5	−22 −51	**28** **1** 51	−22 −51	**28** **1** 51	−24 −56	**51** **15** 56	−24 −56	**51** **15** 56	−26 −62	**74** **29** 62	−26 −62	**74** **29** 62	−28 −70	**97** **41** 70	−28 −70	**97** **41** 70	−30 −80	**130** **60** 80
g6	−22 −66	**28** 4 66	−22 −66	**28** 4 66	−24 −74	**51** **9** 74	−24 −74	**51** **9** 74	−26 −82	**74** **24** 82	−26 −82	**74** **24** 82	−28 −94	**97** **33** 94	−28 −94	**97** **33** 94	−30 −108	**130** **41** 108
h5	0 −29	**50** **23** 29	0 −29	**50** **23** 29	0 −32	**75** **39** 32	0 −32	**75** **39** 32	0 −36	**100** **55** 36	0 −36	**100** **55** 36	0 −42	**125** **69** 42	0 −42	**125** **69** 42	0 −50	**160** **90** 50
h6	0 −44	**50** **18** 44	0 −44	**50** **18** 44	0 −50	**75** **33** 50	0 −50	**75** **33** 50	0 −56	**100** **48** 56	0 −56	**100** **48** 56	0 −66	**125** **61** 66	0 −66	**125** **61** 66	0 −78	**160** **81** 78
j5																		
j6	+22 −22	**72** **40** 22	+22 −22	**72** **40** 22	+25 −25	**100** **58** 25	+25 −25	**100** **58** 25	+28 −28	**128** **76** 28	+28 −28	**128** **76** 28	+33 −33	**158** **94** 33	+33 −33	**158** **94** 33	+39 −39	**199** **120** 39
js5	+14.5 −14.5	**65** **38** 15	+14.5 −14.5	**65** **38** 15	+16 −16	**91** **55** 16	+16 −16	**91** **55** 16	+18 −18	**118** **73** 18	+18 −18	**118** **73** 18	+21 −21	**146** **90** 21	+21 −21	**146** **90** 21	+25 −25	**185** **115** 25
js6	+22 −22	**72** **40** 22	+22 −22	**72** **40** 22	+25 −25	**100** **58** 25	+25 −25	**100** **58** 25	+28 −28	**128** **76** 28	+28 −28	**128** **76** 28	+33 −33	**158** **94** 33	+33 −33	**158** **94** 33	+39 −39	**199** **120** 39
k5	+29 0	**79** **53** **0**	+29 0	**79** **53** **0**	+32 0	**107** **71** **0**	+32 0	**107** **71** **0**	+36 0	**136** **91** **0**	+36 0	**136** **91** **0**	+42 0	**167** **111** **0**	+42 0	**167** **111** **0**	+50 0	**210** **140** **0**
k6	+44 0	**94** **62** **0**	+44 0	**94** **62** **0**	+50 0	**125** **83** **0**	+50 0	**125** **83** **0**	+56 0	**156** **104** **0**	+56 0	**156** **104** **0**	+66 0	**191** **127** **0**	+66 0	**191** **127** **0**	+78 0	**238** **159** **0**
m5	+55 +26	**105** **78** **26**	+55 +26	**105** **78** **26**	+62 +30	**137** **101** **30**	+62 +30	**137** **101** **30**	+70 +34	**170** **125** **34**	+70 +34	**170** **125** **34**	+82 +40	**207** **151** **40**	+82 +40	**207** **151** **40**	+98 +48	**258** **188** **48**
m6	+70 +26	**120** **88** **26**	+70 +26	**120** **88** **26**	+80 +30	**155** **113** **30**	+80 +30	**155** **113** **30**	+90 +34	**190** **138** **34**	+90 +34	**190** **138** **34**	+106 +40	**231** **167** **40**	+106 +40	**231** **167** **40**	+126 +48	**286** **207** **48**

Example: Shaft dia 560 m6

Maximum material	+70	**120**	Interference or clearance when upper shaft deviations coincide with lower bore deviations
		88	Probable interference or clearance
Minimum material	+26	**26**	Interference or clearance when lower shaft deviations coincide with upper bore deviations

Numbers in **boldface print** identify interference.
Standard-type numbers in right column identify clearance.

Design of Surrounding Structure
Shaft fits

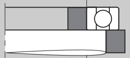
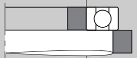
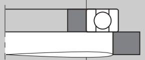
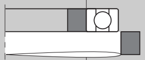
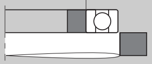
Dimensions in mm

Nominal shaft dimension		over to	3 6	6 10	10 18	18 30	30 50	50 65	65 80	80 100	100 120	120 140	140 160	160 180	180 200	200 225	225 250
Tolerance in microns (0.001 mm) (normal tolerance)																	
Bearing bore diameter deviation		Δ_{dmp}	0 –8	0 –8	0 –8	0 –10	0 –12	0 –15	0 –15	0 –20	0 –20	0 –25	0 –25	0 –25	0 –30	0 –30	0 –30
Diagram of fit Shaft	Δ_{dmp} - 0 +		Shaft tolerance, interference or clearance in microns (0.001 mm)														
n5			+13 +8 **21** **17** **8**	+16 +10 **24** **19** **10**	+20 +12 **28** **23** **12**	+24 +15 **34** **28** **15**	+28 +17 **40** **32** **17**	+33 +20 **48** **39** **20**	+33 +20	+38 +23	+38 +23	+45 +27	+45 +27	+45 +27	+51 +31	+51 +31	+51 +31
n6			+16 +8 **24** **19** **8**	+19 +10 **27** **21** **10**	+23 +12 **31** **25** **12**	+28 +15 **38** **30** **15**	+33 +17 **45** **36** **17**	+39 +20 **54** **43** **20**	+39 +20	+45 +23	+45 +23	+52 +27	+52 +27	+52 +27	+60 +31	+60 +31	+60 +31
p6			+20 +12 **28** **23** **12**	+24 +15 **32** **26** **15**	+29 +18 **37** **31** **18**	+35 +22 **45** **37** **22**	+42 +26 **54** **45** **26**	+51 +32 **66** **55** **32**	+51 +32	+59 +37	+59 +37	+68 +43	+68 +43	+68 +43	+79 +50	+79 +50	+79 +50
p7			+24 +12 **32** **25** **12**	+30 +15 **38** **30** **15**	+36 +18 **44** **35** **18**	+43 +22 **53** **43** **22**	+51 +26 **63** **51** **26**	+62 +32 **77** **62** **32**	+62 +32	+72 +37	+72 +37	+83 +43	+83 +43	+83 +43	+96 +50	+96 +50	+96 +50
r6			+23 +15 **31** **25** **15**	+28 +19 **36** **30** **19**	+34 +23 **42** **35** **23**	+41 +28 **51** **44** **28**	+50 +34 **62** **53** **34**	+60 +41 **75** **64** **41**	+62 +43	+73 +51	+76 +54	+88 +63	+90 +65	+93 +68	+106 +77	+109 +80	+113 +84
r7			+27 +15 **35** **28** **15**	+34 +19 **42** **34** **19**	+41 +23 **49** **40** **23**	+49 +28 **59** **49** **28**	+59 +34 **71** **59** **34**	+71 +41 **86** **71** **41**	+73 +43	+86 +51	+89 +54	+103 +63	+105 +65	+108 +68	+123 +77	+126 +80	+130 +84

Dimensions in mm

Tolerance in microns (0.001 mm) (normal tolerance)

Shaft tolerance, interference or clearance in microns (0.001 mm)

Nominal shaft dimension over / to		250 280		280 315		315 355		355 400		400 450		450 500	
Bearing bore diameter deviation	Δ_{dmp}	0 –35		0 –35		0 –40		0 –40		0 –45		0 –45	
Diagram of fit Shaft	Δ_{dmp} - 0 +												
n5		+57 +34	**92** **73** **34**	+57 +34	**92** **73** **34**	+62 +37	**102** **80** **37**	+62 +37	**102** **80** **37**	+67 +40	**112** **88** **40**	+67 +40	**112** **88** **40**
n6		+66 +34	**101** **79** **34**	+66 +34	**101** **79** **34**	+73 +37	**113** **88** **37**	+73 +37	**113** **88** **37**	+80 +40	**125** **97** **40**	+80 +40	**125** **97** **40**
p6		+88 +56	**123** **101** **56**	+88 +56	**123** **101** **56**	+98 +62	**138** **113** **62**	+98 +62	**138** **113** **62**	+108 +68	**153** **125** **68**	+108 +68	**153** **125** **68**
p7		+108 +56	**143** **114** **56**	+108 +56	**143** **114** **56**	+119 +62	**159** **127** **62**	+119 +62	**159** **127** **62**	+131 +68	**176** **139** **68**	+131 +68	**176** **139** **68**
r6		+126 +94	**161** **138** **94**	+130 +98	**165** **142** **98**	+144 +108	**184** **159** **108**	+150 +114	**190** **165** **114**	+166 +126	**211** **183** **126**	+172 +132	**217** **189** **132**
r7		+146 +94	**181** **152** **94**	+150 +98	**185** **156** **98**	+165 +108	**205** **173** **108**	+171 +114	**211** **179** **114**	+189 +126	**234** **198** **126**	+195 +132	**240** **204** **132**

Nominal shaft dimension over / to	500 560		560 630		630 710		710 800		800 900		900 1000		1000 1120		1120 1250		1250 1600	
Bearing bore diameter deviation Δ_{dmp}	0 –50		0 –50		0 –75		0 –75		0 –100		0 –100		0 –125		0 –125		0 –160	
n5	+73 +44	**123** **96** **44**	+73 +44	**123** **96** **44**	+82 +50	**157** **121** **50**	+82 +50	**157** **121** **50**	+92 +56	**192** **147** **56**	+92 +56	**192** **147** **56**	+108 +66	**233** **177** **66**	+108 +66	**233** **177** **66**	+128 +78	**288** **218** **78**
n6	+88 +44	**138** **106** **44**	+88 +44	**138** **106** **44**	+100 +50	**175** **133** **50**	+100 +50	**175** **133** **50**	+112 +56	**212** **160** **56**	+112 +56	**212** **160** **56**	+132 +66	**257** **193** **66**	+132 +66	**257** **193** **66**	+156 +78	**316** **237** **78**
p6	+122 +78	**172** **140** **78**	+122 +78	**172** **140** **78**	+138 +88	**213** **171** **88**	+138 +88	**213** **171** **88**	+156 +100	**256** **204** **100**	+156 +100	**256** **204** **100**	+186 +120	**311** **247** **120**	+186 +120	**311** **247** **120**	+218 +140	**378** **299** **140**
p7	+148 +78	**198** **158** **78**	+148 +78	**198** **158** **78**	+168 +88	**243** **199** **88**	+168 +88	**243** **199** **88**	+190 +100	**290** **227** **100**	+190 +100	**290** **227** **100**	+225 +120	**350** **273** **120**	+225 +120	**350** **273** **120**	+265 +140	**425** **330** **140**
r6	+194 +150	**244** **212** **150**	+199 +155	**249** **217** **155**	+225 +175	**300** **258** **175**	+235 +185	**310** **268** **185**	+266 +210	**366** **314** **210**	+276 +220	**376** **324** **220**	+316 +250	**441** **377** **250**	+326 +260	**451** **387** **260**		
r7	+220 +150	**270** **230** **150**	+225 +155	**275** **235** **155**	+255 +175	**330** **278** **175**	+265 +185	**340** **288** **185**	+300 +210	**400** **337** **210**	+310 +220	**410** **347** **220**	+355 +250	**480** **403** **250**	+365 +260	**490** **413** **260**		

Example: Shaft dia 560 p6

Maximum material	+122	**172**	Interference or clearance when upper shaft deviations coincide with lower bore deviations
		140	Probable interference or clearance
Minimum material	+78	**78**	Interference or clearance when lower shaft deviations coincide with upper bore deviations

Numbers in **boldface print** identify interference.
Standard-type numbers in right column identify clearance.

