1. INTRODUCTION

Dunlop BTL based in Kent & Durham UK has the enviable reputation of being one of Europe's leading manufacturers and international distributors of Bearings, Transmission & Linkages for the agricultural, automotive, construction, industrial and motor sport industries. With after market products being available throughout Europe and the rest of the world via a network of high quality knowledgeable authorised distributors.

Dunlop BTL is proud to be a committed European manufacturer of Bearings, Transmissions and Linkages. We believe in the future of European manufacturing and will continue to focus and further enhance the requirements and expectations of our customers globally.

New UK Production Facility
During 2016 UK production moved to a newly refurbished manufacturing facility this has enabled further expansion and additional production machinery as well as additional production staff. This state-of-the-art UK site is based in Consett, Co. Durham UK and is ideally located with excellent transport links to Europe.

All of our manufacturing facilities use the latest CNC production machinery available. Our capacity includes CNC lathes and milling machines, multi-spindle auto lathes, automatic and robotic machine loaders and countless ancillary machines, giving us high volume, precision component production.

In addition to our catalogue ranges of Bearings, Transmission and Linkages we produce non-standard items to suit individual customer requirements, 40% of our total production is for bespoke products to specific customer design.

Investment
Investment is constant, continually expanding and upgrading our portfolio of production facilities and machinery globally.

Additional investment also extends to our staff, in addition to having highly skilled engineers and manufacturing staff with years of experience we also believe in training our younger employees, this we feel is key to the company's future growth. We invest in their future by means of an apprenticeship scheme, many of our skilled engineers have come through our own apprenticeship program.

This we believe, combines traditional engineering values with modern production techniques.

The catalogue uses units in accordance with ISO.

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2. BEARING TERMINOLOGY

An illustrative description of terms that characterize individual types of bearings can be seen in the following pictures.

2.1 Radial bearings (fig. 2.1 and 2.2)

1 Inner race
2 Outer race
3 Rolling element – ball, cylindrical roller, spherical roller, tapered roller
4 Cage
5 Seal, shield
6 Outer cylindrical bearing surface
7 Bearing bore
8 Cylindrical surface of inner ring flange
9 Cylindrical surface of outer ring flange
10 Snap ring groove
11 Snap ring
12 Outer ring face
13 Seal groove
14 Outer ring raceway
15 Inner ring raceway
16 Bearing seal recess
17 Inner ring face
18 Installation fillet
19 Bearing mean diameter
20 Bearing width
21 Guiding flange
22 Support flange
23 Contact angle

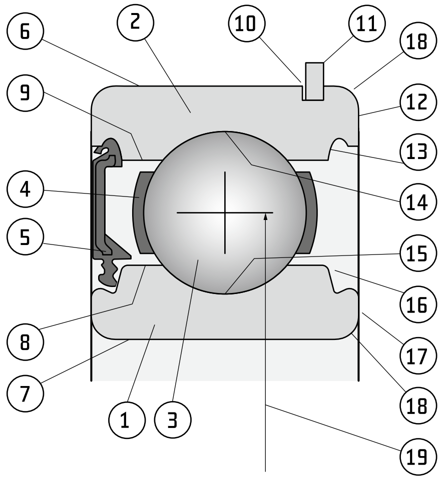

Fig. 2.1

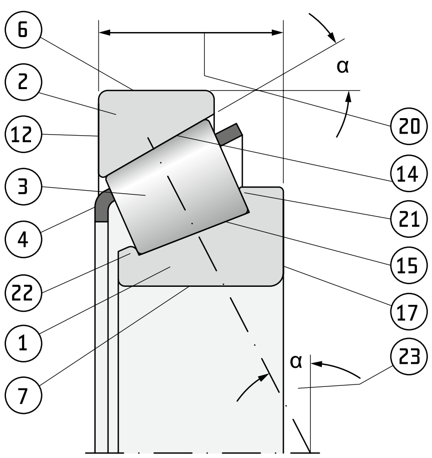

Fig. 2.2

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2.2 **Radial bearings** (fig. 2.3 to 2.5)

1 Shaft ring
2 Cage with rollers
3 Housing ring
4 Housing ring with spherical bearing surface
5 Spherical housing ring

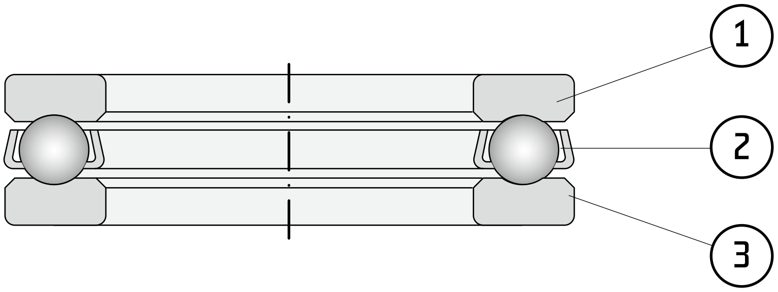

Fig. 2.3

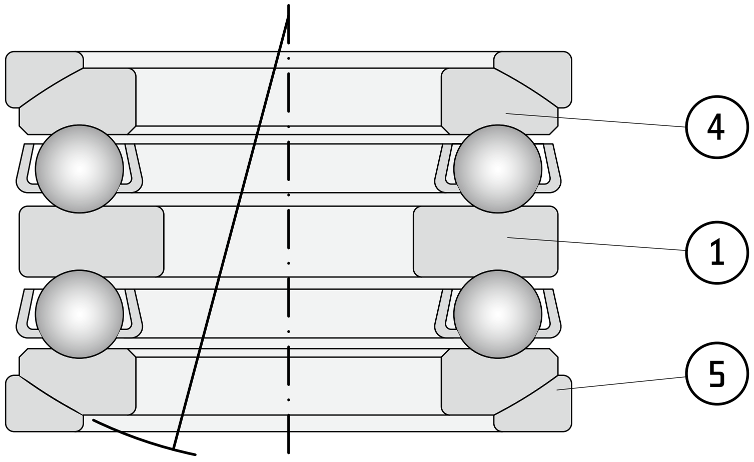

Fig. 2.4

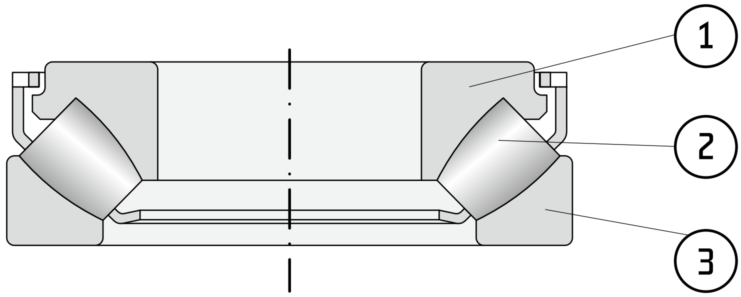

Fig. 2.5

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3. CRITERIA FOR SELECTION AND USE OF BEARINGS

Rolling-contact bearings are an indispensable component of machinery, which are constantly subjected to the process of innovation. They enable mutual rotational motion of machine parts, while simultaneously transfe-rring acting forces. They usually consist of two rings, roller-bearing cases, and a cage. Grease and packing elements are also an integral component of rolling-contact bearings. Proper rolling-contact bearing operation thus requires not only the selection of the proper type and size of bearing, but also the appropriate method of lubrication, heat dissipation, corrosion protection, and design to prevent entry of contaminants into the housing. The housing design a well as bearing connection dimension tolerances and supplemental lubrication method must be adequate. The correct installation, disassembly or de-installation procedure must also be designated to ensure proper bearing operation. A service manual and maintenance instructions should be provided in cases of complicated housing designs and where high operating reliability are needed.

These principles must particularly be observed in housings in which bearing price, high reliability, or costs associated with bearing installation and economic losses due to shutdown of equipment play a significant role. Such housings require a highly qualified approach in the design phase with the use of computations and testing.

3.1 Types of rolling-contact bearings

Dunlop BTL manufactures a full range of bearings, from which the designer can choose the bearings that best meet the specific requirements.

3.1.1 Based on load direction

Rolling-contact bearings are generally divided according to the direction of force, for whose transfer they are predominantly designed, into two basic groups:

Radial bearings
Axial bearings

There is no exact difference between the two groups, however, because the majority of radial bearings can also capture axial forces and certain types of axial bearings also radial forces. This division, however, is important for determining the load-bearing capacity of bearings. The load-bearing capacity in radial bearings specifies the magnitude of radial forces, whereas in axial bearings the value refers to axial forces.

We divide bearings, according to shape, into ball (single-point contact) bearings and roller (line contact) bearings. Contact in ball bearings theoretically occurs at a single point, hence the designation “single-point contact bearings.” In roller, spherical, tapered roller, and needle roller bearings, contact occurs in a line or straight line, resp., thus they are commonly designated as straight-line (vector) or line-contact bearings.

The following overview provides a classification of individual bearing types based on this characteristic.

Single-point contact bearings

Single-row ball bearings . . . .(fig. 3.1)
Single-row angular-contact ball bearing . . . .(fig. 3.2)
Double-row angular-contact ball bearing . . . .(fig. 3.3)
Four-point contact bearing .. . .(fig. 3.4)

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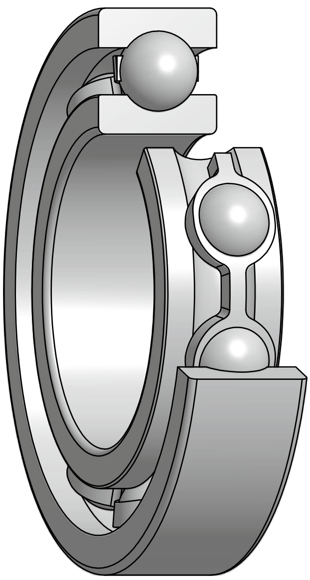
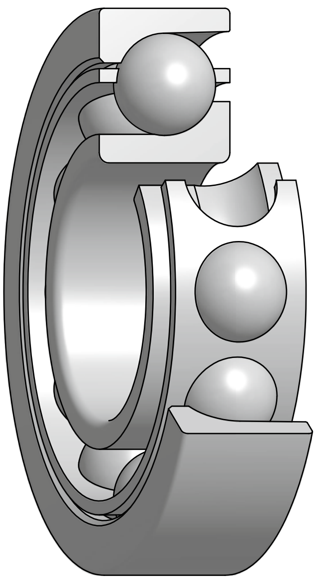
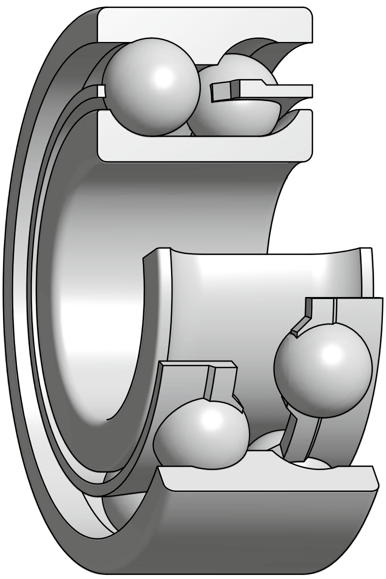


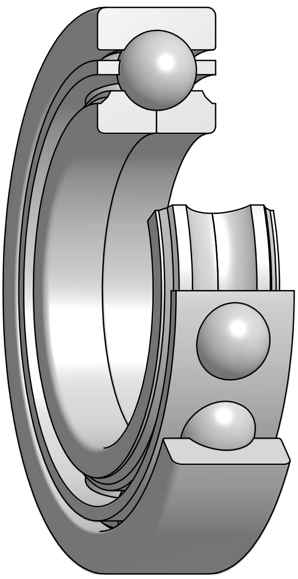
Single-row ball bearings (fig. 3.1)



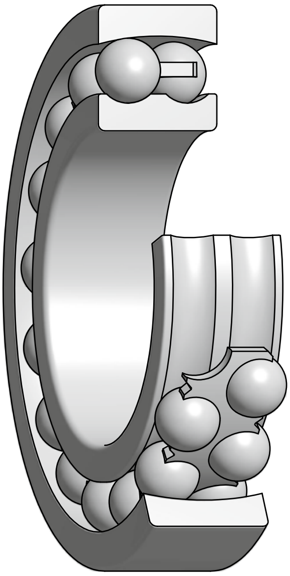
Single-row angular-contact ball bearing (fig. 3.2)