Shaft tolerances for withdrawal and adapter sleeves

Shaft tolerances in microns (0.001 mm)

Nominal shaft dimension over / to	250 280		280 315		315 355		355 400		400 450		450 500	
h7/$\frac{IT5}{2}$	0 –52	*11.5*	0 –52	*11.5*	0 –57	*12.5*	0 –57	*12.5*	0 –63	*13.5*	0 –63	*13.5*
h8/$\frac{IT5}{2}$	0 –81	*11.5*	0 –81	*11.5*	0 –89	*12.5*	0 –89	*12.5*	0 –97	*13.5*	0 –97	*13.5*
h9/$\frac{IT6}{2}$	0 –130	*16*	0 –130	*16*	0 –140	*18*	0 –140	*18*	0 –155	*20*	0 –155	*20*

Nominal shaft dimension over / to	500 560		560 630		630 710		710 800		800 900		900 1000		1000 1120		1120 1250		1250 1600	
h7/$\frac{IT5}{2}$	0 –70	*14.5*	0 –70	*14.5*	0 –80	*16*	0 –80	*16*	0 –90	*18*	0 –90	*18*	0 –105	*21*	0 –105	*21*	0 –125	*25*
h8/$\frac{IT5}{2}$	0 –110	*14.5*	0 –110	*14.5*	0 –125	*16*	0 –125	*16*	0 –140	*18*	0 –140	*18*	0 –165	*21*	0 –165	*21*	0 –195	*25*
h9/$\frac{IT6}{2}$	0 –175	*22*	0 –175	*22*	0 –200	*25*	0 –200	*25*	0 –230	*28*	0 –230	*28*	0 –260	*33*	0 –260	*33*	0 –310	*39*

The numbers printed in italics are guiding values for the tolerance of cylindricity t_1 (DIN ISO 1101).

Radial bearings

Type of load	Axial displaceability Load	Operating conditions	Tolerances
Point load on outer ring	Floating bearing, easy displacement of outer ring	Closeness of tolerance based on required running accuracy	H7 (H6*)
	Outer ring generally displaceable, angular contact ball bearings and tapered roller-bearings with adjustment via outer ring	High running accuracy required	H6 (J6)
		Standard running accuracy	H7 (J7)
		External heating through shaft	G7**)
Circumferential load on outer ring or indeterminate load	Low load	With high running accuracy requirements K6, M6, N6 and P6	K7 (K6)
	Normal load, shocks		M7 (M6)
	High load, shocks		N7 (N6)
	High load, heavy shocks thin-walled housings		P7 (P6)

*) G7 for housings made of GG, with a bearing outside diameter D > 250 mm and the temperature difference between outer ring and housing > 10 K
**) F7 for housings made of GG, with a bearing outside diameter D > 250 mm and the temperature difference between outer ring and housing > 10 K

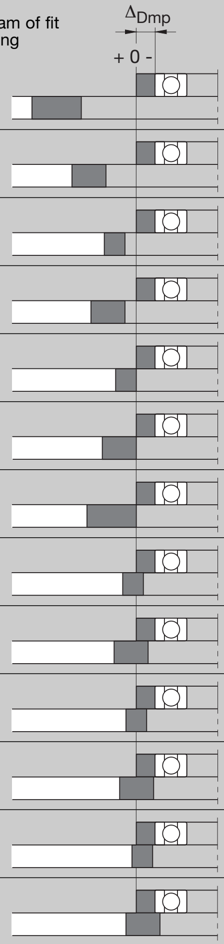
Thrust bearings

Type of load	Bearing type	Operating conditions	Tolerances
Thrust load	Thrust ball bearings	Standard running accuracy High running accuracy	E8 H6
	Cylindrical roller thrust bearings with housing washer		H7 (K7)
	Thrust cylindrical roller and cage assemblies		H10
	Spherical roller thrust bearings	Normal load High load	E8 G7
Combined loading point load on housing washer	Spherical roller thrust bearings		H7
Combined loading circumferential load on housing washer	Spherical roller thrust bearings		K7

Dimensions in mm

Nominal housing bore	over to	6 10	10 18	18 30	30 50	50 80	80 120
		Tolerance in microns (0.001 mm) (normal tolerance)					
Bearing outside diameter deviation	Δ_{Dmp}	0 –8	0 –8	0 –9	0 –11	0 –13	0 –15

Housing tolerance, interference or clearance in microns (0.001 mm)

Diagram of fit Housing		6–10		10–18		18–30		30–50		50–80		80–120	
E8		+47 +25	25 35 55	+59 +32	32 44 67	+73 +40	40 54 82	+89 +50	50 67 100	+106 +60	60 79 119	+126 +72	72 85 141
F7		+28 +13	13 21 36	+34 +16	16 25 42	+41 +20	20 30 50	+50 +25	25 37 61	+60 +30	30 44 73	+71 +36	36 53 86
G6		+14 +5	5 11 22	+17 +6	6 12 25	+20 +7	7 14 29	+25 +9	9 18 36	+29 +10	10 21 42	+34 +12	12 24 49
G7		+20 +5	5 13 28	+24 +6	6 15 32	+28 +7	7 17 37	+34 +9	9 21 45	+40 +10	10 24 53	+47 +12	12 29 62
H6		+9 0	**0** 6 17	+11 0	**0** 6 19	+13 0	**0** 7 22	+16 0	**0** 9 27	+19 0	**0** 11 32	+22 0	**0** 12 37
H7		+15 0	**0** 8 23	+18 0	**0** 9 26	+21 0	**0** 10 30	+25 0	**0** 12 36	+30 0	**0** 14 43	+35 0	**0** 17 50
H8		+22 0	**0** 10 30	+27 0	**0** 12 35	+33 0	**0** 14 42	+39 0	**0** 17 50	+46 0	**0** 20 59	+54 0	**0** 23 69
J6		+5 –4	**4** 2 13	+6 –5	**5** 1 14	+8 –5	**5** 2 17	+10 –6	**6** 3 21	+13 –6	**6** 5 26	+16 –6	**6** 6 31
J7		+8 –7	**7** 1 16	+10 –8	**8** 1 18	+12 –9	**9** 1 21	+14 –11	**11** 1 25	+18 –12	**12** 2 31	+22 –13	**13** 4 37
JS6		+4.5 –4.5	**4.5** 2 12.5	+5.5 –5.5	**5.5** 1 13.5	+6.5 –6.5	**6.5** **0** 15.5	+8 –8	**8** 1 19	+9.5 –9.5	**9.5** **0** 22.5	+11 –11	**11** 1 26
JS7		+7.5 –7.5	**7.5** 1 15.5	+9 –9	**9** **0** 17	+10.5 –10.5	**10.5** **1** 19.5	+12.5 –12.5	**12.5** **1** 23.5	+15 –15	**15** **1** 28	+17.5 –17.5	**17.5** **1** 32.5
K6		+2 –7	**7** **1** 10	+2 –9	**9** **3** 10	+2 –11	**11** **4** 11	+3 –13	**13** **4** 14	+4 –15	**15** **4** 17	+4 –18	**18** **6** 19
K7		+5 –10	**10** **2** 13	+6 –12	**12** **3** 14	+6 –15	**15** **5** 15	+7 –18	**18** **6** 18	+9 –21	**21** **7** 22	+10 –25	**25** **8** 25

Example: Housing bore dia 100 K6

Minimum material	+4	**18**	Interference or clearance when upper outside diameter deviations of ring coincide with lower housing bore deviations
		6	Probable interference or clearance
Maximum material	-18	19	Interference or clearance when lower outside diameter deviations of ring coincide with upper housing bore deviations

Numbers in **boldface print** identify interference.
Standard-type numbers in right column identify clearance.

Design of Surrounding Structure
Housing fits

Dimensions in mm

Tolerance in microns (0.001 mm) (normal tolerance)

Housing tolerance, interference or clearance in microns (0.001 mm)

Nominal housing bore over to	120 150		150 180		180 250		250 315		315 400		400 500	
Bearing outside diameter deviation Δ_{Dmp}	0 −18		0 −25		0 −30		0 −35		0 −40		0 −45	
E8	+148 +85	85 112 166	+148 +85	85 114 173	+172 +100	100 134 202	+191 +110	110 149 226	+214 +125	125 168 254	+232 +135	135 182 277
F7	+83 +43	43 62 101	+83 +43	43 64 108	+96 +50	50 75 126	+108 +56	56 85 143	+119 +62	62 94 159	+131 +68	68 104 176
G6	+39 +14	14 28 57	+39 +14	14 31 64	+44 +15	15 35 74	+49 +17	17 39 84	+54 +18	18 43 94	+60 +20	20 48 105
G7	+54 +14	14 33 72	+54 +14	14 36 79	+61 +15	15 40 91	+69 +17	17 46 104	+75 +18	18 50 115	+83 +20	20 56 128
H6	+25 0	**0** 14 43	+25 0	**0** 17 50	+29 0	**0** 20 59	+32 0	**0** 22 67	+36 0	**0** 25 76	+40 0	**0** 28 85
H7	+40 0	**0** 19 58	+40 0	**0** 22 65	+46 0	**0** 25 76	+52 0	**0** 29 87	+57 0	**0** 32 97	+63 0	**0** 36 108
H8	+63 0	**0** 27 81	+63 0	**0** 29 88	+72 0	**0** 34 102	+81 0	**0** 39 116	+89 0	**0** 43 129	+97 0	**0** 47 142
J6	+18 −7	**7** 7 36	+18 −7	**7** 10 43	+22 −7	**7** 13 52	+25 −7	**7** 15 60	+29 −7	**7** 18 69	+33 −7	**7** 21 78
J7	+26 −14	**14** 5 44	+26 −14	**14** 8 51	+30 −16	**16** 9 60	+36 −16	**16** 13 71	+39 −18	**18** 14 79	+43 −20	**20** 16 88
JS6	+12.5 −12.5	**12.5** 1 30.5	+12.5 −12.5	**12.5** 3 37.5	+14.5 −14.5	**14.5** 5 44.5	+16 −16	**16** 7 51	+18 −18	**18** 6 58	+20 −20	**20** 8 65
JS7	+20 −20	**20** 1 38	+20 −20	**20** 1 45	+23 −23	**23** 2 53	+26 −26	**26** 3 61	+28.5 −28.5	**28.5** 3 68.5	+31.5 −31.5	**31.5** 4 76.5
K6	+4 −21	**21** **7** 22	+4 −21	**21** **4** 29	+5 −24	**24** **4** 35	+5 −27	**27** **5** 40	+7 −29	**29** **4** 47	+8 −32	**32** **4** 53
K7	+12 −28	**28** **9** 30	+12 −28	**28** **6** 37	+13 −33	**33** **8** 43	+16 −36	**36** **7** 51	+17 −40	**40** **8** 57	+18 −45	**45** **9** 63

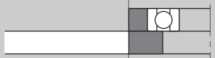
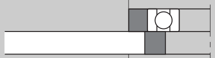
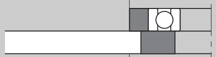
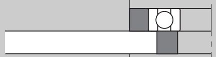
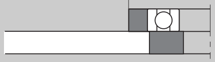
Nominal housing bore over to	500 630		630 800		800 1000		1000 1250		1250 1600		1600 2000		2000 2500	
Bearing outside diameter deviation Δ_{Dmp}	0 −50		0 −75		0 −100		0 −125		0 −160		0 −200		0 −250	
E8	+255 +145	145 199 305	+285 +160	160 227 360	+310 +170	170 250 410	+360 +195	195 292 485	+415 +220	220 338 575	+470 +240	240 384 670	+540 +260	260 436 790
F7	+146 +76	76 116 196	+160 +80	80 132 235	+176 +86	86 149 276	+203 +98	98 175 328	+235 +110	110 205 395	+270 +120	120 237 470	+305 +130	130 271 555
G6	+66 +22	22 54 116	+74 +24	24 66 149	+82 +26	26 78 182	+94 +28	28 93 219	+108 +30	30 109 268	+124 +32	32 130 324	+144 +34	34 154 394
G7	+92 +22	22 62 142	+104 +24	24 76 179	+116 +26	26 89 216	+133 +28	28 105 258	+155 +30	30 125 315	+182 +32	32 149 382	+209 +34	34 175 459
H6	+44 0	**0** 32 94	+50 0	**0** 42 125	+56 0	**0** 52 156	+66 0	**0** 64 191	+78 0	**0** 79 238	+92 0	**0** 98 292	+110 0	**0** 120 360
H7	+70 0	**0** 40 120	+80 0	**0** 52 155	+90 0	**0** 63 190	+105 0	**0** 77 230	+125 0	**0** 95 285	+150 0	**0** 117 350	+175 0	**0** 142 425
H8	+110 0	**0** 54 160	+125 0	**0** 67 200	+140 0	**0** 80 240	+165 0	**0** 97 290	+195 0	**0** 118 355	+230 0	**0** 143 430	+280 0	**0** 177 530
J6														
J7														
JS6	+22 −22	**22** 10 72	+25 −25	**25** 17 100	+28 −28	**28** 24 128	+33 −33	**33** 31 158	+39 −39	**39** 40 199	+46 −46	**46** 52 246	+55 −55	**55** 65 305
JS7	+35 −35	**35** 5 85	+40 −40	**40** 12 115	+45 −45	**45** 18 145	+52 −52	**52** 24 177	+62 −62	**62** 32 222	+75 −75	**75** 42 275	+87 −87	**87** 54 337
K6	0 −44	**44** **12** 50	0 −50	**50** **8** 75	0 −56	**56** **4** 100	0 −66	**66** **2** 125	0 −78	**78** 1 160	0 −92	**92** 6 200	0 −110	**110** 10 250
K7	0 −70	**70** **30** 50	0 −80	**80** **28** 75	0 −90	**90** **27** 100	0 −105	**105** **28** 125	0 −125	**125** **30** 160	0 −150	**150** **33** 200	0 −175	**175** **34** 250

Example: Housing bore dia 560 K6

Minimum material	0	**44**	Interference or clearance when upper outside diameter deviations of ring coincide with lower housing bore deviations
		12	Probable interference or clearance
Maximum material	-44	50	Interference or clearance when lower outside diameter deviations of ring coincide with upper housing bore deviations

Numbers in **boldface print** identify interference.
Standard-type numbers in right column identify clearance.