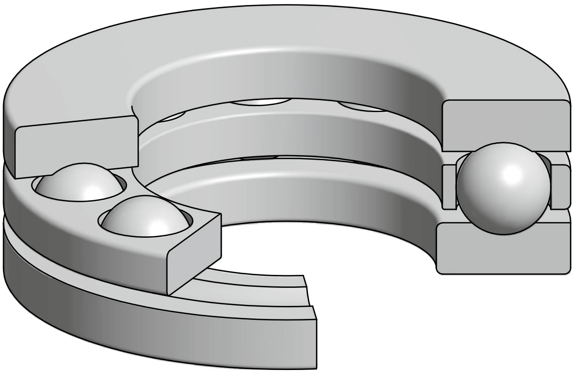
Double-row angular-contact ball bearing (fig. 3.3)



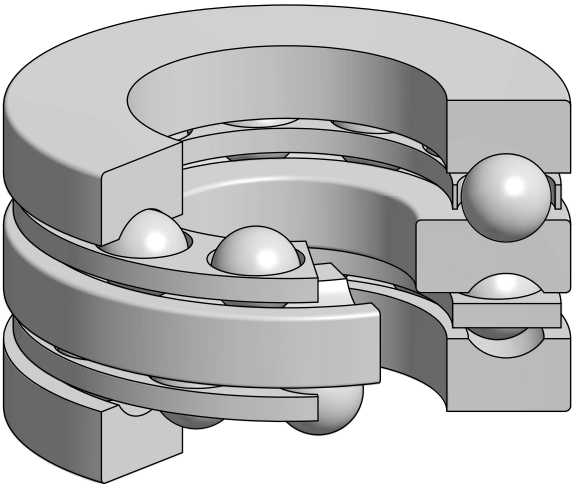
Four-point contact bearing (fig. 3.4)



Double-row, self-aligning ball bearing (fig. 3.5)



Single direction thrust ball bearings (fig. 3.6)



Double direction thrust ball bearings (fig. 3.7)

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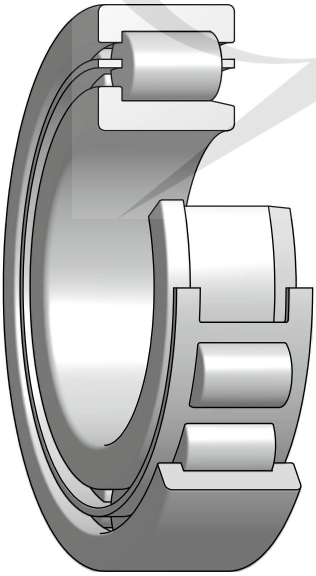
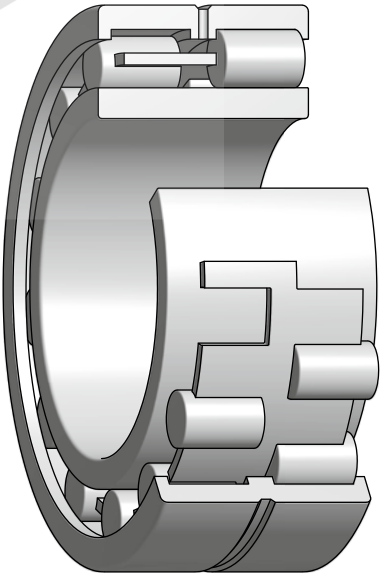
Double-row, self-aligning ball bearing	(fig. 3.5)
Single direction thrust ball bearings	(fig. 3.6)
Double direction thrust ball bearings	(fig. 3.7)
Line-contact bearings Single row cylindrical roller bearing	(fig. 3.8)
Double row cylindrical roller bearing	(fig. 3.9)
Single row full complement cylindrical roller bearing	(fig. 3.10)
Double row full complement cylindrical roller bearing	(fig. 3.11)
Tapered roller bearing	(fig. 3.12)
Double row tapered roller bearing	(fig. 3.13)
Double row spherical roller bearing	(fig. 3.14)
Thrust cylindrical roller bearing	(fig. 3.15)
Thrust spherical roller bearing	(fig. 3.16)

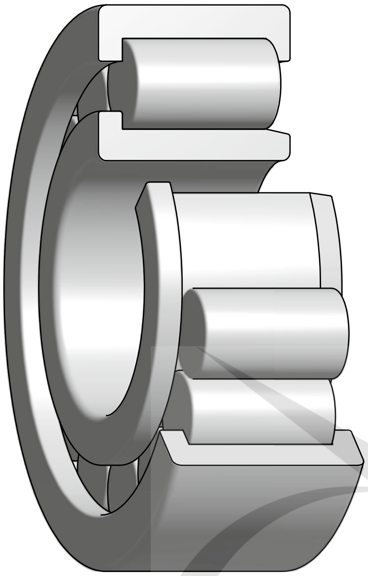
We separate each type of roller bearing then into several types according to dimensions and design variati-ons. Specific information on characteristics of individual types of bearings is available in the sections of text provided before the tables of individual bearings.



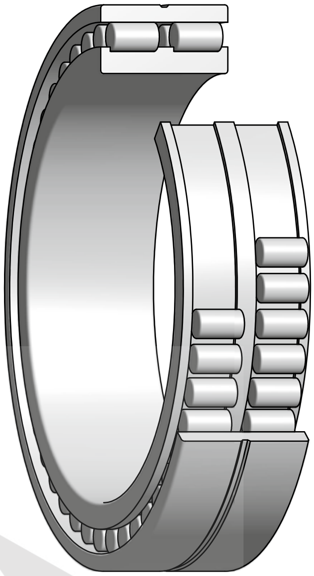
Single row cylindrical roller bearing (fig. 3.8)



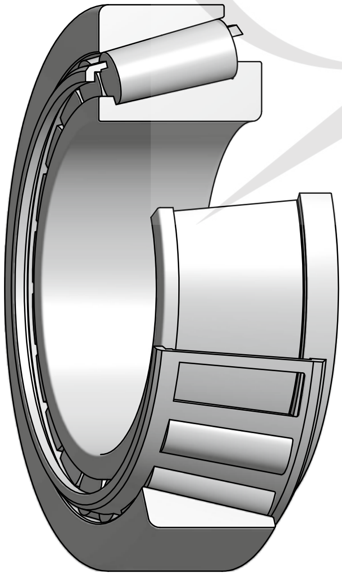
Double row cylindrical roller bearing (fig. 3.9)



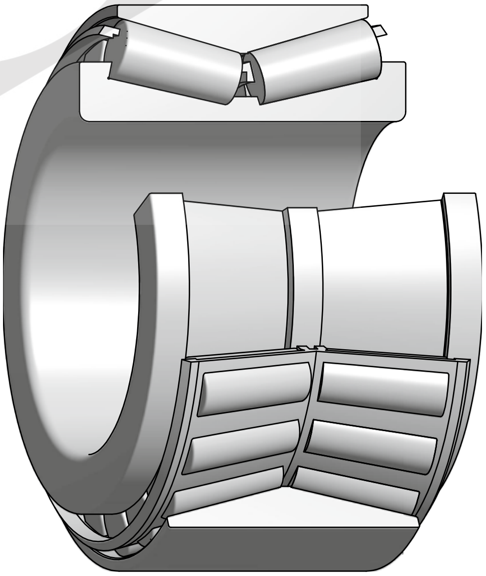
Single row full complement cylindrical roller bearing
(fig. 3.10)



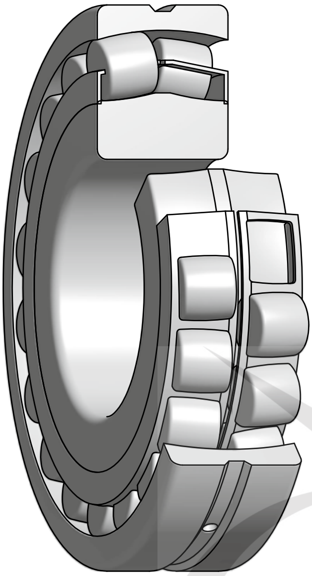
Double row full complement cylindrical roller bearing
(fig. 3.11)



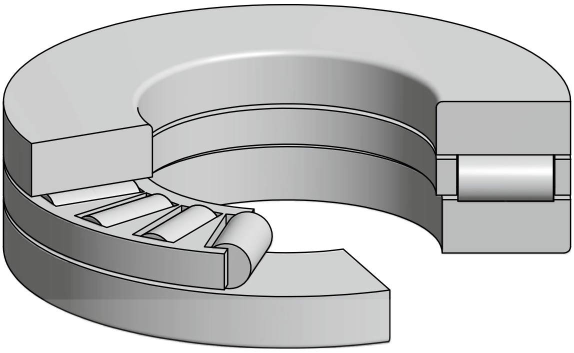
Tapered roller bearing
(fig. 3.12)



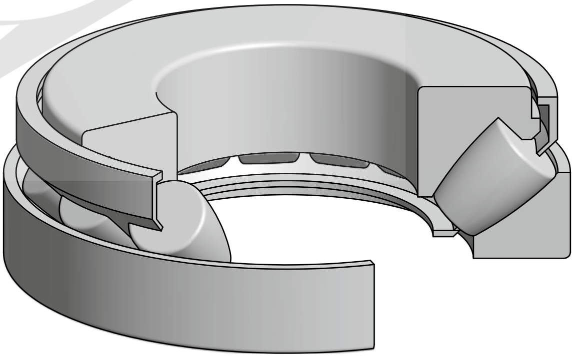
Double row tapered roller bearing
(fig. 3.13)



Double row spherical roller bearing
(fig. 3.14)



Thrust cylindrical roller bearing
(fig. 3.15)



Thrust spherical roller bearing
(fig. 3.16)

3.1.2 Separable and non-separable bearings

Separable bearings allow separate installation of both rings, which is of particular advantage when installing both rings with an overlap. Sequential installation of individual parts can also be used in certain complex housings and assembly units. Separable bearings are, e.g. four-point contact bearings, double-row ball bea-rings with split inner ring, roller bearings, tapered roller bearings, thrust ball bearings, thrust roller bearings, and spherical roller thrust bearings. In contrast, non-separable bearings include, e.g. single row ball bearings, single row angular-contact ball bearings, self-aligning ball bearings, and double-row spherical roller bearings.

3.2 Criteria for selecting bearings

The Dunlop BTL production program offers a full range of bearings, from which the designer can choose the bea-rings that best meet the specific requirements. The bearing type and size are generally chosen according to its loading capacity with consideration to its operating conditions and expected bearing service life. To determine the proper type of bearing thus requires a thorough knowledge of the loading capacity of the bearing during operation. Proper principles for selecting, fitting, and installing them must be followed, but it also requires knowledge of the prerequisites for which the proposed results apply. In the following chapters, we thus present general principles for selecting and using contact-roller bearings, which may be used by drafting engineers in the bearing design process. The chapters are organized in logical consecutive order. The technical part of the publication contains important regarding calculations, design data, housing, lubrication designs, as well as installation and removal information on rolling-contact bearings. The table provides a list of currently manufactured Dunlop BTL rolling-contact bearings with main dimensions and functional parameters.

Even though they list detailed information, this publication is unable to provide full information on all housings for their wide varieties of application. We therefore recommend that complex housing designs be consulted with Dunlop BTL technical and consultation service specialists.

4. SELECTING TYPE OF BEARING

Each type of bearing is characterized by specific properties unique to the given design and dimens ions, which determine its suitability for the given type of application. Ball bearings for example are characterized by low friction and low noise. They are designed for translating medium-large radial as well as axial loads. They may be manufactured at higher precision enable them to operate at higher rpms. Due to their properties and affor-dability, they are among the most common types of bearings used. In contrast, spherical-roller bearings are designed for housings under high loads and are capable of compensating to a certain extent misalignments. They are thus particularly suitable for industrial use. It is thus important, when selecting the type of bearing, to consider various influences and to evaluate them according to their measure of importance for the given housing. The selection of a standard bearing is influenced particularly by:

- Load
- Available space
- Revolutions
- Precision of operation
- Alignment
- Slide-able axial movement
- Housing rigidity
- Installation and de-installation options
- Sealing methods

4.1 Loads

4.1.1 Radial loads

Bearings designed primarily for transferring radial loads are called radial bearings (fig. 4.1). They have a nominal contact angle of $\alpha \leqslant 45°$. Line contact bea-rings are more suitable for higher radial loads than single-point contact bearings, and bearings with a full number of rolling bodies have a higher load capacity than corresponding bearings with a cage.