Dimensions in mm

Nominal housing bore over to		6 10		10 18		18 30		30 50		50 80		80 120	
		Tolerance in microns (0.001 mm) (normal tolerance)											
Bearing outside diameter deviation Δ_{Dmp}		0 −8		0 −8		0 −9		0 −11		0 −13		0 −15	
Diagram of fit Housing Δ_{Dmp} + 0 -		Housing tolerance, interference or clearance in microns (0.001 mm)											
M6		−3 −12	**12** **6** 5	−4 −15	**15** **9** 4	−4 −17	**17** **10** 5	−4 −20	**20** **11** 7	−5 −24	**24** **13** 8	−6 −28	**28** **16** 9
M7		0 −15	**15** **7** 8	0 −18	**18** **9** 8	0 −21	**21** **11** 9	0 −25	**25** **13** 11	0 −30	**30** **16** 13	0 −35	**35** **18** 15
N6		−7 −16	**16** **10** 1	−9 −20	**20** **14** **1**	−11 −24	**24** **17** **2**	−12 −28	**28** **19** **1**	−14 −33	**33** **22** **1**	−16 −38	**38** **26** **1**
N7		−4 −19	**19** **11** 4	−5 −23	**23** **14** 3	−7 −28	**28** **18** 2	−8 −33	**33** **21** 3	−9 −39	**39** **25** 4	−10 −45	**45** **28** 5
P6		−12 −21	**21** **15** **4**	−15 −26	**26** **20** **7**	−18 −31	**31** **24** **9**	−21 −37	**37** **28** **10**	−26 −45	**45** **34** **13**	−30 −52	**52** **40** **15**
P7		−9 −24	**24** **16** **1**	−11 −29	**29** **20** **3**	−14 −35	**35** **25** **5**	−17 −42	**42** **30** **6**	−21 −51	**51** **37** **8**	−24 −59	**59** **42** **9**

Nominal housing bore over to	120 150		150 180		180 250		250 315		315 400		400 500		500 630		630 800		800 1000	
Bearing outside diameter deviation Δ_{Dmp}	0 −18		0 −25		0 −30		0 −35		0 −40		0 −45		0 −50		0 −75		0 −100	
M6	−8 −33	**33** **19** 10	−8 −33	**33** **16** 17	−8 −37	**37** **17** 22	−9 −41	**41** **19** 26	−10 −46	**46** **21** 30	−10 −50	**50** **22** 35	−26 −70	**70** **38** 24	−30 −80	**80** **38** 45	−34 −90	**90** **38** 66
M7	0 −40	**40** **21** 18	0 −40	**40** **18** 25	0 −46	**46** **21** 30	0 −52	**52** **23** 35	0 −57	**57** **25** 40	0 −63	**63** **27** 45	−26 −96	**96** **56** 24	−30 −110	**110** **58** 45	−34 −124	**124** **61** 66
N6	−20 −45	**45** **31** **2**	−20 −45	**45** **28** 5	−22 −51	**51** **31** 8	−25 −57	**57** **35** 10	−26 −62	**62** **37** 14	−27 −67	**67** **39** 18	−44 −88	**88** **56** 6	−50 −100	**100** **58** 25	−56 −112	**112** **60** 44
N7	−12 −52	**52** **33** 6	−12 −52	**52** **30** 13	−14 −60	**60** **35** 16	−14 −66	**66** **37** 21	−16 −73	**73** **41** 24	−17 −80	**80** **44** 28	−44 −114	**114** **74** 6	−50 −130	**130** **78** 25	−56 −146	**146** **83** 44
P6	−36 −61	**61** **47** **18**	−36 −61	**61** **44** **11**	−41 −70	**70** **50** **11**	−47 −79	**79** **57** **12**	−51 −87	**87** **62** **11**	−55 −95	**95** **67** **10**	−78 −122	**122** **90** **28**	−88 −138	**138** **96** **13**	−100 −156	**156** **104** **0**
P7	−28 −68	**68** **49** **10**	−28 −68	**68** **46** **3**	−33 −79	**79** **54** **3**	−36 −88	**88** **59** **1**	−41 −98	**98** **66** **1**	−45 −108	**108** **72** **0**	−78 −148	**148** **108** **28**	−88 −168	**168** **126** **13**	−100 −190	**190** **127** **0**

Example: Housing bore dia 100 M7

Minimum material	0	**35**	Interference or clearance when upper outside diameter deviations of ring coincide with lower housing bore deviations
		18	Probable interference or clearance
Maximum material	-35	15	Interference or clearance when lower outside diameter deviations of ring coincide with upper housing bore deviations

Numbers in **boldface print** identify interference.
Standard-type numbers in right column identify clearance.

Dimensions in mm									
Nominal housing bore	over to	1000 1250		1250 1600		1600 2000		2000 2500	
Tolerance in microns (0.001 mm) (normal tolerance)									
Bearing outside diameter deviation	Δ_{Dmp}	0 –125		0 –160		0 –200		0 –250	
Diagram of fit Housing (Δ_{Dmp}, + 0 -)		Housing tolerance, interference or clearance in microns (0.001 mm)							
M6		–40 –106	**106** **45** 85	–48 –126	**126** **47** 112	–58 –150	**150** **52** 142	–68 –178	**178** **58** 182
M7		–40 –145	**145** **68** 85	–48 –173	**173** **78** 112	–58 –208	**208** **91** 142	–68 –178	**243** **102** 182
N6		–66 –132	**132** **67** 59	–78 –156	**156** **77** 82	–92 –184	**184** **86** 108	–110 –220	**220** **100** 140
N7		–66 –171	**171** **94** 59	–78 –203	**203** **108** 82	–92 –242	**242** **125** 108	–110 –285	**285** **144** 140
P6		–120 –186	**186** **121** 5	–140 –218	**218** **139** 20	–170 –262	**262** **164** 30	–195 –305	**305** **185** 55
P7		–120 –225	**225** **148** 5	–140 –265	**265** **159** 20	–170 –320	**320** **203** 30	–195 –370	**370** **229** 55

Raceways with direct bearing arrangements

In the case of cylindrical roller bearings without inner ring or outer ring (designs RNU, RN, available on request), the rollers run directly on the hardened and ground shaft or in the housing.

The raceways must have a hardness between 58 and 64 HRC and an average roughness value $R_a \leq 0.2$ µm, so that the full load carrying capacity of the bearing is reached.

Contact washers and shaft shoulders must also be hardened.

Proven materials for raceways include through-hardening steels according to DIN 17230, e.g. the rolling bearing steel 100 Cr 6 (mat. no. 1.3505) and casehardening steels, e.g. 17 MnCr 5 (mat. no. 1.3521) and 16 CrNiMo 6 (mat. no. 1.3531).

With casehardening steels the minimum case depth Eht_{min} of the ground raceways depends on the load, the diameter of the rolling elements and the core strength of the steel used. The following formula applies to approximate calculations:

Min. case depth $Eht_{min} = (0.07 \text{ to } 0.12)\, D_w$

where D_w is the diameter of the rolling element.

The higher value should be applied to low core strength and/or heavy loads. The case depth should not drop below 0.3 mm.

High-alloy steels can also be used such as Cf 54 (mat. no. 1.1219) or 43 CrMo 4 (mat. no. 1.3563). These steel grades may be flame-hardened or induction-hardened. The following formula applies to the minimum depth of the hardened surface layer:

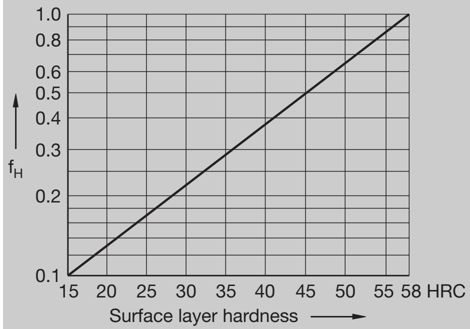
$Rht_{min} = (0.1 \text{ to } 0.18)\, D_w$

where D_w is the diameter of the rolling element.

The higher value should be applied to low core strength and/or heavy loads.

If the surface layer hardness of the raceways is less than 58 HRC, the bearing will not attain its full load carrying capacity. In such a case, the dynamic load rating C or the static load rating C_0 must be reduced by the factor f_H, see diagram.

▼ Factor f_H to take raceway hardness into account



A wave-free finish is required for the raceways. With an average roughness value $R_a > 0.2$ µm the bearing load carrying capacity cannot be fully utilized.

In direct bearing arrangements, the bearing clearance is determined by the diameter tolerances of the shaft and the housing. More information on the bearing clearance and on the machining tolerances can be found in the texts preceding the individual catalogue sections.

The table below shows values recommended for the machining tolerance and the form tolerance of direct bearing arrangement raceways at normal and high demands on running accuracy.

Axial fixation of the bearings

Depending on their different guidance functions, locating bearings, floating bearings, adjusted and floating bearing arrangements are distinguished between (cf. "Selection of bearing arrangement" page 24). The axial fixation of the bearing rings is adapted to the bearing arrangement in question.

Locating bearings and floating bearings

Locating bearings have to accommodate axial forces of varying magnitude, which is also a decisive factor for the holding element. Examples of holding elements are: shoulders on shafts and housings, snap rings, housing covers, shaft end caps, nuts, spacers, etc.

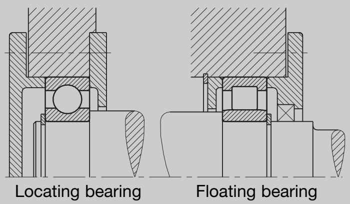
Floating bearings have to transmit only small axial forces resulting from thermal expansions so that the axial location merely has to prevent lateral displacement of the ring. A tight fit frequently does the job. With non-separable bearings, only one ring has to be firmly fitted; the other ring is held by the rolling elements.

▼ Values recommended for machining the raceways in direct bearing arrangements

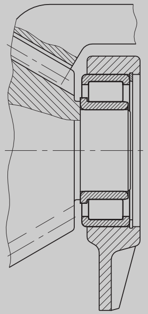
Running accuracy	Raceway	Machining tolerance	Cylindricity DIN ISO 1101	Squareness of abutment shoulder	Axial runout of raceways
Radial bearings					
Normal	Shaft	IT6	$\frac{IT3}{2}$	IT3	
	Housing	IT6	$\frac{IT3}{2}$	IT3	
High	Shaft	IT4	$\frac{IT1}{2}$	IT1	
	Housing	IT5	$\frac{IT2}{2}$	IT2	
Thrust bearings					
Normal					IT5
High					IT4

The IT qualities for high running accuracy should also be applied with high speeds and small radial clearance.

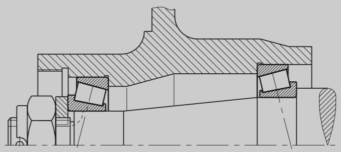
▼ Axial fixation of a deep groove ball bearing and a cylindrical roller bearing outer ring due to positive contact



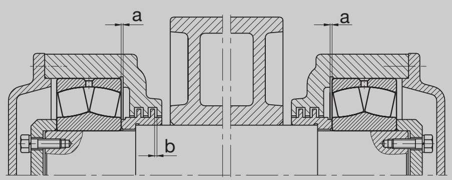
▼ Cylindrical roller bearing of design NJ mounted as floating bearing where the inner ring lip prevents axial movement to one side



▼ Axial fixation in adjusted bearing arrangements



▼ Axial fixation in floating bearing arrangements
a = guiding clearance; a < b (b = axial labyrinth gap)



Adjusted and floating bearing arrangements

Since adjusted and floating bearing arrangements transmit axial forces only in one direction, the bearing rings need to be supported only on one side. Another bearing, which is symmetrically arranged, accommodates the opposite force. Locknuts, ring nuts, covers or spacers are used as adjusting elements. In floating bearing arrangements, the movement of the rings to the side is restricted by shaft or housing shoulders, covers, snap rings etc.