Ball bearings are designed for small and medium--large loads. N- and NU-type ball bearings can only be burdened radially. Different type radial bearings can transfer both radial as well as axial loads.

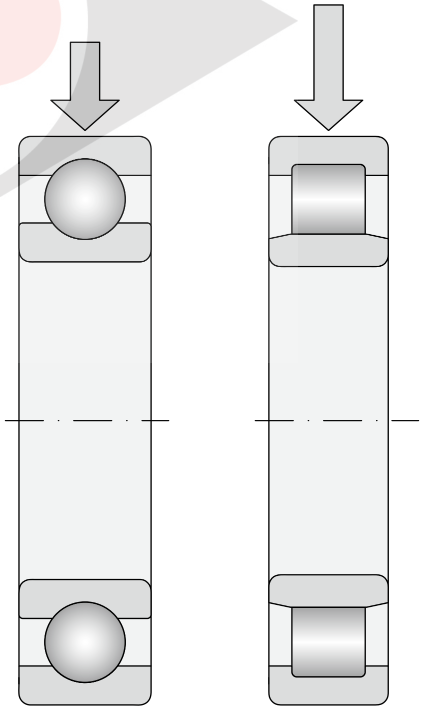


Fig. 4.1

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4.1.2 Axial loads

Bearings designed mainly for axial loads (thrust ball bearings) have a contact angle α>45°.

Axial ball bearings and angular contact thrust ball bearings may, depending on the design, transfer axial loads in one or both directions (fig. 4.2a). In cases of extremely high axial loads, a thrust cylindrical roller or thrust roller bearings (fig. 4.2b). Other thrust bearings are only suitable for axial loads. Double direction bearings are designed for bi-directional axial loads.

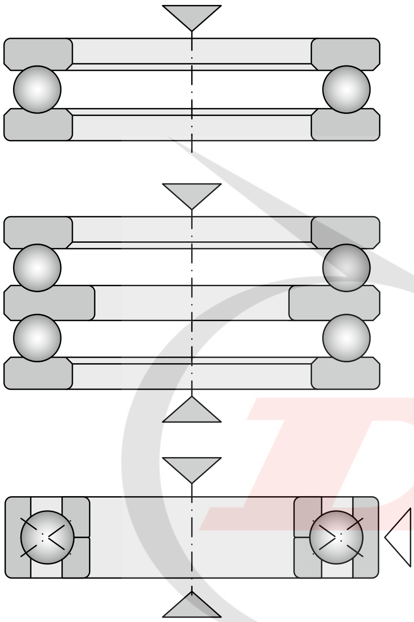

Fig. 4.2a

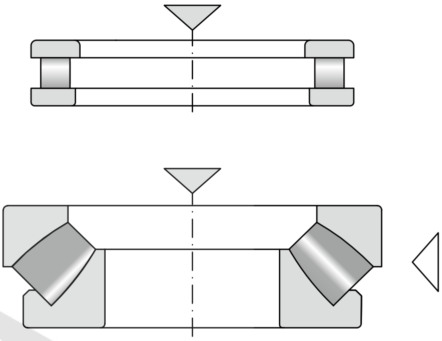

Fig. 4.2b

4.1.3 Combined loads

Combined loads are composed of simultaneously acting radial and axial loads.

Axial load capacity of a bearing depends on the angle of contact. The larger the angle, the larger the axial load bearing capacity of the bearing. Larger axial clearance in single row ball bearings increases their load bearing capacity. Single and double row angular contact ball bearings or tapered roller bearings are best for capturing combined loads (fig. 4.3a). Combined loads can also be borne by double row spherical roller bearings, thrust ball angular-contact bearings, and to a limited extent, also spherical roller thrust bearings. Self-aligning ball bearings, NJ, NUP, or NJ roller-contact bearings and NU bearings with HJ attachment rings (fig. 4.3b) can be used for combined loads with a relatively small axial component.

Single row angular contact ball bearings, tapered roller bearings, NJ roller-contact bearings, and NU+HJ and axial spherical roller bearings can only transfer unidirectional axial loads. If the arrangement of the active load changes, an additional bearing must be used. Combined single row angular contact ball bearings or single row tapered roller bearings are provided for best capturing such combined loads.

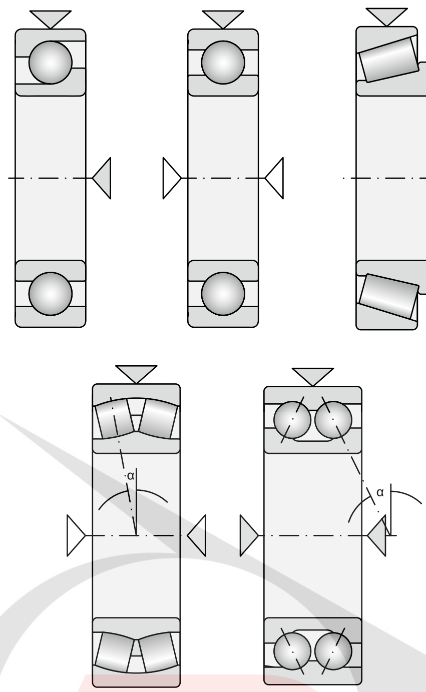

Fig. 4.3a

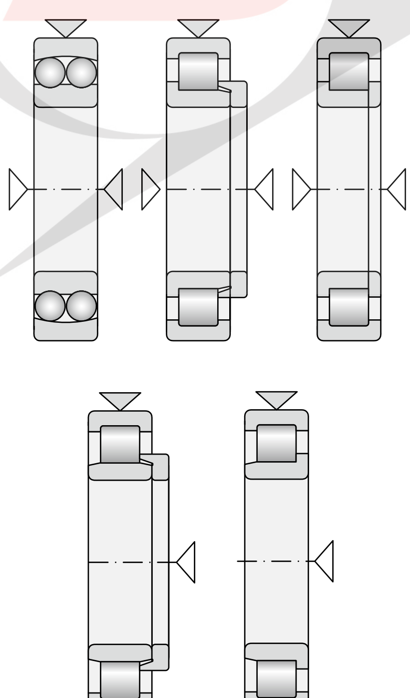

Fig. 4.3b

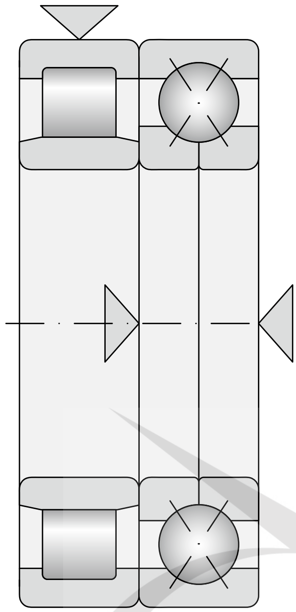

Fig. 4.4

In addition to thrust bearings, ball bearings or four--point ball bearings can be used for capturing axial forces (fig. 4.4)

4.1.4 Torque load

If the load application point lies outside of the bea-ring axis, then an overturning torque is created. The use of a radial double row bearing or a double row angular contact ball bearing usually suffices for its transfer. The use of a par of single row angular con-tact ball bearings or tapered roller bearings installed back-to-back in pairs (into an "O"), however, are preferred (fig. 4.5).

4.2 Available space

In certain circumstances, it presents as a limiting condition for the bearing design. In small-diameter housing, the single row ball bearing is most often

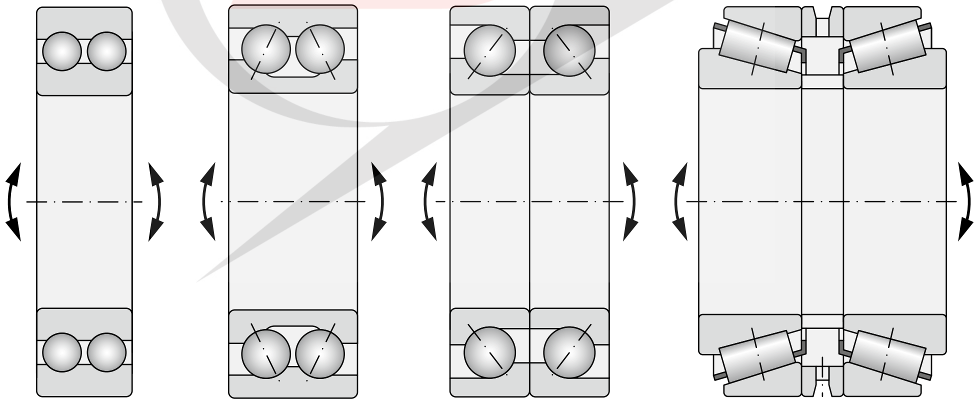

Fig. 4.5

applied (fig. 4.6). Cylindrical roller, spherical roller, and taper roller bearings may optionally be used for large diameter shafts (fig. 4.7). Various types of bearings also allow for a variety of types with various bearing section strengths. Where there is limited space in the radial or axial direction, bearings with a suitable cross--section are selected (fig. 4.8).

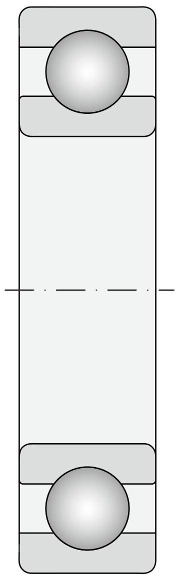

Fig. 4.6

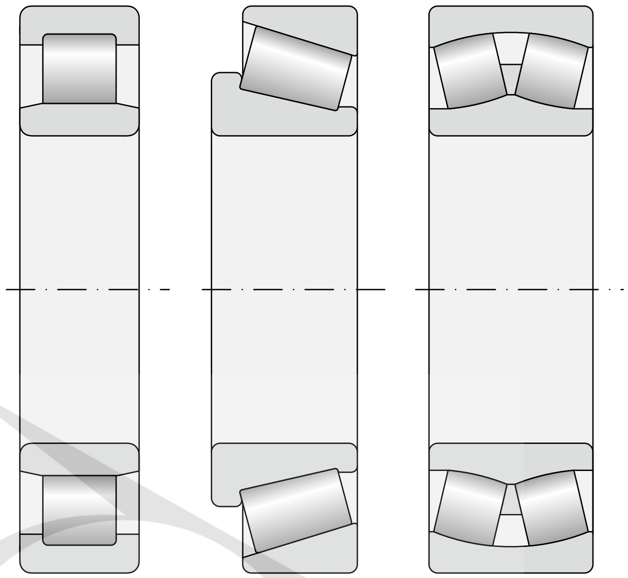

Fig. 4.7

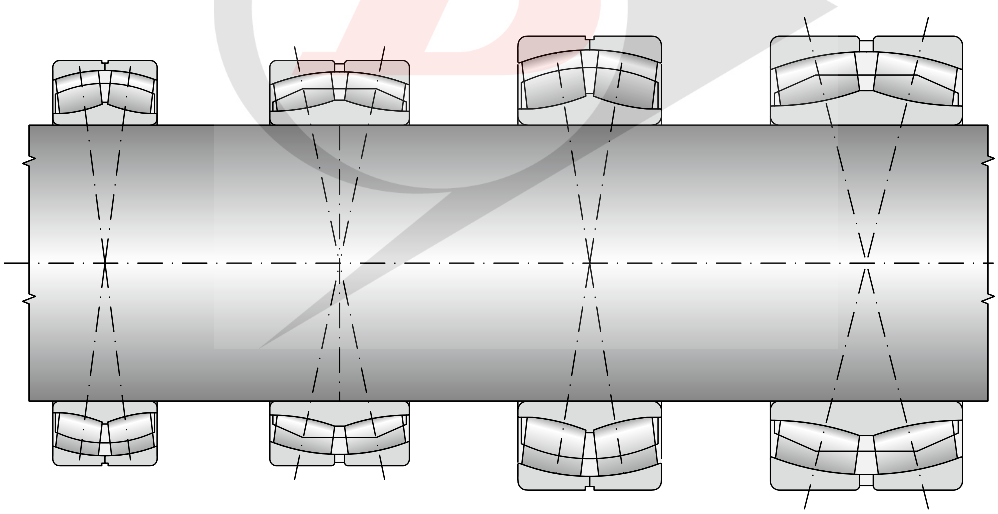

Fig. 4.8

4.3 Revolutions

Low-friction bearings should be used in housing subjected to high revolutions. Among such bearings are single-row ball bearings for purely radial loads. Angular-contact ball bearings in combined loads equally generate little heat. Both types of bearings are thus the most suitable for high revolution applications. Single row cylindrical roller bearings are additionally suitable for high revolutions.