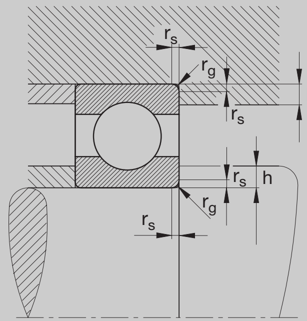
Abutment dimensions

The bearing rings should closely fit the shaft or housing shoulder, they must not be allowed to foul the shoulder fillet. Consequently, the maximum fillet radius r_g of the mating part must be smaller than the minimum corner r_{smin} (see page 52) of the bearing.

The shoulder of the mating parts must be so high that even with maximum bearing corner there is an adequate abutment surface (DIN 5418).

The bearing tables list the maximum fillet radius r_g and the diameters of the abutment shoulders. Special features of individual bearing types, e.g. cylindrical roller bearings, tapered roller bearings and thrust bearings are explained in the text preceding the tables.

▼ Abutment dimensions according to DIN 5418



Sealing

The seal has a considerable influence on the service life of a bearing arrangement. On the one hand, it should prevent the lubricant from escaping from the bearing, and, on the other, prevent contaminants from entering the bearing.

Contaminants have diverse effects:

- A large number of tiny particles act as abrasives and lead to wear in the bearing. An increase in clearance or the development of more noise puts an end to the service life of the bearing.
- Larger, cycled hard particles reduce the fatigue life because pittings develop at indentations when the bearing loads are high.

In principle, a distinction is made between contact-free or non-rubbing and contact or rubbing seals.

Non-rubbing seals

The only friction arising with non-rubbing seals is the lubricant friction in the lubricating gap. The seals do not show any wear and can function for a long time. Since non-rubbing seals do not generate any heat, they are suitable for very high speeds.

A simple means of protection which is frequently adequate, is a narrow sealing gap between shaft and housing (a).

Labyrinths (b), whose gaps are filled with grease, have a far greater sealing effect. If the environment is dirty, grease is pressed from the inside into the sealing gaps in short time intervals.

In the case of oil lubrication with horizontal shafts, splash rings (c) are suitable for preventing oil from escaping. The oil drain hole at the bottom of the sealing area should be large enough to prevent its being clogged by dirt.

Flinger rings (d) which rotate with the shaft protect the sealing gap from heavy dirt.

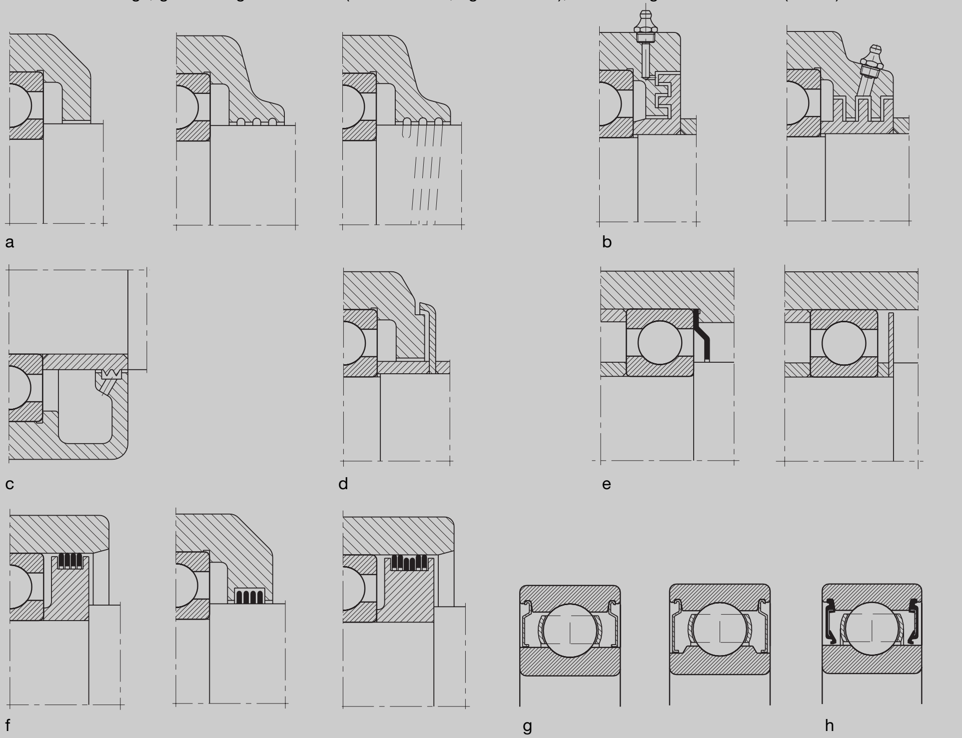
Stationary baffle plates (e) prevent grease from escaping from the bearing. The grease collar which forms at the sealing gap protects the bearing from contaminants.

Lamellar rings of steel (f) with spring disks to the outside or to the inside need a small mounting space. They seal against grease loss and dust penetration and are also used as a preseal against splashing water.

Space-saving sealing elements are dust shields (g) mounted in the bearing at either one or both ends. Bearings with dust shields at both ends (suffix .2ZR, with very small bearings .2Z) are supplied with a grease filling.

The sealing lip of RSD seals (h) forms a narrow gap at the inner ring. The friction is as low as with bearing shields. The advantage of sealing washers over dust shields is their outer rubber-elastic bead which ensures efficient sealing in the outer ring groove. This is important for rotating outer rings because the base oil extracted from the base soap by the centrifugal force would escape through the gap between the metallic shield and the outer ring. With RSD seals, outer ring speeds up to the permissible limit can be attained.

▼ Non-rubbing seals
a = gap type seals, b = labyrinth seals, c = splash ring, d = flinger ring, e = baffle plates,
f = lamellar rings, g = bearing with shields (left side .2ZR, right side .2Z), h = bearing with RSD seals (.2RSD)



Rubbing seals

Rubbing seals (see page 126) contact their metallic running surfaces under a certain force (usually radial). This force should be kept to a minimum to prevent excessive increases in the frictional moment and the temperature. The lubrication condition at the contact surface, the roughness of the contact surface, and the sliding velocity also influence the frictional moment and the temperature as well as the seal wear.

Felt rings (a) are simple sealing elements which prove particularly successful with grease lubrication. They are soaked in oil before mounting, and are an especially good means of sealing against dust. If environmental conditions are adverse, two felt rings can be arranged side by side.

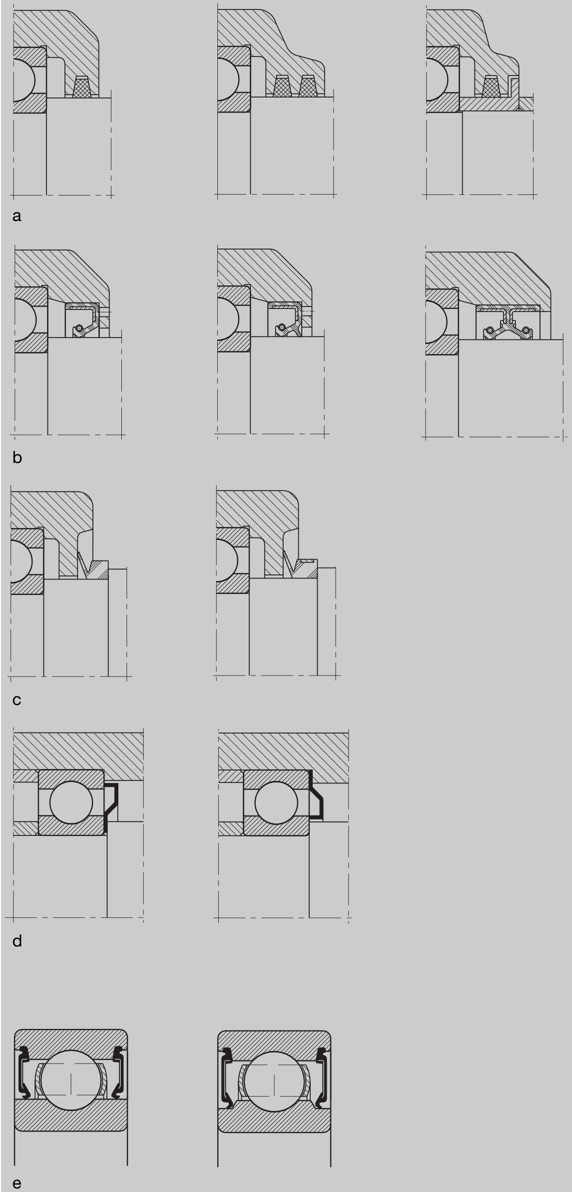
Radial shaft seals (b) are, above all, used at oil lubrication. The sealing ring, equipped with a lip, is forced against the sliding surface of the shaft by a spring. If the chief aim is to prevent the escape of lubricant, the lip is on the inside. A sealing ring with an additional protection lip also prevents the dirt penetration. With oil lubrication, sealing lips of the usual material, nitrile butadiene rubber (NBR), are suitable for circumferential velocities at the contact surface of up to 12 m/s.

The V-ring (c) is a lip seal with axial effect. During mounting, this one-piece rubber ring is pushed onto the shaft under tension until its lip contacts the housing wall. The sealing lip also acts as a flinger ring. Axial lip seals are insensitive to radial misalignment and slight shaft inclinations. With grease lubrication, rotating V-rings are suitable for circumferential velocities of up to 12 m/s, stationary ones up to 20 m/s. For circumferential velocities over 8 m/s, V-rings must be axially supported and for those with 12 m/s or more they must also be radially clamped. V-rings are frequently used as preseals in order to keep dirt away from a radial shaft seal.

Spring seals (d) are highly efficient for grease lubrication. They consist of thin sheet metal and are clamped to the face of the inner or the outer ring while the sealing edge contacts the other ring under slight tension.

Simple designs are possible with bearings with one or two sealing washers (e). The washers are suitable to seal against dust, dirt, a moist atmosphere, and slight pressure differences. FAG supply maintenance-free bearings with two sealing washers and a grease filling (cf. "Grease supply to bearings", page 130). The most commonly used seal design RSR made of acrylo-nitrile-butadiene rubber (NBR) for deep groove ball bearings is lightly pressed on the ground inner ring. Design RS for deep groove ball bearings contacts a chamfer at the inner ring.

▼ Rubbing seals
a = felt rings or felt strips, b = radial shaft seals c = V-rings, d = spring seals,
e = bearing with seals (left side .2RSR, right side .2RS)



Lubrication and maintenance

Lubricating film formation

The primary task of the lubrication of rolling bearings is the avoidance of wear and premature fatigue, thus ensuring sufficiently long service life. Lubrication is also intended to promote favourable running properties such as low noise operation and slight friction. The lubricating film created between the load-transmitting parts is supposed to prevent metal-to-metal contact. Film thickness is calculated by means of the theory of elastohydrodynamic lubrication (cf. FAG Publication No. WL 81 115 "Rolling Bearing Lubrication").

With a simplified method, the lubrication condition is described by means of the ratio of the operating viscosity ν to the rated viscosity ν_1. The latter depends on the speed n and the mean bearing diameter d_m, see upper diagram on page 43.

According to DIN ISO 281, the nominal rating life of the rolling bearings is based on the assumption that the operating viscosity ν of the oil used is at least as high as the rated viscosity ν_1. The operating viscosity for mineral oils can be computed from the viscosity at 40 °C and the operating temperature with the V-T diagram on page 43.

The adjusted rating life calculation (cf. page 40) takes into account also the effect of an operating viscosity deviating from the rated viscosity, of lubricant doping, and of cleanliness in the lubricating gap on the attainable fatigue life.

The viscosity of the lubricating oil changes with the pressure between the areas in rolling contact. The following formula applies:

$$\eta = \eta_o \cdot e^{\alpha p}$$

where

η	dynamic viscosity at pressure p	[Pa s]
η_o	dynamic viscosity at normal pressure	[Pa s]
e	(=2.71828) basis of the natural logarithms	
α	pressure-viscosity coefficient	[m²/N]
p	pressure	[N/m²]

This is taken into account in the calculation of the lubrication condition according to the EHD theory for mineral oil base lubricants. The upper diagram on page 128 shows the pressure-viscosity behaviour of some lubricants. The zone a to b for mineral oils is the basis for the a_{23} diagram. Mineral oils with EP additives also have α values in this zone.

When the effect of the pressure-viscosity coefficient on the viscosity ratio is strong, e.g. in the case of diester, fluorocarbon or silicone oil, correction factors B_1 and B_2 must be considered for the viscosity ratio κ as follows:

$$\kappa_{B1,2} = \kappa \cdot B_1 \cdot B_2$$

where

κ viscosity ratio at mineral oil

B_1 correction factor for pressure-viscosity behaviour
$= \alpha_{\text{synthetic oil}} / \alpha_{\text{mineral oil}}$

B_2 correction factor for varying density
$= \rho_{\text{synthetic oil}} / \rho_{\text{mineral oil}}$

The lower diagram on page 128 shows the pattern of the density ρ versus the temperature for mineral oils. The pattern for a synthetic oil can be assessed when the density ρ at 15 °C is known.

Selection of lubrication system

The decision as to whether the bearings should be lubricated with grease or oil should be made as early as possible when designing a machine. In special cases, a dry lubrication is also possible (cf. FAG Publication No. WL 81 115 "Rolling Bearing Lubrication").

Grease lubrication

Grease lubrication is used for 90 % of all rolling bearings.

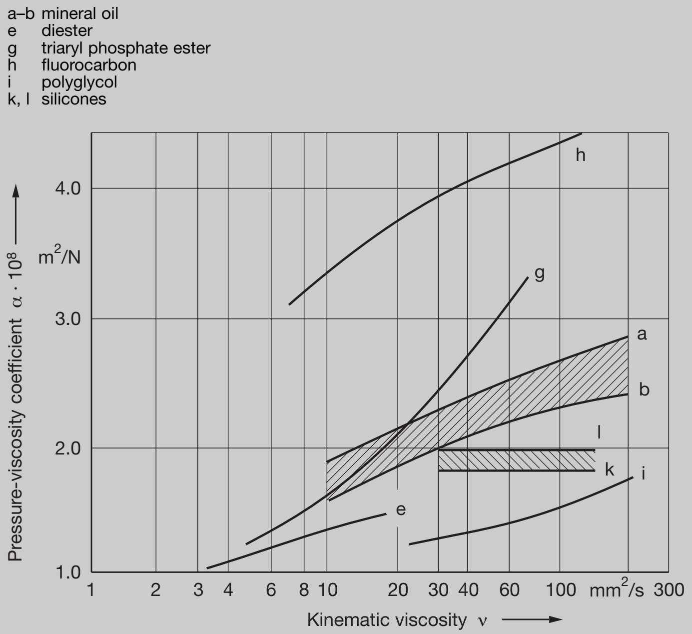
The essential advantages of grease lubrication are:

- simple design
- good sealing properties of grease
- long service life with little maintenance expenditure

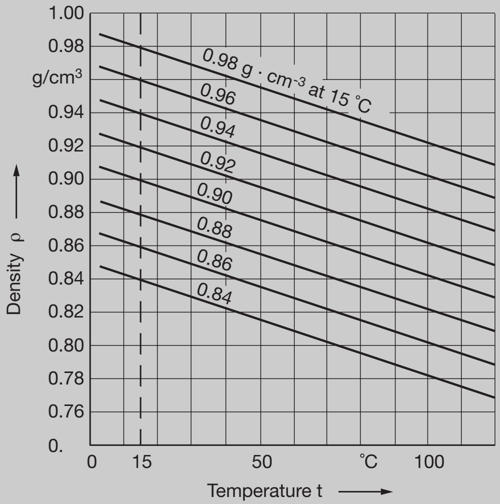
For-life grease lubrication is often used for normal operating and environmental conditions.

If there are high stresses (speed, temperature, loads), relubrication at appropriate intervals must be planned. For relubrication, grease supply and removal ducts and a collecting chamber for the used grease must be provided; in the case of short relubrication intervals, possibly also a grease pump and a grease valve should be available.

▼ Pressure-viscosity coefficient α as a function of the kinematic viscosity ν, applicable to a pressure range from 0 to 2000 bar



▼ Density ρ of mineral oils depending on the temperature t



Oil lubrication

Oil lubrication is practical when adjacent machine elements are already being supplied with oil or when heat should be dissipated by the lubricant. Heat dissipation may be required for high loads and/or high speeds or if the bearing is exposed to extraneous heat.

For oil lubrication with small quantities (throw-away lubrication), designed as drip feed lubrication, oil mist lubrication or oil-air lubrication, the churning friction and, therefore, the bearing friction is kept low.

When using air as a carrier, a direct supply and an air current which supports the sealing are possible.

Direct supply to all contact areas of very fast rotating bearings and good cooling are possible by injecting larger quantities of oil.

Selection of suitable greases

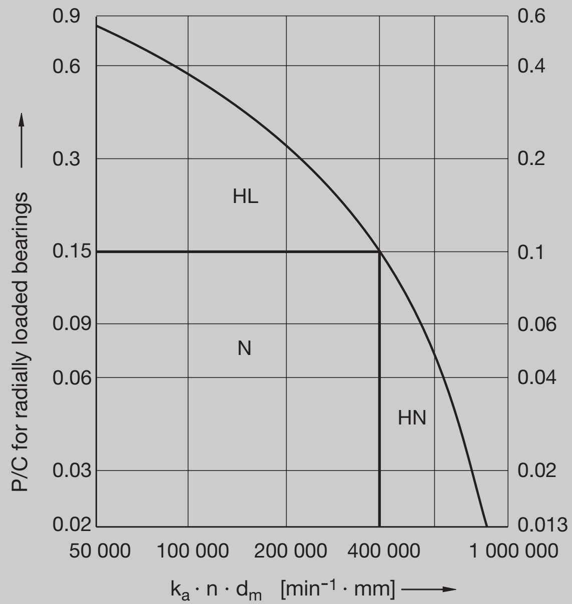
Greases are classified according to thickeners of various composition and to base oils. In principle, the rules of oil lubrication apply to the base oils of greases.

Conventional greases have metal soaps as a thickener and mineral base oil. They are available in various penetration classes (NLGI classes). These greases respond very differently to environmental influences such as temperature and moisture. The diagram below shows an overview for grease selection based on load and speed.

Key:

P/C	specific load	
P	equivalent dynamic load	[kN]
C	dynamic load rating	[kN]
k_a	factor for the bearing type	
n	speed	[min^{-1}]
d_m	mean bearing diameter	[mm]

▼ Grease selection from the load ratio P/C and the relevant bearing speed index $k_a \cdot n \cdot d_m$



Range N

Normal operating conditions
Rolling bearing greases K according to DIN 51 825

Range HL

Range of heavy loads
Rolling bearing greases KP according to DIN 51 825 or other suitable greases

Range HN

High speed range. Greases for high-speed bearings. For bearing types with $k_a > 1$ greases KP according to DIN 51 825 or other suitable greases

k_a values

$k_a = 1$ deep groove ball bearings, angular contact ball bearings, four-point bearings, self-aligning ball bearings, radially loaded cylindrical roller bearings, thrust ball bearings.

$k_a = 2$ spherical roller bearings, tapered roller bearings.

$k_a = 3$ axially loaded cylindrical roller bearings, full-complement cylindrical roller bearings.

For operating cases near the limiting curve, the steady-state temperature is usually high which is why special greases for higher temperatures are required. See the FAG publ. no. WL 81 115 "Rolling Bearing Lubrication" for more details on grease selection.

FAG Arcanol rolling bearing greases are proved lubricants with which almost all requirements for the lubrication of rolling bearings are met. See pages 679 to 681 and FAG publ. no. WL 81 116 "Arcanol – Rolling Bearing-tested Grease" for chemico-physical data, user tips, and data on availability.

Grease supply to bearings

In FAG bearings greased for life, about 30 % of the free inner space is filled with grease which is distributed during the first few operating hours. Afterwards the bearing runs with only 30 % to 50 % of the initial friction.

FAG supply numerous bearings with grease charges:

– deep groove ball bearings of the designs .2ZR (.2Z), .2RSR (.2RS), and .2RSD
– double row angular contact ball bearings of the designs B.TVH, .2ZR and .2RSR
– high-speed spindle bearings of series HSS70 and HSS719 as well as ceramic hybrid spindle bearings of series HCS70 and HCS719,
– self-aligning ball bearings of design .2RS
– double row, full complement cylindrical roller bearings, series NNF50B.2LS.V and NNF50C.2LS.V
– S-type bearings of series 162, 362, 562, 762.2RSR

The user must fill the bearings with grease when they have not already been greased by FAG. Recommendation:

– Fill bearing with grease to such an extent that all functional surfaces are safely covered with grease.
– Fill the housing spaces left and right of the bearing only to such an extent that there is ample room for the grease expelled from the bearing.
– Fill cavities in very quickly rotating bearings ($n \cdot d_m > 500{,}000\ min^{-1} \cdot mm$) only to 20 % to 30 % .
– Bearing and housing cavities can be packed with grease when $n \cdot d_m < 50{,}000\ min^{-1} \cdot mm$.

Bearings running at very high speeds require grease distribution runs, see FAG publ. no. WL 81 115 "Rolling Bearing Lubrication".

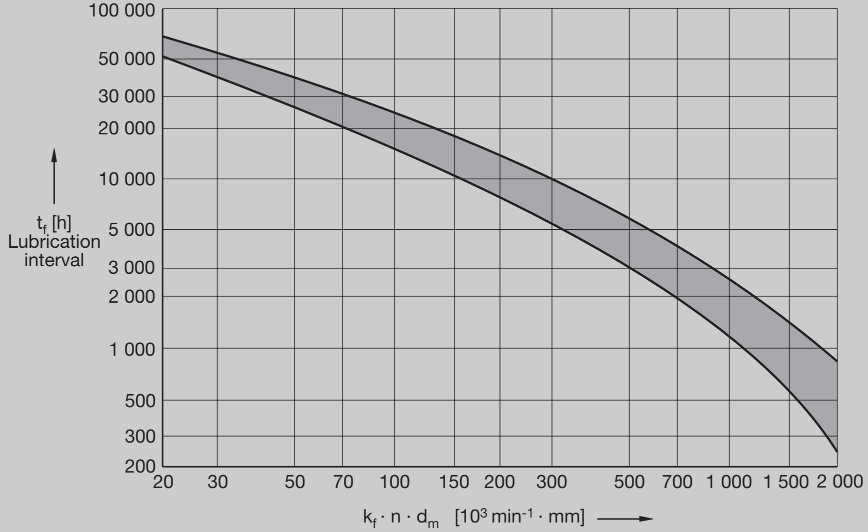
The grease life is the time from the start-up until the bearing fails as a result of lubrication failure. The grease life curve of a certain grease for a failure probability of 10 % is called F_{10}. It is located by means of field trials in the laboratory, for example with the FAG rolling bearing grease test rig FE9. In many cases, the user does not know F_{10} and therefore FAG provide the lubrication interval t_f as a recommended value for the minimum service life of standard greases. The relubrication interval (see below) should be far shorter than the lubrication interval for safety reasons.

The lubrication interval curve, see diagram page 131, guarantees sufficient reliability even for those greases which only fulfill minimum requirements according to DIN 51 825. The lubrication interval is dependent on the bearing-related speed index $k_f \cdot n \cdot d_m$. Various k_f values are indicated for some bearing types. The higher k_f values apply to higher load carrying capacity series and the smaller values to the lighter series of a bearing type. The diagram applies to lithium soap base greases and a temperature of up to 70 °C, measured at the bearing outer ring, as well as a mean bearing load corresponding to $P/C < 0.1$. With higher loads and temperatures, the lubrication interval is shorter. The reduced lubricating interval t_{fq} is the product of lubricating interval t_f and the reduction factors f_1 to f_6 (see FAG publ. no. WL 81 115 "Rolling Bearing Lubrication")

If the grease life is considerably shorter than the expected bearing life, either relubrication or a grease exchange is required. Since the fresh grease only partly replaces the used grease when relubricating, the relubrication interval should be shorter than the lubrication interval (normal: 0.5 to $0.7 \cdot t_f$).

A mixture of diverse grease types cannot be ruled out when relubricating. Mixtures of greases with the same thickener can be considered relatively safe. Details on the miscibility of lubricating greases can be found in the FAG publication no. WL 81 115.