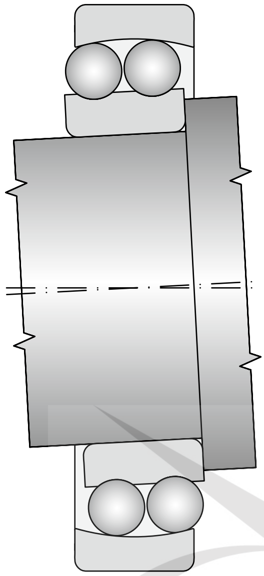

Fig. 4.9a

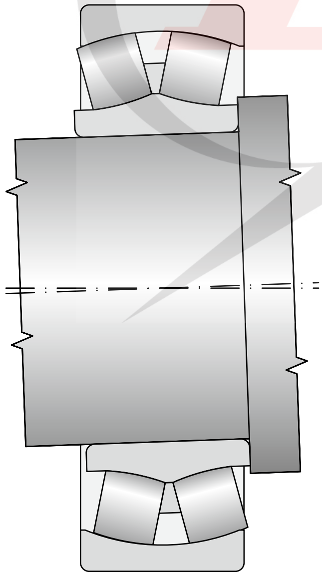

Fig. 4.9b

From a design aspect, the rpms in thrust bearings are always lower than those of radial bearings.

4.4 Precision of operation

Bearings with normal diameter precision and opera-tion (precision class P0) are sufficient for the most housing. In more demanding housing, e.g. for fitting machine tool spindles, bearings with higher precision must be used. Such bearings are designated by pre-cision classes P6, P6E, P6X, P5, P5A, P4, P4A, P2, SP, UP. In the text, which is located at the beginning of individual tables, you are provided with more detailed information about precision classes, in which individual types are produced.

4.5 Alignment

With regard to manufacturing inaccuracies and spin-dle deflections, mutual inclinations of bearing rings occur in the housing. This phenomenon should be expected and it is necessary to select bearings that compensate for the misalignment and installation inaccuracy. Self-aligning ball bearings (fig. 4.9a), double row spherical roller bearings (fig. 4.9b), and thrust spherical roller bearings (fig. 4.9c), are such types. The angle of inclination of such bearings depends on the type, size, and load. High rigidity bearings, such as cylindrical roller bearing or ball bearings, can compensate for small misalignments, assuming that they are unburdened.

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4.6 Sliding axial movement

A fixed axial and free axial bearing is general used for supporting shafts, while the fixed axial bearing provides shaft guidance in both directions and the free axial bearing compensation for the axial change in length and thermal expansion. If axial displacement of thermally expanding components is prevented, then uncontrolled axial overloading of firmly fixed bearings may result.

Bearings that can carry combined loads are most suitable for capturing axial forces. Bearings that are best able to afford axial movement are NU and N cylindrical roller bearings (fig. 4.10). If ball or cylin-drical roller bearings are used as free bearings, then one of the bearing rings (usually the outer) must be attached freely (fig. 4.11).

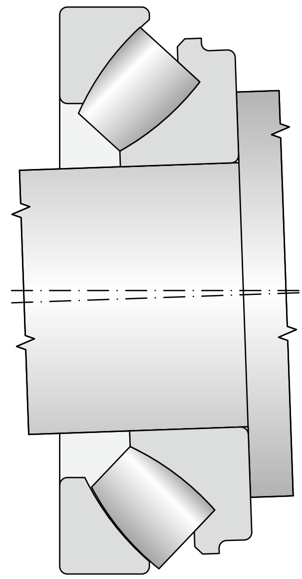

Fig. 4.9c

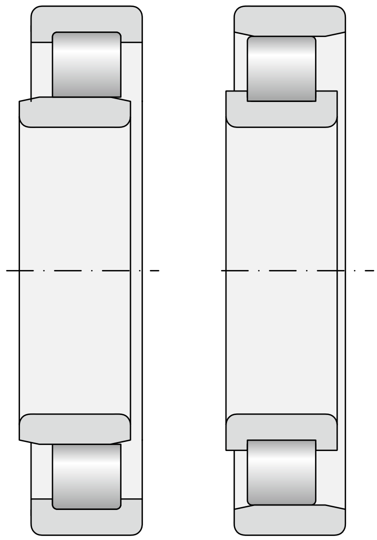

Fig. 4.10

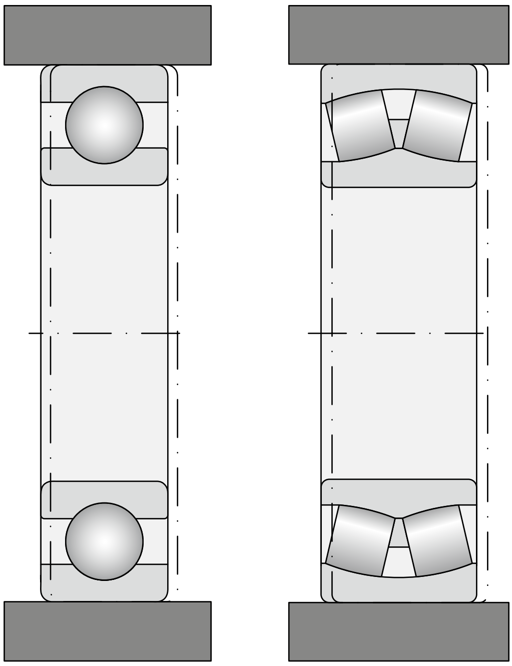

Fig. 4.11

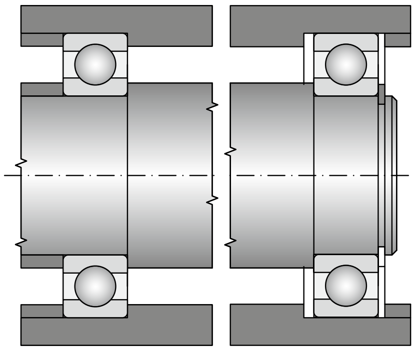

Fig. 4.12a

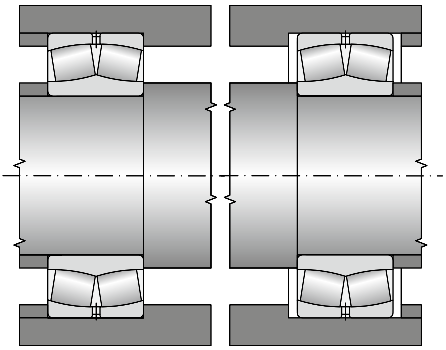

Fig. 4.12b

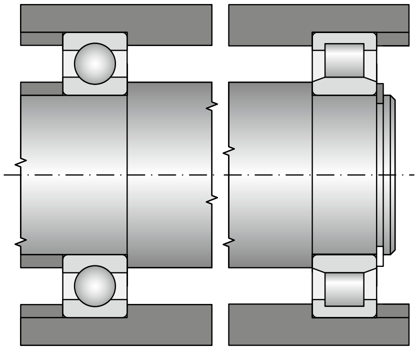

Fig. 4.12c

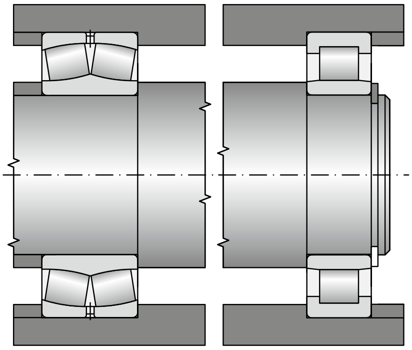

Fig. 4.12d

Examples of axially guided and free axial bearing supports are illustrated in figures 4.12a to 4.12

a) Axially guided ball bearing, free axial ball bearing
b) Axially guided spherical-roller bearing, free axial cylindrical roller bearing
c) Axially guided ball bearing, free axial NU cylindrical roller bearing
d) Axially guided spherical-roller bearing, free axial NU roller-contact bearing
e) Axially guided double-row angular-contact ball bearing, axially free NU cylindrical roller bearing
f) Axially guided four-point contact ball bearing and an NU cylindrical roller bearing, free axial NU roller-contact bearing
g) Axially guided double-row tapered-roller bearing, free axial NU cylindrical roller bearing
h) Axially guided NUP cylindrical roller bearing, free axial NU cylindrical roller bearing

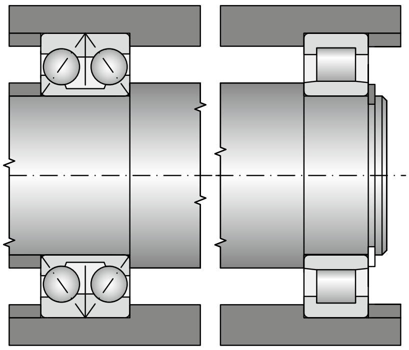

Fig. 4.12e

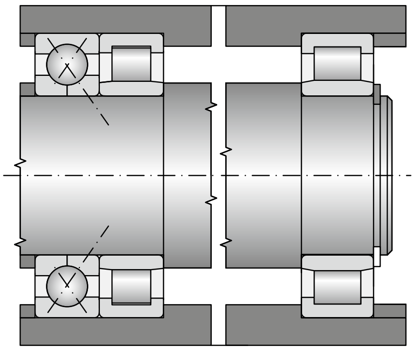

Fig. 4.12f

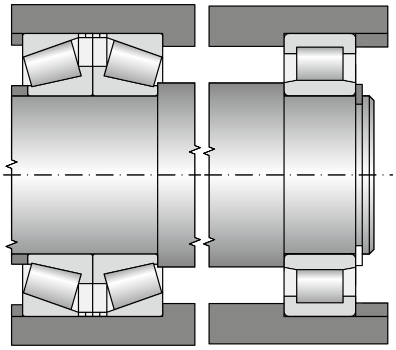

Fig. 4.12g

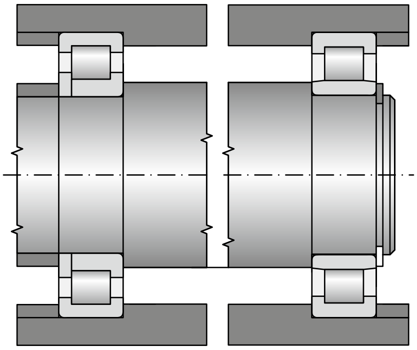

Fig. 4.12h

4.7 Support rigidity

The support rigidity expresses the force required to achieve a defined deflection when using a flexible support. High rigidity is demanded, for example when supporting the main spindle in machine tools and pinion gear sets.

The rigidity of line-contact bearings such as, e.g. cylindrical roller bearing and tapered roller bearings is higher than in ball bearings due to the contact ratios between the rolling elements and raceways.

The bearings are pre-stressed to increase their rigidity.