▼ Lubrication intervals under favourable environmental conditions. Grease service life F_{10} for standard lithium soap base greases according to DIN 51 825, at 70 °C; failure probability 10 %.



Bearing type		k_f
Deep groove ball bearings	single row	0.9 . . . 1.1
	double row	1.5
Angular contact ball bearings	single row	1.6
	double row	2
Spindle bearings	α = 15°	0.75
	α = 25°	0.9
Four-point bearings		1.6
Self-aligning ball bearings		1.3 . . . 1.6
Thrust ball bearings		5 . . . 6
Angular contact thrust ball bearings	double row	1.4

Bearing type		k_f
Cylindrical roller bearings	single row	3 . . . 3.5*)
	double row	3.5
	full complement	25
Cylindrical roller thrust bearings		90
Tapered roller bearings		4
Barrel roller bearings		10
Spherical roller bearings without lips (E)		7 . . . 9
Spherical roller bearings with centre lip		9 . . . 12

*) for radially and constantly axially loaded bearings; at changing axial load $k_f = 2$

Selection of suitable oil

Mineral oils and synthetic oils are generally suitable for the lubrication of rolling bearings. The mineral-base lubricating oils are used the most frequently. They have to meet the requirements specified in DIN 51 501 at least. Special oils, often synthetic oils, are used for extreme operating conditions or for specific demands on the oil stability. Oil characteristics and the effect of additives are described in the FAG publication no. WL 81 115 "Rolling Bearing Lubrication".

Recommended oil viscosity

The better the contact surfaces are separated by a lubricant film, the longer the attainable life and the more safety against wear. An oil with a high operating viscosity should be selected. A very long life can be reached if the viscosity ratio amounts to $\varkappa = \nu/\nu_1 = 3...4$ (ν = operating viscosity, ν_1 = rated viscosity, see page 42).

High-viscosity oils, however, also have disadvantages. Higher viscosity means more lubricant friction. Problems in supply and drainage of the oil can occur also at low and normal temperatures. An oil viscosity should be selected with which a maximum fatigue life is attained and an adequate supply of oil to the bearings is ensured.

Sometimes, e.g. with slowly rotating gear output shafts, the required operating viscosity cannot be reached. Then an oil with a lower viscosity than the recommended viscosity can be selected. The oil must contain efficient EP additives and its suitability for the application in question must be proved by a test on the FAG test rig FE8. If this is not observed, a reduced fatigue life and wear at the functional areas must be expected (see adjusted life calculation, page 40). The amount of life reduction and wear depends on the deviation from the target value. When mineral oils are particularly highly doped, attention must be paid to compatibility with sealing materials and cage materials (see page 85).

Oil selection according to operating conditions

Under normal operating conditions (atmospheric pressure, maximum temperature of 100 °C at oil sump lubrication and 150 °C at cirulating oil, load ratio $P/C < 0.1$, speeds up to the permissible speed) straight oils can be used but oils with corrosion inhibitors and deterioration inhibitors (letter L in DIN 51 502) are preferable. If the recommended viscosity cannot be maintained, oils with suitable EP additives must be provided.

For high speeds ($k_a \cdot n \cdot d_m > 500\,000$ min^{-1} · mm), an oil should be used which is stable to oxidation, has good defoaming properties, and a positive viscosity-temperature behaviour. In the start-up phase, when the temperature is generally low, high friction due to churning and therefore heating is avoided; the viscosity at the higher steady-state operating temperature is sufficient to ensure adequate lubrication.

If the bearings are subjected to high loads ($P/C > 0.1$) or if the operating viscosity ν is lower than the rated viscosity ν_1, oils with anti-wear additives (EP oils, letter P in DIN 51 502) should be used. The suitability of EP additives varies and usually depends largely on the temperature. Their effectiveness can only be evaluated by means of tests in rolling bearings (FAG test rig FE8).

The selection of oils suitable for high operating temperatures mainly depends on the operating temperature limit and on the V-T behaviour. The oils have to be selected based on the oil properties. Details are given in the FAG publication no. WL 81 115 "Rolling Bearing Lubrication".

Supply of bearings with oil

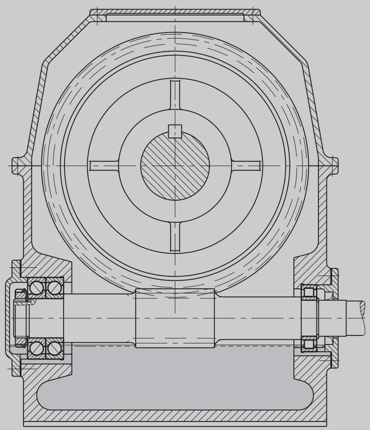
Rolling bearings can generally be provided with oil by means of oil sump lubrication, throwaway lubrication, or circulation lubrication. Unless oil sump lubrication is provided, the oil must be fed to the bearing locations by means of lubricating devices.

In an **oil sump** or, as it is also called, an oil bath, the bearing is partly immersed in oil. When the shaft is in the horizontal position, the bottom rolling element should be half or completely immersed in oil when the bearing is stationary. When the bearing rotates, oil is conveyed by the rolling elements and the cage and distributed over the circumference. For bearings with an asymmetrical cross-section which convey oil due to their pumping effect, oil return holes or ducts should be provided to ensure circulation of the oil. If the oil level rises above the bottom rolling element at high speeds, churning of the oil raises the bearing temperature. The oil level may be higher if the speed index $n \cdot d_m$ is less than 150,000 min^{-1} · mm. Oil sump lubrication is generally used up to a speed index $n \cdot d_m$ = 300,000 min^{-1} · mm. The oil level should be checked regularly.

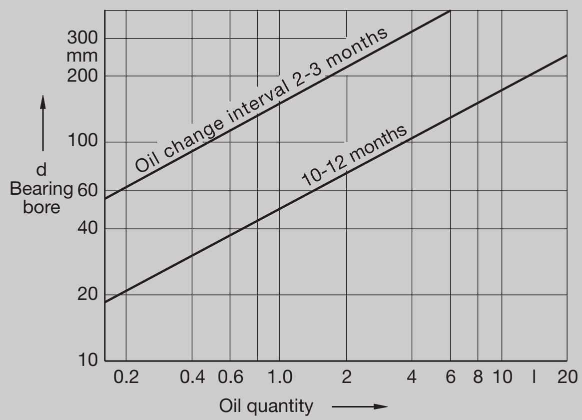
Recommended oil change intervals for normal conditions (bearing temperature up to 80 °C, low contamination) are shown in the upper diagram on page 133. Housings with small oil quantities require frequent oil changes. During the run-in period, an early oil change may be required due to the higher temperature and heavy contamination by wear particles.

In **circulation lubrication**, the oil is fed to an oil collecting tank after passing through the bearings and then returned to them. A filter is a must because contaminants in the lubricating gap may strongly impair the attainable life (see page 40).

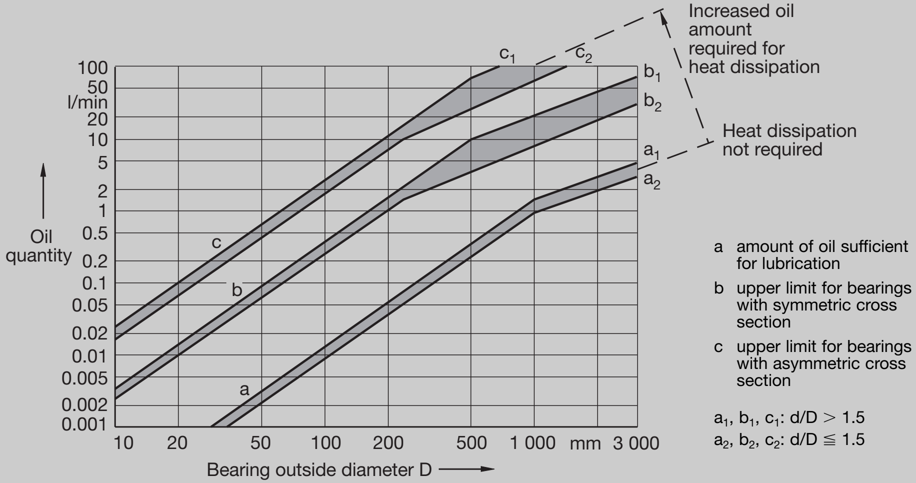
▼ Oil level for sump lubrication



▼ Oil quantity and oil change interval as a function of the bearing bore



▼ Oil quantities for circulation lubrication



The quantity of circulating oil (see lower diagram on page 133) is based on the operating conditions. Due to their conveying effect, higher flow rates are permissible for bearings with an asymmetrical cross section (angular contact ball bearings, tapered roller bearings, spherical roller thrust bearings) than for bearings with a symmetrical cross section. With large quantities small wear particles can be removed or heat dissipated.

Oil is injected into the gap between the cage and bearing ring in fast rotating bearings. Injection lubrication with large quantities of circulating oil means a great loss in energy; keeping the resulting bearing heat at an acceptable level can only be done with a great amount of trouble. The appropriate upper limit of the speed index ($n \cdot d_m = 10^6$ $min^{-1} \cdot mm$ for suitable bearings, e.g. spindle bearings) for circulation lubrication can be well exceeded with injection lubrication.

With **throwaway lubrication,** a low frictional moment and low operating temperature can be reached. The quantity of oil required for the supply to be sufficient depends to a large extent on the bearing type. Thus, double row cylindrical roller bearings for example, need extremely small quantities, bearings with a conveying effect such as angular contact ball bearings need, on the other hand, relatively large quantities, see Publ. No. WL 81 115 also. Speed indices of approximately $1.5 \cdot 10^6$ $min^{-1} \cdot mm$ can be attained.

Rolling bearing storage

Preservation medium and packaging of FAG rolling bearings are designed to retain the bearing properties as long as possible. Certain requirements must therefore be met for storage and handling.

During storage, the bearings must not be exposed to the effects of aggressive media such as gasses, mists or aerosols of acids, alkaline solutions or salts. Direct sunlight should also be avoided because it can cause large temperature variations in the package, apart from the harmful effects of UV radiation. The formation of condensation water is avoided under the following conditions:

– Temperatures +6 to +25 °C, for a short time 30 °C,
– temperature difference day/night ≤ 8 K,
– relative air humidity ≤ 65 %.

Permissible bearing storage periods

With standard preservation, bearings can be stored up to 5 years if the said conditions are met. If this is not the case, shorter storage periods must be taken into consideration.

If the permissible storage period is exceeded, it is recommended to check the bearing for its preservation state and corrosion prior to use. On request, FAG will help to judge the risk of longer storage or use of older bearings.

In special cases, bearings are subjected to a preservation treatment for either longer or shorter storage periods than possible with standard preservation.

Bearings with shields (.2ZR) or seals (.2RSR) on both sides should not be kept to their very limit of storage time. The lubricating greases contained in the bearings may change their chemico-physical behaviour due to aging. Even if a minimum capacity is maintained, safety reserves of the lubricating grease can be reduced (also see following section).

Storage of FAG Arcanol rolling bearing greases (also see page 679)

The storage conditions for rolling bearings apply analogously to Arcanol rolling bearing greases. Supplementary recommendations:

– Temperatures +6 to +40 °C, if possible room temperature,
– closed, filled original containers.

Permissible storage periods for Arcanol rolling bearing greases

– 2 years for lubricating greases of consistency class ≥ 2,
– 1 year for lubricating greases of consistency class < 2.

For these periods, Arcanol rolling bearing greases can be stored at room temperature in closed original containers without quality loss.

The permissible storage time cannot be regarded as a rigid limit. As compounds of oil, thickener, and additives, rolling bearing greases may change their chemico-physical properties during storage and should therefore be soon used. At careful storage, that is, observing all conditions described, low room temperature, full and airtight containers, most rolling bearing greases can be used even after 5 years if minor changes are accepted.

Higher temperatures and only partly filled containers should be avoided because they promote separation of the base oil from the grease. In case of doubt, a grease should be inspected chemicophysically for alterations. On request, FAG will help to judge the risk of longer storage or use of older lubricating greases.

When opened containers are to be kept in storage, the grease surface should, in any case, be smoothed, the container closed airtight and stored with the hollow space on top.

Cleaning contaminated bearings

Petroleum ether, petroleum, ethyl alcohol, dewatering fluids, aqueous neutral, and alkaline cleaning agents can be used to clean rolling bearings. It should be remembered that petroleum, petroleum ether, ethyl alcohol and dewatering fluids are inflammable and alkaline agents are caustic.

There is a risk of fire, explosion, and decomposition when using chlorinated hydrocarbons as well as a health hazard. These risks and appropriate protective measures are described in detail in the Commercial Trade Association's instruction leaflet ZH1/425.

Paint brushes, brushes or lint-free cloths should be used for cleaning. Immediately after cleaning and the evaporation of the solvent, which should be as fresh as possible, the bearings must be preserved in order to avoid corrosion. Precleaning by hand and treatment with an aqueous, strong alkaline cleansing agent is advisable when the bearings contain gummed oil or grease residues.

Mounting and dismounting

Rolling bearings are heavy-duty machine elements with high precision. In order to fully utilize their capacity, mounting and dismounting should be taken into consideration when selecting the bearing type and design and when designing the surrounding structure.

For the rolling bearings to reach a long service life, the use of suitable mounting aids as well as utmost cleanliness and care at the assembly site are essential requirements. The mechanical, thermal*) and hydraulic methods for mounting and dismounting bearings of diverse types and sizes can be taken from the chart on page 136. Fundamental aspects on mounting and customary mounting procedures are explained later on.

Further details on mounting and dismounting are contained in the FAG publication WL 80 100 "Mounting and Dismounting Rolling Bearings".

The relevant FAG programme is contained in the FAG publication WL 80 200 "Methods and Equipment for the Mounting and Maintenance of Rolling Bearings".

For many years FAG have been offering an efficient damage diagnosis as a service. With portable electronic FAG measuring devices the user can himself provide for condition-related maintenance of machines and plants, also see Section "FAG services programme" on page 685 et seq.

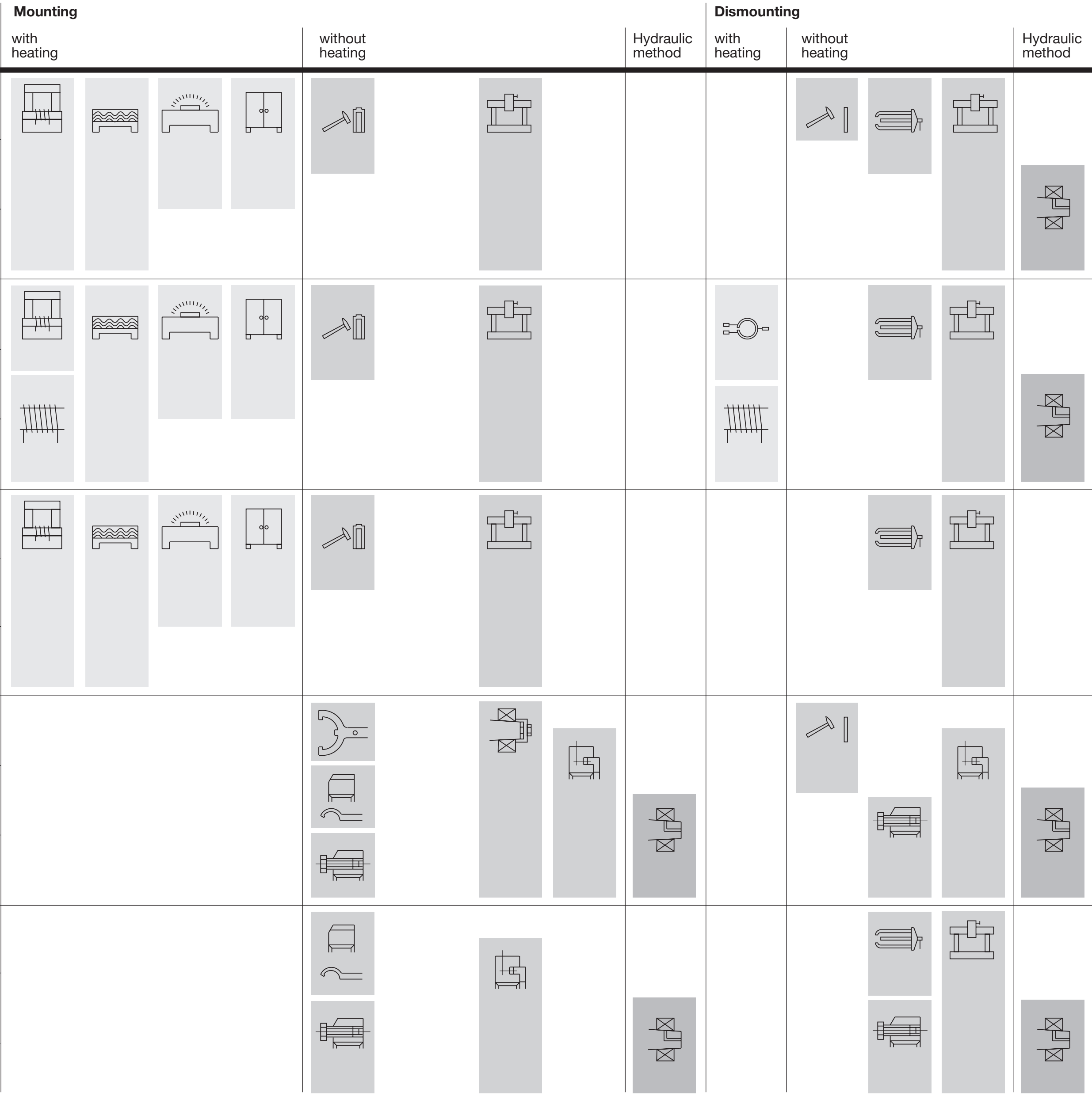
*) If, for example, a temperature of about 300 °C or more is reached when dismounting a bearing with a welding torch, fluorinated materials can release gasses and fumes which are a danger to health. FAG use fluorinated materials for seals made of fluorocaoutchouc (FKM, FPM, e.g. Viton®) or for fluorinated lubricating greases such as the FAG rolling bearing grease Arcanol L79V, for instance. If the high temperatures cannot be avoided the applicable safety data sheet for the fluorinated material in question must be observed. It is available on request.

Mounting and Dismounting

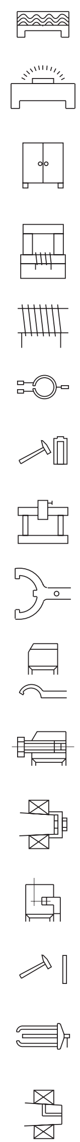
Synoptic table: Tools and methods

▼ Synoptic table: Tools and methods for mounting and dismounting rolling bearings · Symbols

Bearing type	Bearing bore	Bearing size
Deep groove ball bearing; Angular contact ball bearing; Four-point bearing; Self-aligning ball bearing; Tapered roller bearing; Barrel roller bearing; Spherical roller bearing	cylindrical	small
		medium
		large
Cylindrical roller bearing	cylindrical	small
		medium
		large
Thrust ball bearing; Angular contact thrust ball bearing; Cylindrical roller thrust bearing; Spherical roller thrust bearing	cylindrical	small
		medium
		large
Self-aligning ball bearing; Self-aligning ball bearing with adapter sleeve; Barrel roller bearing; Barrel roller bearing with adapter sleeve; Spherical roller bearing; Spherical roller bearing with adapter sleeve; Spherical roller bearing with withdrawal sleeve; Adapter sleeve; Withdrawal sleeve	tapered	small
		medium
		large
Cylindrical roller bearing, double row	tapered	small
		medium
		large



Symbols



Mounting and Dismounting

Preparations · Mounting bearings with cylindrical bore and O.D.

Mounting and dismounting preparations

FAG publications WL 80 100 "Mounting and Dismounting Rolling Bearings" and WL 80 200 "Methods and Equipment for the Mounting and Maintenance of Rolling Bearings" contain details on mounting and dismounting.

The shop drawing is studied prior to mounting to become familiar with the design. The order of the individual work steps is schematically laid down including the required heating temperatures, mounting forces, and grease quantities. For big jobs, the fitter should be supplied with mounting instructions in which each step is accurately described. The instructions also include details on transportation means, mounting equipment, measuring tools, lubricant type and quantity, and a precise description of the mounting procedure.

Before mounting, the fitter has to check whether the bearing to be mounted corresponds to the data on the drawing. This requires basic knowledge on the structure of the rolling bearing code numbers, see section "Bearing design", page 50.

The anti-corrosion agent of the packed FAG rolling bearing has no effect on the standard greases which are most commonly used (lithium soap base greases on a mineral oil base) and does not have to be washed out prior to mounting. It is only wiped off the seats and mating surfaces.

The anti-corrosion agent should, however, be washed out of tapered bearing bores in order to guarantee a tight fit on the shaft or sleeve, cf. Section: "Cleaning contaminated bearings", page 135.

Rolling bearings must be protected from dirt and humidity under all circumstances so as to avoid damage to the running areas. The work area must therefore be clean and free of dust. It should not be near grinders and the use of compressed air is to be avoided. Shafts and housings must be clean. Anti-rust compounds and paint residues are to be removed from the seats and castings freed from sand. Turned parts must be free from burrs and sharp edges.

All surrounding parts are carefully checked for dimensional and form accuracy prior to assembly.

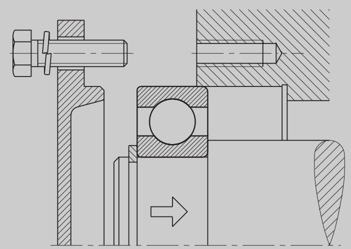
Mounting bearings with cylindrical bore and O.D.

Blows with the hammer applied directly to the bearing rings must be avoided completely. In the case of non-separable bearings the mounting forces are applied to the ring which is to have a tight fit and which is first mounted. The rings of separable bearings however, can be mounted individually.

Bearings with a maximum bore of approximately 80 mm can be mounted cold. The use of a mechanical or hydraulic press is recommended. Should no press be available, the bearing can be driven on with hammer and mounting sleeve. The FAG mounting tool set 172013 would be suitable for this (see FAG publ. no. WL 80200). For self-aligning bearings, misalignment of the outer ring can be avoided by means of a disk which abuts both bearing rings. In bearings where the cage or balls project laterally (e.g. some self-aligning ball bearings), the disk must be relieved.

Bearings with a cylindrical bore for which tight fits on a shaft are specified and which cannot be pressed mechanically onto the shaft without great effort, are heated before mounting. The chart on page 139 shows the heat-up temperature [°C] required for easy mounting as a function of the bearing bore d. The data applies to the maximum interference, a room temperature of 20 °C plus 30 K to be on the safe side.

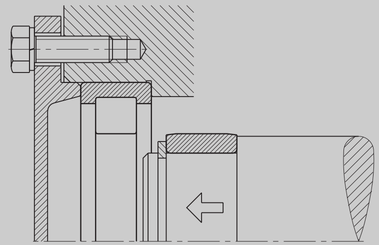
▼ If the inner ring of a non-separable bearing gets the tight fit, the bearing is pressed onto the shaft. The bearing is then pushed with the shaft into the housing (loose fit).



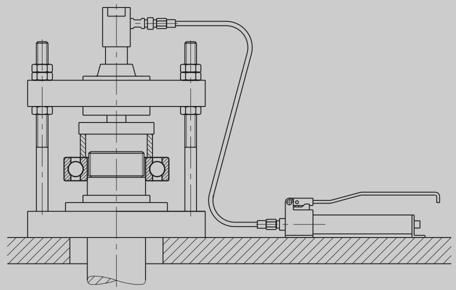
Mounting and Dismounting

Mounting bearings with cylindrical bore and O.D.

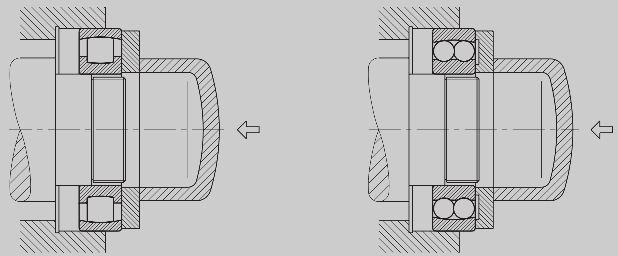
▼ The rings of cylindrical roller bearings are mounted separately (tight fits).