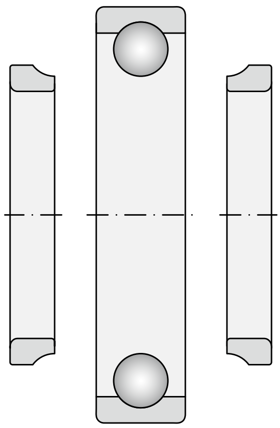

Fig. 4.13a

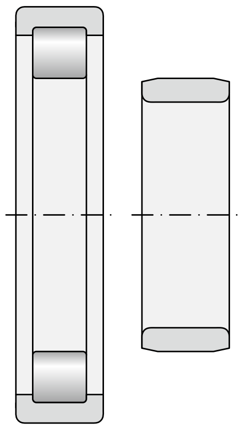

Fig. 4.13b

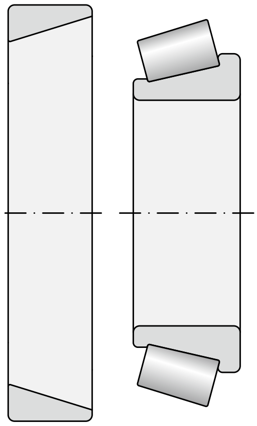

Fig. 4.13c

4.8 Installation options

4.8.1 Bearings with a cylindrical bore

These bearings are more easily installed and removed, if they can be taken apart. This particularly applies for bearings within a fixed housing. Separable bearings are also suitable for use where frequent installation and removal are required. A ring with roller elements may be installed separately, irrespective of the second ring (fig. 4.13a – 4.13c).

four-point contact ball bearing (obr.
4.13a) NU cylindrical roller bearing (fig.
4.13b) tapered-roller bearing (fig. 4.13c)

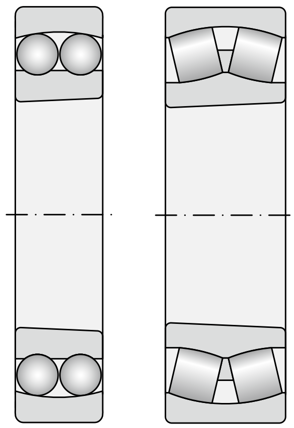

Fig. 4.14

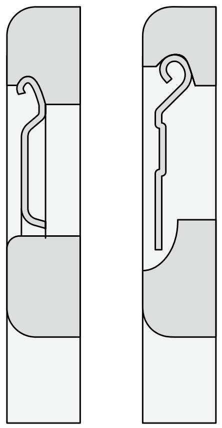

Fig. 4.15

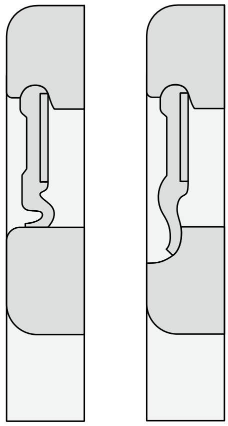

Fig. 4.16

4.8.2 Bearings with a tapered bore

Bearings with a tapered bore (fig. 4.14) are installed on a conical or cylindrical shaft using a adapter sleeve or withdrawal sleeve. The radial clearance of bearings can be set during installation. Installation and removal of bearings is relatively simple.

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5. DETERMINING BEARING SIZE

5.1 General information

A properly installed and lubricated roller-contact bearing will operate under normal conditions, i.e. absent extreme speeds and temperatures, until it fails due to fatigue of materials at acting surfaces. Repeated stress on the contact surfaces between roller-contact surfaces and rings will manifest after a certain period depending on the magnitude of load as a stress fracture. This will expand until a part of the bearing ring material or roller element material breaks off (pitting) and causes failure. Many bearings are also discarded for other reasons than material fatigue, but these failures can be avoided if the bearing is treated properly, if it is properly installed, lubricated, and overloading is avoided.

When a certain number of identical bearings are tested for fatigue under specified operating conditions (load and rpm), there is a large variance of durability between individual bearings. In a group of 30 or more bearings, the ratio between the shortest and longest durability can be 20-fold or more. A durability variance curve can be drawn for each tested group of bearings that illustrates the relationship between the durability and the number of bearings, which were discarded.

The required bearing size is determined on the basis of externally acting forces and based on the dura-bility and reliability demands of the seated bearing. The size, direction, purpose, and nature of the bearing load as well as the revolution operating speed are determinant when selecting the bearing type and size. Meanwhile, other special or important conditions of each individual case must be considered, e.g. operating temperature, spatial allowances, ease of installation, lubrication requirements, packing, etc., which can affect the selection of the most suitable bearing. Various types of bearings may, in many cases, be suitable for the given specific conditions.

In terms of the action of external forces and the function of the bearing in the respective node or unit, we distinguish two types of roller bearing loads in bearing technology:

- If the bearing rings turn in relation to one another and the bearing is exposed, under such conditions, to external forces (which applies for the majority of bearing applications), we refer to this as a dynamic bearing load,
- If the bearing rings do not turn in relation to one another or turn very slowly, the bearing transmits oscillating motion, or external forces act for shorter period than the time of one bearing revolution, we refer to this as a static bearing load.

The durability limited by failure of a particular bearing component (bearing rings, roller elements, cage, lubricant and seal) is, in the first case, decisive for calculating bearing safety. In the second case, permanent deformities of functional surfaces at contact points between rolling elements and orbits is decisive.

5.2 Roller bearing reliability

The reliability of a group of apparently identical roller bearings, operating under identical conditions, is the percentage of the group, expected to achieve or exceed the specified durability.

The reliability of an individual roller bearing is the probability that the bearing will achieve or exceed the specified durability.

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The equation for calculating durability includes the effect of stress induced by external loads, lubrication, and surface kinematics at the site of rolling contact. Including the impact of the comprehensive system of stress on bearing durability makes it possible to better anticipate the actual manner, in which a bearing behaves within a specific housing. International standards, such as e.g. ISO 281, are based on the theory of material fatigue at the site of rolling contact. One must keep in mind that a complete bearing can be consi-dered as a system, the individual components of which (bearing rings, rolling elements, cage, lubricant and seal) have the same effect on durability and, in certain cases, are even a decisive factor in determining the bearing durability during operation. The optimal operating durability is theoretically achieved when all of the components achieve the same durability. In other words, the calculated durability corresponds to the actual operating durability if the operating durability of related components is at least as long as the calculated bearing durability. Related components in such case are the cage, seal and lubricant. The most important factor in practise is metal fatigue.

5.3 Dynamic Load Capacity

Dynamic load capacity is, according to ISO 281:1990, a constant invariable load that a bearing can theoretically carry at a basic durability of one million revolutions.

The dynamic load capacity C_r for radial bearings relates to constant, invariable, entirely radial loads. For thrust bearings, the dynamic load capacity C_a relates to the invariable, purely axial load acting in the bearing's axis.

The dynamic load capacity C_r and C_a, whose magnitude depends on the bearing dimensions, the number of rolling elements, the bearing material and design, is provided in the table for each bearing. The dynamic load capacity values were determined in accordance with ISO standard 280. These values are verified on testing equipment and confirmed in operating results.

The numeric values specified in this catalogue apply for chrome steel bearings, heat treated to a minimal hardness of 58 HRC and normal operating conditions. NEW FORCE bearings display, among others, improved material properties and advanced manufacturing processes. To determine the dynamic load capacity in these bearings, thus requires the use of correction factors according to ISO 281. More information about these bearings is available in separate chapter 7.7.

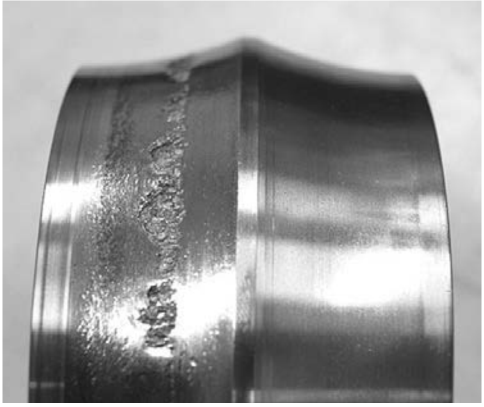

Fig. 5.1 Photo-illustration of fatigue damage on the raceway

Fig. 5.2 Photo-illustration of fatigue damage on the raceway

5.4 Durability

It is the number of revolutions that a bearing lasts, before fatigue of one of its components occurs, which manifests as flaking of material. It is expressed either as the total number of revolutions or operating hours, or in vehicles, by the distance travelled (number of driven km).

The material is primarily responsible for significant variance in durability in a broader range of identical bearings tested under the same conditions. No material or bearing steel is entirely homogenous and contains certain weak points. If a weak point is located on the orbit, where large load (stress) is generated, then the durability of the bearing will be small. The durability is higher where the load is decreased. Poor material has a large amount of weak points and, in all likeliness, some of them lie in areas of greatest load. The variance of durability will thus be less in poor material and larger in first-class material.

Variance of durability is also affected by manufacturing tolerances of individual components. The toleran-ces of roller diameters and radiuses of raceways significantly affect loads on roller surfaces. For manufactu-ring reasons, the radial clearance in a bearing varies within a specific tolerance, and as such, it also affects the distribution of pressure on individual roller elements. The distribution of forces within the bearing in the same manner cause expansion and decrease the orbit diameter due to the prescribed placement of rings on the shaft and within the housing.

Adherence to the prescribed material composition, its purity, and heat treatment is also an important indi-cator of bearing quality. Large variances in durability of large quantities of identical roller bearings, testing under identical conditions is but a natural consequence of the specified individual influences. Current research shows that even the quality of lubrication, its purity, and quantity may significantly impact bearing durability. Lubrication is taken into account in the modified durability calculation, see further.

The results of performed durability tests and practical operating experiences indicate that identical bea-rings, operating under identical conditions, do not achieve the same durability. The term "durability" must thus be correctly defined.

5.5 Basic durability equation

The basic durability of a bearing is mathematically defined by the durability equation, which applies for all types of bearings.

$$L_{10} = \left[\frac{C}{P}\right]^p \quad \text{or} \quad \frac{C}{P} = (L_{10})^{1/p}$$

L_{10} . . . . . .basic durability [10^6 rev]

C . . . . . . .dynamic load capacity [kN]
(the C_r and C_a values are specified in the product section of the catalogue)

P . . . . . . .bearing equivalent dynamic load capacity [kN]
(the equations for calculating P_r and P_a are provided in the chapter Equivalent Dynamic Load Capacity and for each structural group of bearings)

p . . . . . . .ball bearing exponent $p = 3$

. . . . .For cylindrical roller, needle roller, spherical-roller, and tapered-roller bearings $p = \frac{10}{3}$

The basic durability of a bearing is thus understood to mean the durability that 90 % of bearings achieve or exceed from a set of identical bearings, working under the same operating conditions. All standard durability calculations are performed for this reliability level. Mean durability L_s is the durability that 50 % of bearing from the same set achieve; it is about 5 times higher than the basic durability. In contrast, the durability achieved by 99 % of bearings is about one fifth when compared with the basic durability. The impact of the degree of reliability on the durability calculation is specified in chapter 5.6.

Table 5.1 lists the relationship of durability L_{10} in millions of revolutions and the corresponding C/P ratio. If the revolution speed is unchanged, then the durability can be calculated using the modified equation, which expresses the basic durability in terms of operating hours:

$$L_{10h} = \left[\frac{C}{P}\right]^p \cdot \left[\frac{10^6}{60n}\right]$$

L_{10h} . . . . . basic durability [h]

n . . . . revolution speed $[min^{-1}]$

The relationship of the C/P ratio on basic durability L_{10h} and on the revolution speed for ball bearings is specified in table 5.2 and in table 5.3 for cylindrical roller, needle, spherical-roller, and tapered-roller bearings.

In road and rail vehicle axle supports, we can express the basic durability using the modified relationship in terms of kilometres driven.

$$L_{10km} = \left[\frac{C}{P}\right]^p \cdot \frac{\pi \cdot D}{1000}$$

L_{10km}. . . . . basic durability $[10^6\,km]$

D . . . . wheel diameter [m]

5.5.1 Standard values of basic durability

In cases, when the required durability for the given housing is not provided in advance, we can appropriately use the values provided in tables 5.4 and 5.5.

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Table 5.1

C/P ratio depending on durability L_{10h}							
Ball bearings				Cylindrical roller, needle-roller, spherical-roller, and tapered-roller bearings			
L_{10} Durability	C/P	L_{10} Durability	C/P	L_{10} Durability	C/P	L_{10} Durability	C/P
× 10^6 rev		× 10^6 rev		× 10^6 rev		× 10^6 rev	
0,5	0,79	600	8,43	0,5	0,81	600	6,81
0,75	0,91	650	8,66	0,75	0,92	650	6,98
1	1,00	700	8,88	1	1,00	700	7,14
1,5	1,14	750	9,09	1,5	1,13	750	7,29
2	1,26	800	9,28	2	1,24	800	7,43
3	1,44	850	9,47	3	1,39	850	7,56
4	1,59	900	9,65	4	1,52	900	7,70
5	1,71	950	9,83	5	1,62	950	7,82
6	1,82	1 000	10,00	6	1,71	1 000	7,94
8	2,00	1 100	10,30	8	1,87	1 100	8,17
10	2,15	1 200	10,60	10	2,00	1 200	8,39
12	2,29	1 300	10,90	12	2,11	1 300	8,59
14	2,41	1 400	11,20	14	2,21	1 400	8,79
16	2,52	1 500	11,40	16	2,30	1 500	8,97
18	2,62	1 600	11,70	18	2,38	1 600	9,15
20	2,71	1 700	11,90	20	2,46	1 700	9,31
25	2,92	1 800	12,20	25	2,63	1 800	9,48
30	3,11	1 900	12,40	30	2,77	1 900	9,63
35	3,27	2 000	12,60	35	2,91	2 000	9,78
40	3,42	2 200	13,00	40	3,02	2 200	10,10
45	3,56	2 400	13,40	45	3,13	2 400	10,30
50	3,68	2 600	13,80	50	3,23	2 600	10,60
60	3,91	2 800	14,10	60	3,42	2 800	10,80
70	4,12	3 000	14,40	70	3,58	3 000	11,00
80	4,31	3 500	15,20	80	3,72	3 500	11,50
90	4,48	4 000	15,90	90	3,86	4 000	12,00
100	4,64	4 500	16,50	100	3,98	4 500	12,50
120	4,93	5 000	17,10	120	4,20	5 000	12,90
140	5,19	5 500	17,70	140	4,40	5 500	13,20
160	5,43	6 000	18,20	160	4,58	6 000	13,60
180	5,65	7 000	19,10	180	4,75	7 000	14,20
200	5,85	8 000	20,00	200	4,90	8 000	14,80
250	6,30	9 000	20,80	250	5,24	9 000	15,40
300	6,69	10 000	21,50	300	5,54	10 000	15,80
350	7,05	12 500	23,20	350	5,80	12 500	16,90
400	7,37	15 000	24,70	400	6,03	15 000	17,90
450	7,66	17 500	26,00	450	6,25	17 500	18,70
500	7,94	20 000	27,10	500	6,45	20 000	19,50
550	8,19	25 000	29,20	550	6,64	25 000	20,90

Table 5.2

C/P ratio dependent on L_{10h} durability and rotation speed n for ball bearings													
L_{10h} Durability	**Rotation speed n [min^{-1}]**												
Hod	10	16	25	40	63	100	125	160	200	250	320	400	500
100	-	-	-	-	-	-	-	-	1,06	1,15	1,24	1,34	1,45
500	-	-	-	1,06	1,24	1,45	1,56	1,68	1,82	1,96	2,12	2,29	2,47
1 000	-	-	1,15	1,34	1,56	1,82	1,96	2,12	2,29	2,47	2,67	2,88	3,11
1 250	-	1,06	1,24	1,45	1,68	1,96	2,12	2,29	2,47	2,67	2,88	3,11	3,36
1 600	-	1,15	1,34	1,56	1,82	2,12	2,29	2,47	2,67	2,88	3,11	3,36	3,63
2 000	1,06	1,24	1,45	1,68	1,96	2,29	2,47	2,67	2,88	3,11	3,36	3,63	3,91
2 500	1,15	1,34	1,56	1,82	2,12	2,47	2,67	2,88	3,11	3,36	3,63	3,91	4,23
3 200	1,24	1,45	1,68	1,96	2,29	2,67	2,88	3,11	3,36	3,63	3,91	4,23	4,56
4 000	1,34	1,56	1,82	2,12	2,47	2,88	3,11	3,36	3,63	3,91	4,23	4,56	4,93
5 000	1,45	1,68	1,96	2,29	2,67	3,11	3,36	3,63	3,91	4,23	4,56	4,93	5,32
6 300	1,56	1,82	2,12	2,47	2,88	3,36	3,63	3,91	4,23	4,56	4,93	5,32	5,75
8 000	1,68	1,96	2,29	2,67	3,11	3,63	3,91	4,23	4,56	4,93	5,32	5,75	6,20
10 000	1,82	2,12	2,47	2,88	3,36	3,91	4,23	4,56	4,93	5,32	5,75	6,20	6,70
12 500	1,96	2,29	2,67	3,11	3,36	4,23	4,56	4,93	5,32	5,75	6,20	6,70	7,23
16 000	2,12	2,47	2,88	3,36	3,91	4,56	4,93	5,23	5,75	6,20	6,70	7,23	7,81
20 000	2,29	2,67	3,11	3,63	4,23	4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43
25 000	2,47	2,88	3,36	3,91	4,56	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11
32 000	2,67	3,11	3,63	4,23	4,93	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83
40 000	2,88	3,36	3,91	4,56	5,32	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60
50 000	3,11	3,63	4,23	4,93	5,75	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50
63 000	3,36	3,91	4,56	5,32	6,20	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40
80 000	3,36	4,23	4,93	5,75	6,70	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40
100 000	3,91	4,56	5,32	6,20	7,23	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50
200 000	4,93	5,75	6,70	7,81	9,11	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20

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C/P ratio dependent on L_{10h} durability and rotation speed n for ball bearings

Rotation speed n [min^{-1}]

630	800	1 000	1 250	1 600	2 000	2 500	3 200	4 000	5 000	6 300	8 000	10 000	12 500	16 000
1,56	1,68	1,82	1,96	2,12	2,29	2,47	2,67	2,88	3,11	3,36	3,63	3,91	4,23	4,56
2,67	2,88	3,11	3,36	3,63	3,91	4,23	4,56	4,93	5,32	5,75	6,20	6,70	7,23	7,81
3,36	3,63	3,91	4,23	4,56	4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83
3,63	3,91	4,23	4,56	4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60
3,91	4,23	4,56	4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50
4,23	4,56	4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40
4,56	4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40
4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50
5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60
5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80
6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20
6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60
7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20
7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90
8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70
9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70
9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80
10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80	31,10
11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80	31,10	-
12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80	31,10	-	-
13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80	31,10	-	-	-
14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80	31,10	-	-	-	-
15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80	31,10	-	-	-	-	-
19,60	21,20	22,90	24,70	26,70	28,80	31,10	-	-	-	-	-	-	-	-

Table 5.3

C/P ratio dependent on L_{10h} durability and rotation speed n for cylindrical roller, spherical-roller, and tapered-roller bearings

L_{10h} Durability	Rotation speed n [min^{-1}]												
Hod	10	16	25	40	63	100	125	160	200	250	320	400	500
100	-	-	-	-	-	-	-	-	1,05	1,10	1,21	1,30	1,39
500	-	-	-	1,05	1,21	1,39	1,49	1,60	1,71	1,83	1,97	2,11	2,26
1 000	-	-	1,13	1,30	1,49	1,71	1,83	1,97	2,11	2,26	2,42	2,59	2,78
1 250	-	1,05	1,21	1,39	1,60	1,83	1,97	2,11	2,26	2,42	2,59	52,78	2,97
1 600	-	1,13	1,30	1,49	1,71	1,97	2,11	2,26	2,42	2,59	2,78	2,97	3,19
2 000	1,05	1,21	1,39	1,60	1,83	2,11	2,26	2,42	2,59	2,78	2,97	3,19	3,42
2 500	1,13	1,30	1,49	1,71	1,97	2,26	2,42	2,59	2,78	2,97	3,19	3,42	3,66
3 200	1,21	1,39	1,60	1,83	2,11	2,42	2,59	2,78	2,97	3,19	3,42	3,66	3,92
4 000	1,30	1,49	1,71	1,97	2,26	2,59	2,78	2,97	3,19	3,42	3,66	3,92	4,20
5 000	1,39	1,60	1,83	2,11	2,42	2,78	2,97	3,19	3,42	3,66	3,92	4,20	4,50
6 300	1,49	1,71	1,97	2,26	2,59	2,97	3,19	3,42	3,66	3,92	4,20	4,50	4,82
8 000	1,60	1,83	2,11	2,42	2,78	3,19	3,42	3,66	3,92	4,20	4,50	4,82	5,17
10 000	1,71	1,97	2,26	2,59	2,97	3,42	3,66	3,92	4,20	4,50	4,82	5,17	5,54
12 500	1,83	2,11	2,42	2,78	3,19	3,66	3,92	4,20	4,50	4,82	5,17	5,54	5,94
16 000	1,97	2,26	2,59	2,97	3,42	3,92	4,20	4,50	4,82	5,17	5,54	5,94	6,36
20 000	2,11	2,42	2,78	3,19	3,66	4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81
25 000	2,26	2,59	2,97	3,42	3,92	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30
32 000	2,42	2,78	3,19	3,66	4,20	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82
40 000	2,59	2,97	3,42	3,92	4,50	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38
50 000	2,78	3,19	3,66	4,20	4,82	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98
63 000	2,97	3,42	3,92	4,50	5,17	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62
80 000	3,19	3,66	4,20	4,82	5,54	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30
100 000	3,42	3,92	4,50	5,17	5,94	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00
200 000	4,20	4,82	5,54	6,36	7,30	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60

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C/P ratio dependent on L_{10h} durability and rotation speed n for cylindrical roller, spherical-roller, and tapered-roller bearings

Rotation speed n [min^{-1}]

630	800	1 000	1 250	1 600	2 000	2 500	3 200	4 000	5 000	6 300	8 000	10 000	12 500	16 000
1,49	1,60	1,71	1,83	1,97	2,11	2,26	2,42	2,59	2,78	2,97	3,19	3,42	3,66	3,92
2,42	2,59	2,78	2,97	3,19	3,42	3,66	3,92	4,20	4,50	4,82	5,70	5,54	5,94	6,36
2,97	3,19	3,42	3,66	3,92	4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82
3,19	3,42	3,66	3,92	4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38
3,42	3,66	3,92	4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98
3,66	3,92	4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62
3,92	4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30
4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00
4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80
4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70
5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60
5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60
5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60
6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70
6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90
7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20
7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60
8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60	-
8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60	-	-
9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60	-	-	-
10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60	-	-	-	-
11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60	-	-	-	-	-
11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60	-	-	-	-	-	-
14,60	15,60	16,70	17,90	19,20	20,60	-	-	-	-	-	-	-	-	-

Table 5.4

Standard basic durability values in operating hours	
Type of machine	**Basic durability L_{10h}**
Seldom used machines and tools	1 000
Electrical household appliances, small fans	2 000 to 4 000
tools for intermittent use, hand tools, workshop cranes, agricultural machines	4 000 to 8 000
machines for intermittent use with high reliability demands, auxiliary machines for use in power plants, belt conveyors, transport trolleys, elevators	8 000 to 15 000
rolling mills	6 000 to 12 000
machines for 8-16 hour shifts, stationary motors, gears, spindles for textile machines, plastic processing machinery, printing machinery, cranes	15 000 to 30 000
machine tools, in general	20 000 to 30 000
machines for continuous operation: stationary electrical machines, transportation equipment, roller conveyors, pumps, centrifuges, blowers, compressors, hammer mills, shredders, briquetting presses, mine hoists, cable reels	40 000 to 60 000
machines for continuous operation with high operating safety requirements: power plant machinery, waterworks machines, paper mill machinery, ship machinery	100 000 to 200 000

Table 5.5

Standard values of basic durability in kilometres	
Type of vehicle	**Basic durability L_{10km}**
Road vehicle wheel bearings	
motorcycles	60 000
personal automobiles	150 000 to 250 000
lorries, buses	400 000 to 500 000
Axle bearings of rail vehicles	
freight cars (according to UIC) under constant maximal load per axle	800 000
trams	1 500 000
personal rail vehicles	3 000 000
motorized vehicles and motorized units	3 000 000 to 4 000 000
locomotives	3 000 000 to 5 000 000

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5.6 Modified durability equation

The operating durability, as previously described, depends on many factors. Research and operating results demonstrated that greater durability can be achieved through thorough lubrication, when the roller elements are entirely separated by a layer of grease. It was further demonstrated that greater resistance against stress damage of materials is provided using advanced manufacturing processes. This technical advance was incor-porated into standard ISO 281 as a modified durability calculation, which includes reliability a_1, material a_2, and operating condition a_3 factors. Additional test results concluded that the impact of materials on operating conditions, in particular, lubrication, are in close correlation. This led to the merger of both factors into one a_{23}.