▼ Deep groove ball bearing mounted with a hydraulic press

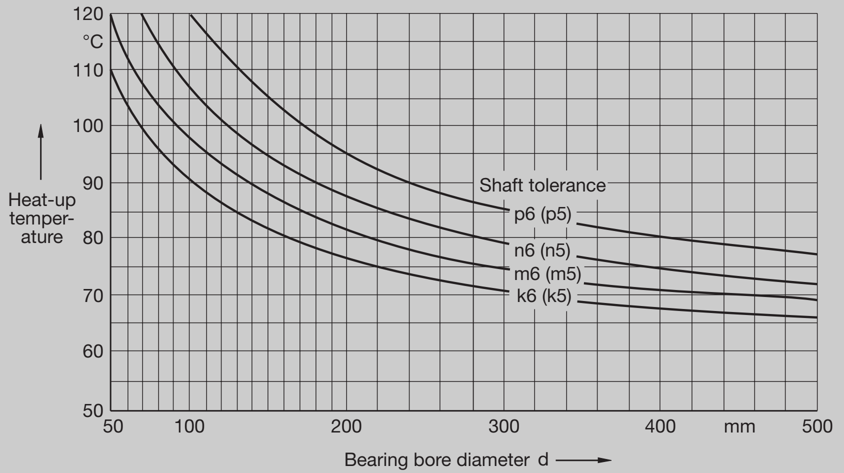


▼ Simultaneous pressing bearings on the shaft and pushing in the housing with the aid of
a) an unrelieved mounting disk for barrel roller bearings and
b) a relieved mounting disk for some self-aligning ball bearings



a b

▼ Diagram for determining the heat-up temperature



Induction heating devices are particularly suitable for fast, safe and clean heating. The devices are used above all for batch mounting. FAG offer six induction heating devices. The smallest device AWG.MINI is used for bearings with 20 mm bores upwards. The maximum bearing mass is about 20 kg. The field of application of the largest device AWG40 starts at 85 mm bores. The maximum bearing mass may amount to approximately 800 kg. See FAG publication TI no. WL 80-47 for description.

Induction heating devices are used for extracting and shrinking on the inner rings of cylindrical roller bearings from 100 mm bores upwards which have either no lip or an integral one. See publ. no. WL 80 107 "FAG Induction Heating Equipment" for details.

Individual bearings can be heated provisionally on an electric heating plate. The bearing is covered with a metal sheet and turned several times. A thermostatic control is an absolute must, such as the FAG heating plates 172017 and 172018 have (see FAG publ. no. WL 80200).

A safe and clean method of heating rolling bearings is to use a thermostatically controlled hot air or heating cabinet. It is used mainly for small and medium-sized bearings. The heat-up times are relatively long.

Bearings of all sizes and types can be heated in an oil bath except for sealed and greased bearings as well as precision bearings. A thermostatic control is advisable (temperature 80 to 100 °C). The bearings are placed on a grate or hung up for them to heat uniformly. Disadvantages: accident hazard, pollution of the environment by oil vapours, inflammability of hot oil, danger of bearing contamination.

Mounting tapered bore bearings

Rolling bearings with a tapered bore are either fitted directly onto the tapered shaft seat or onto a cylindrical shaft with an adapter sleeve or a withdrawal sleeve. By driving up the inner ring on the shaft or sleeve, the tight fit required is obtained and is measured by checking the radial clearance reduction due to the expansion of the inner ring or by measuring the axial drive-up distance. See page 368 for radial clearance reduction values and the drive-up distance for spherical roller bearings. The FAG 172031 and 172032 feeler gauges are suitable accessories for measuring the radial clearance.

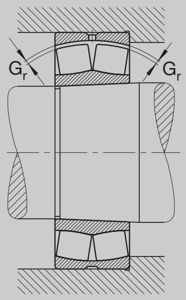
Small bearings (up to approx. 80 mm bore) can be pressed with a locknut onto the tapered seat of the shaft or the adapter sleeve. A hook spanner is used to tighten the nut. Suitable spanners of the series FAG HN can be taken from publ. no. WL 80 200. Small withdrawal sleeves are also pressed with a locknut into the gap between the shaft and inner ring bore.

Considerable force is required to tighten the nut with medium-sized bearings. Locknuts with thrust bolts facilitate mounting in such cases (not suitable for FAG spherical roller bearings of E design).

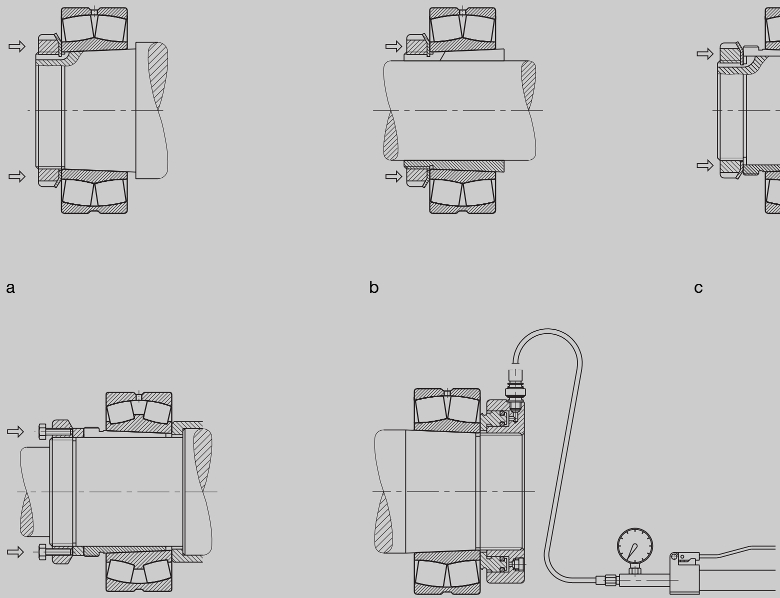
It is advisable to use a hydraulic press for driving-up larger bearings or pressing them onto the sleeve. Hydraulic nuts are available for all popular sleeve and shaft threads (cf. publ. no. WL 80 103 "FAG Hydraulic Nuts").

For bearings with a bore of approximately 160 mm and upwards mounting and especially dismounting are greatly facilitated by the hydraulic method, (cf. page 142, detailed description in publ. no. WL 80102 "How to Mount and Dismount Rolling Bearings Hydraulically"). An oil with a viscosity of ≈ 75 mm^2/s at 20 °C (nominal viscosity at 40 °C: 32 mm^2/s) is recommended for mounting.

▼ With spherical roller bearings the radial clearance (G_r) must be measured across both roller rows



▼ Mounting tapered bore bearings
a) on a tapered shaft with a locknut
b) on an adapter sleeve with the adapter sleeve nut
c) on a withdrawal sleeve with the locknut
d) on a withdrawal sleeve with locknut and thrust bolts
e) on a tapered shaft with a hydraulic nut



a b c d e

Dismounting bearings with cylindrical bore and O.D.

If the bearings are to be used again the extraction tool should be applied to the tightly fitted bearing ring. With non-separable bearings, one should proceed as follows: if the outer ring is tightly fitted, the bearing and the housing are removed from the shaft and then the bearing is extracted from the housing by pressing off the outer ring. If the inner ring is tightly fitted, the shaft with the bearing is removed from the housing and then the inner ring pressed off.

Mechanic extractors or hydraulic presses are suitable for extracting small bearings. Dismounting is facilitated when there are extraction slots on the shaft and housing. The extraction tool can then be applied directly to the tightly-fitted ring. Special devices are available if there are no extraction slots.

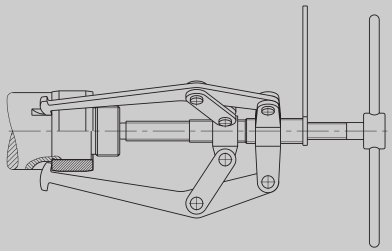
Induction heating devices are chiefly used for extracting the shrunk-on inner rings of cylindrical roller bearings. Heating occurs rapidly and the rings easily loosen without much heat reaching the shaft.

The bearings can also be pressed off cylindrical seats with the aid of the hydraulic method (see page 143).

Heating rings of light metal with radial slots are used when dismounting the inner rings of cylindrical roller bearings which either have no lip or just one integral lip. The heating rings are heated to 200 – 300 °C with an electric heating plate, placed around the bearing ring to be removed and clamped by means of the handles.

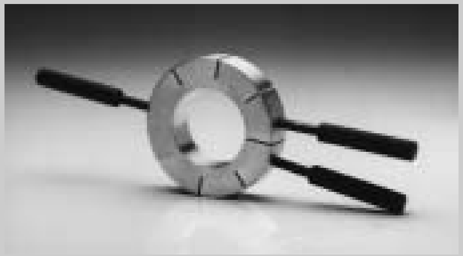
When the tight inner ring fit on the shaft is loosened, withdraw both rings together. The bearing ring must be removed immediately from the heating ring to avoid overheating. If an inductive device is not available and there are no oil ducts for the hydraulic method, the inner rings of separable bearings can be heated by a flame if necessary – preferably with a ring burner. Great care is required because the rings are sensitive to non-uniform heating and local overheating.

▼ Extracting device with three adjustable arms for withdrawing separable bearings



▼ Induction heating device for removing the inner rings of cylindrical roller bearings

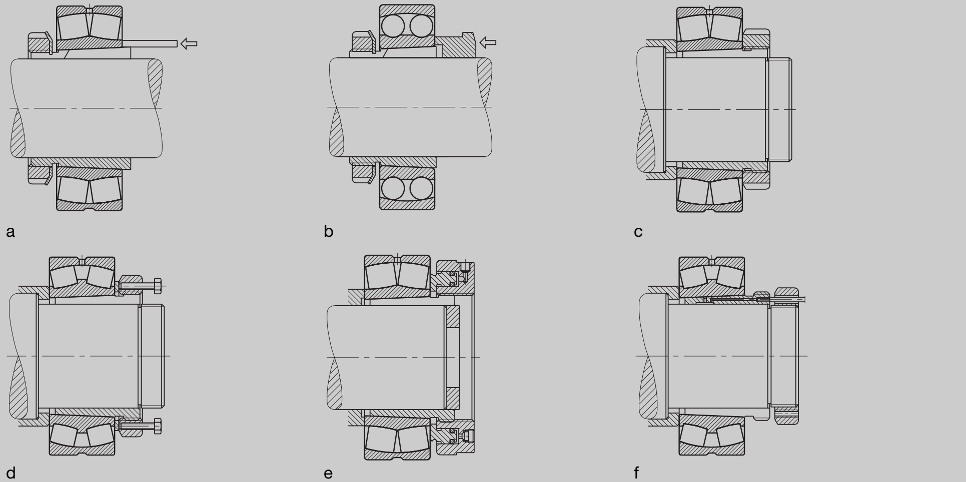
▼ Heating rings are suitable for dismounting the inner rings of cylindrical roller bearings



Dismounting bearings with tapered bore

When the bearings are directly on the tapered seat or an adapter sleeve, the locking device of the shaft or sleeve nut is loosened first. The nut is then turned back by the amount corresponding to the drive-up distance. The inner ring is then driven off the sleeve or the shaft by means of a hammer and piece of tubing. When a press is used the adapter sleeve is supported and the bearing pressed off.

▼ Dismounting tapered bore bearings
a) Dismounting a spherical roller bearing with an adapter sleeve. The inner ring is driven off the sleeve by means of a metal drift.
b) Dismounting a self-aligning ball bearing with an adapter sleeve. The use of a piece of tubing prevents damage to the bearing.
c) Dismounting a withdrawal sleeve with an extraction nut.
d) Dismounting with nut and thrust bolts applied to the inner ring via a washer.
e) Dismounting a withdrawal sleeve with a hydraulic nut. The projecting withdrawal sleeve is supported by a thick-walled ring.
f) Dismounting a spherical roller bearing from the withdrawal sleeve with the hydraulic method. Oil is pressed between the surfaces. The withdrawal sleeve is released abruptly. Nut is left on the shaft.



Withdrawal sleeve mounted bearings are removed by means of the extraction nut. High forces are required for large-size bearings. Extraction nuts with additional thrust bolts are then used. A washer is inserted between the inner ring and thrust bolts.

The dismounting of withdrawal sleeves is much easier and more cost-effective with hydraulic nuts.

The hydraulic method is applied to facilitate the dismounting of large-size bearings. Oil is injected between the mating surfaces and enables the mating parts to be moved separately without risking surface damage.

Tapered shafts must be provided with oil grooves and supply bores. Oil injectors are sufficient for the generation of pressure.

Large adapter and withdrawal sleeves already have the necessary grooves and bores. The required oil pressure has to be generated with a pump.

When dismounting, an oil with a viscosity of about 150 mm^2/s at 20 °C is used (nominal viscosity: 46 mm^2/s at 40 °C). Fretting corrosion can be dissolved by adding rust-removing additives to the oil.

For tapered bore bearings, oil is pressed between the mating surfaces. Since the press fit is released abruptly, a stop such as a nut should be provided to control the movement of the bearing.

▼ Position of oil grooves for hydraulically dismounting a spherical roller bearing from the tapered shaft seat