The modified durability is thus the modified basic durability which, aside from taking into account load, also considers the impact of bearing material components, the physical and chemical properties of the lubricant, and the temperature regime of the bearing operating environment.

$$L_{na} = a_1 \cdot a_{23} \cdot L_{10}$$

L_{na} . . . . modified durability for reliability (100 - n) % and other than normal operating conditions [10^6 rev]

a_1. . . . . eliability coefficient for other than 90 % reliability, see table 5.6

a_{23} . . . . material, lubricant, manufacturing technology, and operating condition coefficient, see fig. 5.3

L_{10} . . . . basic durability [10^6 rev]

Table 5.6

Coefficient a_1 values		
Reliability (%)	**L_n**	**a_1**
90	L_{10}	1,000
95	L_5	0,640
96	L_4	0,550
97	L_3	0,470
98	L_2	0,370
99	L_1	0,250
99,2	$L_{0,8}$	0,220
99,4	$L_{0,6}$	0,190
99,6	$L_{0,4}$	0,160
99,8	$L_{0,2}$	0,120
99,9	$L_{0,1}$	0,093
99,92	$L_{0,08}$	0,087
99,94	$L_{0,06}$	0,080
99,95	$L_{0,05}$	0,077

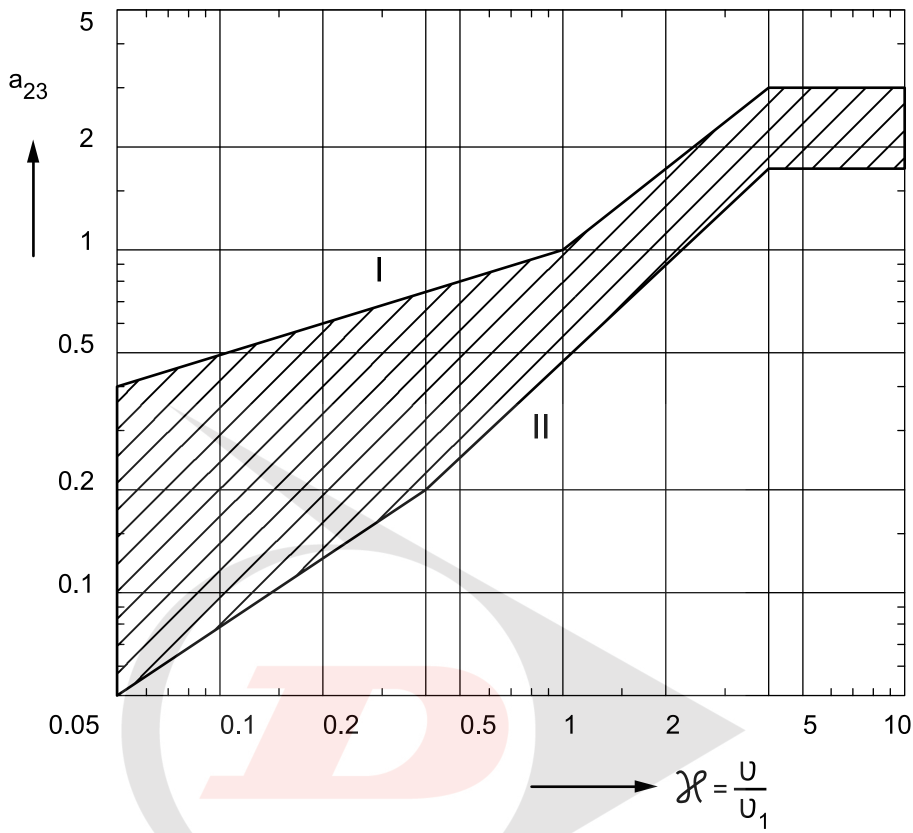

Fig. 5.3

The diagram in fig. 5.4 is used to determine the basic values of coefficient a_{23}.

The quality of the lubrication process is given by the extent of separation of the roller surfaces. Viscosity is a decisive factor for the formation of lubricant film, which is strongly related to temperature. The viscosity ratio, as follows, decides on the use of lubricant:

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

ν . . . . . .lubricant kinematic viscosity at bearing operating temperature $[mm^2 \cdot s^{-1}]$

ν_1 . . . . .kinematic viscosity for the defined revolution speed and the given dimension of the bearing $[mm^2 \cdot s^{-1}]$

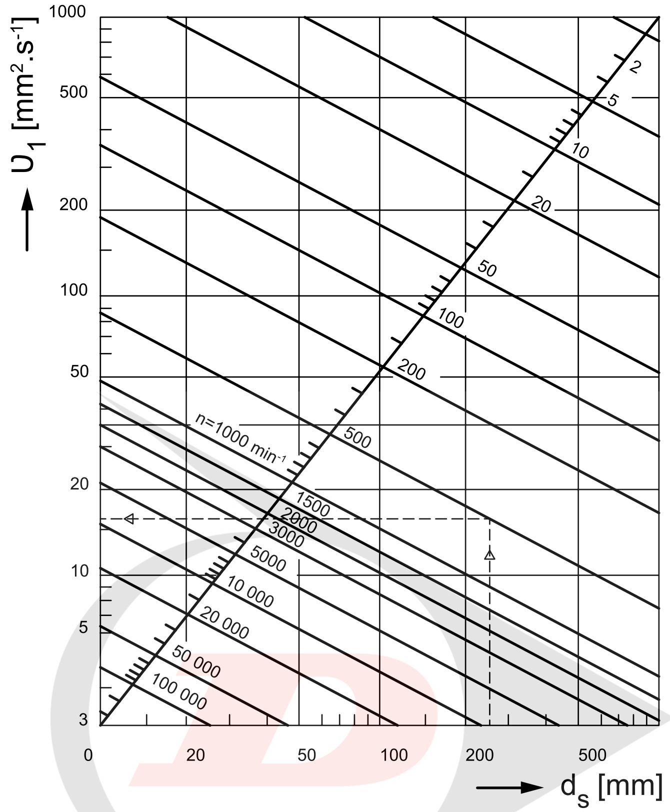

Fig. 5.4

We determine the ν and ν_1 values based on the diagram found in fig. 5.4 and 5.5. In the diagram on fig. 5.3, line I applies for radial ball bearings that operate in a very clean environment. In all other cases, we select a lower a_{23} coefficient, proportional to the cleanliness of the environment, while a decreasing tendency is dependent on the structural group of the bearing in the following order:

- Angular-contact ball bearings
- Tapered-roller bearings
- Cylindrical roller bearings
- Double-row self-aligning bearings
- Spherical-roller bearings

Line II can be used to determine coefficient a_{23} for spherical-roller bearings that operate in a dusty environ-ment.

We recommend that these issues be resolved in consultation with the Dunlop BTL technical and consultation services department.

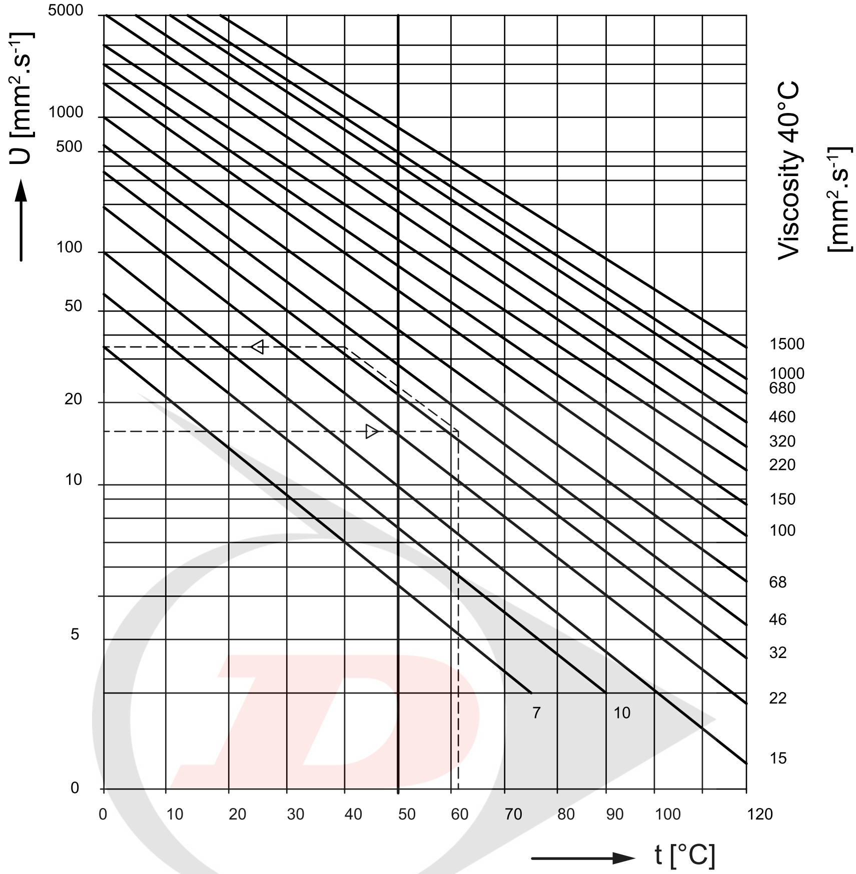
U [mm²·s⁻¹]
5000
1000
500
100
50
20
10
5
0
Viscosity 40°C [mm²·s⁻¹]
1500
1000
680
460
320
220
150
100
68
46
32
22
15
7
10
0
10
20
30
40
50
60
70
80
90
100
120
t [°C]

Fig. 5.5

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5.7 Durability according Dunlop BTL

The use of the L_{10} calculation of basic durability as bearing performance parameter criteria has demonstrated, over many years, to be satisfactory. This calculation is associated with 90 % reliability in conjunction with the use of superior materials, a superior technological design, and under normal operating conditions.

Notwithstanding, many applications require that the calculation be performed for a different reliability level or for more precise lubrication and contamination conditions. It was determined, with the use of advanced high quality bearing steel, that under favourable operating conditions and when contact stresses fall below the limit values and provided that the bearing steel fatigue stress limit is not exceeded, a higher durability than L_{10} can be achieved. Under unfavourable operating conditions, on the other hand, the bearing durability can in fact be shorter than L_{10}.

A system approach of fatigue-related durability was applied when creating the method of calculating Dunlop BTL modified durability. The impact on the durability of the system (bearing) is described in the following text and considers the influence of variance and the interaction of mutually related factors on the overall life. These factors are demonstrated through increased contact stress in the contact area, which leads to decreased service life.

These factors are used in the modified durability equation.

$$L_m = a_1 \cdot a_{\text{Dunlop BTL}} \cdot L_{10}$$

a_1 . . . . reliability coefficient for other than 90 % reliability, see table 5.6

$a_{\text{Dunlop BTL}}$ . . . . . modified life coefficient

L_{10} . . . . . . basic durability [10^6 rev]

Provided that the lubrication conditions, cleanliness of the environment, and other operation conditions are favourable, an advanced, high-quality bearing can, under a certain load, achieve infinite service life. The fatigue load limit for bearings manufactured from generally high-quality bearing material and workmanship is such a load, that the contact pressure exerted on roller elements in the bearing is approximately 1500 MPa. This stress value takes into account the additional stresses caused by manufacturing tolerances and operating conditions. Decreased product precision and quality of materials leads to a lower fatigue load limit.

The contact stress in many applications is greater than 1500 MPa. Such operating conditions lead to reduced bearing life.

The operating influences can be related to the applied stress and rigidity of the material.

- Notches lead to the formation of edge stresses.
- A thin film of oil increases the stress at the contact area between the raceway and the roller element.
- Increased temperature decreases the fatigue load limit (its strength) of the material.
- A static inner ring (increased overlap) leads to increased orbital stress

Various influences on bearing durability are mutually dependant. Consequently, a systemic approach to calculating fatigue durability is entirely appropriate.

A theoretical explanation of how to incorporate additional influences, such as the radial clearance during operation and the variable stress on raceways from tilting, is explained in ISO/TS 162 81.

5.7.1 Fatigue load limit

The modified durability coefficient $a_{\text{Dunlop BTL}}$ can be expressed as function

$$\frac{\sigma_u}{\sigma}$$

(fatigue load limit divided by the real stress σ, while considering all potential influencing factors).

If the actual stress decreases to fatigue stress limit, then $a_{\text{Dunlop BTL}}$ asymptotically approaches infinity. Generally, the orthogonal shear stress is used as a fatigue criterion. The diagram on fig. 5.6 is also based on the shear fatigue limit.

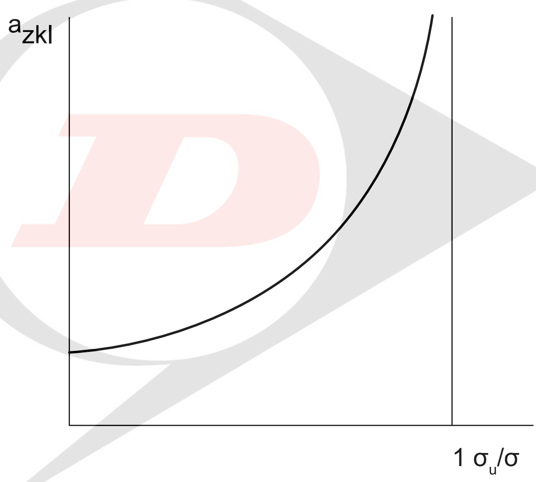

Fig. 5.6

Analogous to the C_{or} static load rating, defined in ISO 76, the fatigue load limit is defined as the load, during which the stress fatigue limit is reached at the most burdened point on the orbit.

The ratio $\dfrac{\sigma_u}{\sigma}$ can then be estimated according to the ratio $\dfrac{C_{or}}{P}$

and the modified life coefficient can be expressed as:

$$a_{\text{Dunlop BTL}}\left[\frac{C_{or}}{P}\right]$$

The following must be considered when calculating the C_{or} static load rating:

- The type, size, and internal geometry of the bearing
- The profile of rolling elements and the raceways
- The quality of technological processes
- The fatigue limit for the raceway materials

5.7.2 Determining the modified durability coefficient

The modified durability coefficient takes into consideration the following:

- The fatigue load and bearing load
- Lubrication (type of lubricant, viscosity, revolution speed, bearing size, additives)
- Environment (degree of contamination, packing)
- Contaminating particles (strength and size of particles in relation to bearing size, lubrication and filtration method)
- Installation (cleanliness during installation)

The effect of bearing clearance and the effect of tilt on bearing durability is described in ISO/TS 16281.

The $a_{\text{Dunlop BTL}}$ Fatigue life coefficient is derived from the following equation:

$$a_{\text{Dunlop BTL}} = f\left[\frac{e_c \cdot C_{or}}{P}, \kappa\right]$$

Factors e_c and κ adjust for contamination and lubrication conditions.

5.7.3 Contamination factor

If the grease is contaminated with sold particles, notches may form in the orbit due to rolling. Stress points (concentrations) form later on these notches, which results in decreased bearing life. The given decrease in life caused by the contamination of lubricant is adjusted for in the e_c contamination factor.

Decreased bearing life caused by the effect of sold particles in the lubricant film depends on:

- The type, size, strength, and amount of particles
- The lubricating film thickness (relative viscosity)
- Bearing size

Approximate contamination factor values can be taken from table 5.7.

Table 5.7

Contamination level	e_c	
	D_{pw} < 100 mm	D_{pw} ≥ 100 mm
Extremely clean Particle size in the order of lubricating film thickness, Laboratory conditions	1	1
Highly clean Oil filtered through a very fine filter, typical conditions for a bearing with plastic housing and lifetime lubricant filling	0,8 to 0,6	0,9 to 0,8
Normally clean Oil filtered through a fine filter, typical conditions for a bearing with metal-sheet housing and lifetime lubricant filling	0,6 to 0,5	0,8 to 0,6
Mild contamination Minor contamination in lubricant	0,5 to 0,3	0,6 to 0,4
Typical contamination Typical bearing conditions without integrated bearing glands, particles causing wear enter bearing from vicinity	0,3 to 0,1	0,4 to 0,2
Strong contamination The bearing environment is strongly contaminated, bearing housing with insufficient bearing glands	0,1 to 0	0,1 to 0
Very strong contamination	0	0

Detailed calculation of the contamination factor

Table 5.7 lists the approximate contamination factor values. If the situation requires the use of more detailed calculations, the more precise calculation, provided below, must be used.

A contamination factor may be established for the following types of lubricants:

- Circulating oil lubrication with on-line filtration
- Oil bath lubrication or circulating lubrication with off-line filtration
- Grease

Definition of the β_x filtration ratio:

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

β_x . . . . . . filtration ratio for particles of determined size x

n_1. . . . . . .number of particles per unit of volume (100 ml) larger than x, prior to passage through filter

n_2. . . . . . .number of particles per unit of volume (100 ml) larger than x, after passage through filter

The filter ratio determined the filter efficiency.

Circulating lubrication with on-line filtration

The β_x filter ratio with particles of size x in μm according to standard ISO 16889 is the most influential factor when choosing the corresponding diagram.

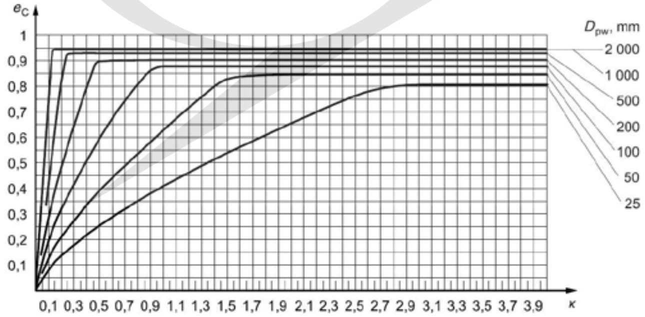

Fig. 5.7Fouling factor for a circulating oil lubrication system with on-line filtration $\beta_6 = 200$

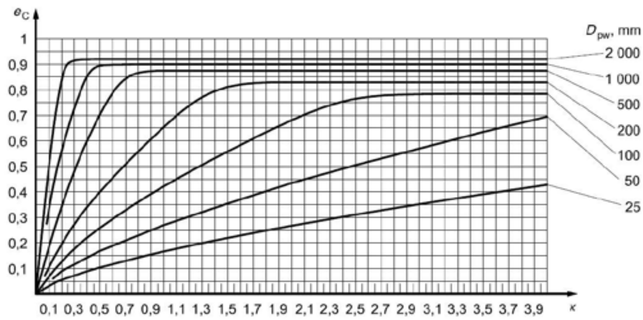

Fig. 5.8Fouling factor for a circulating oil lubrication system with on-line filtration $\beta_{12} = 200$

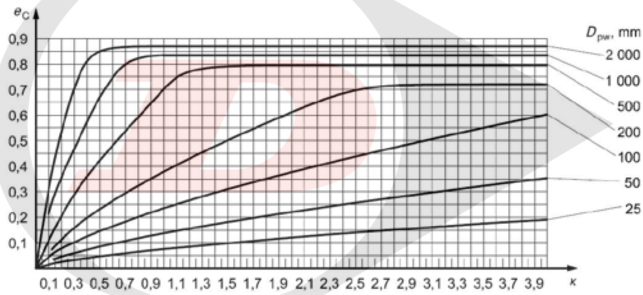

Fig. 5.9Fouling factor for a circulating oil lubrication system with on-line filtration $\beta_{25} = 75$

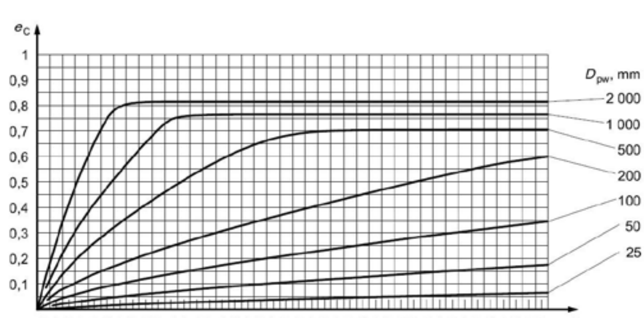

Fig. 5.10Fouling factor for a circulating oil lubrication system with on-line filtration $\beta_{40} = 75$

Oil bath lubrication or circulating lubrication with off-line filtration

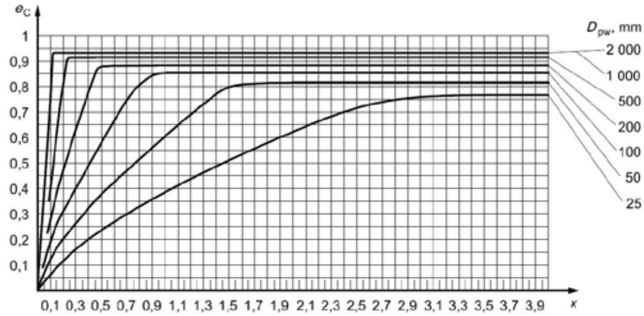

Fig. 5.11Fouling factor for oil bath lubrication or for oil lubrication with offline filtration
ISO 4406 – degree of contamination by solid particles -13/10

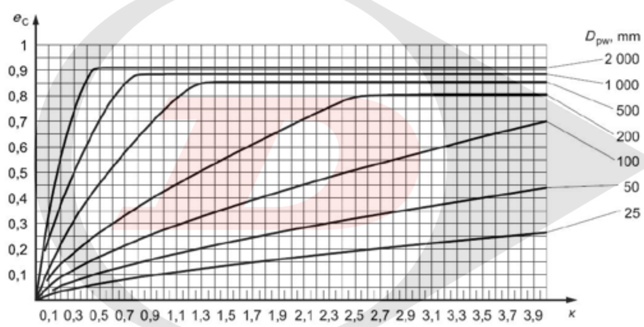

Fig. 5.12Fouling factor for oil bath lubrication or for oil lubrication with offline filtration
ISO 4406 – degree of contamination by solid particles -15/12

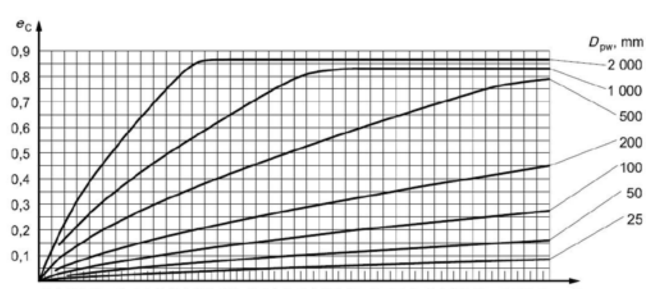

Fig. 5.13Fouling factor for oil bath lubrication or for oil lubrication with offline filtration
ISO 4406 – degree of contamination by solid particles -17/14

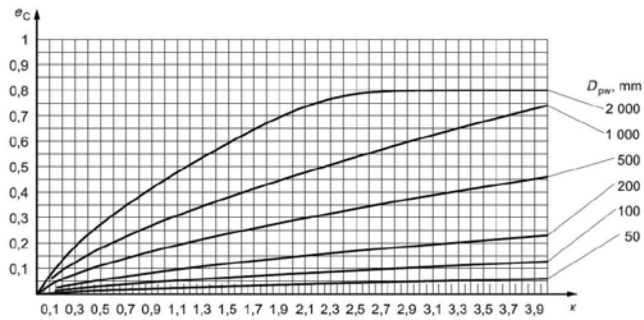

Fig. 5.14Fouling factor for oil bath lubrication or for oil lubrication with offline filtration
ISO 4406 – degree of contamination by solid particles -19/16

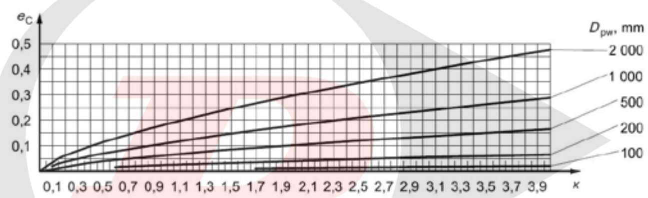

Fig. 5.15Fouling factor for oil bath lubrication or for oil lubrication with offline filtration
ISO 4406 – degree of contamination by solid particles -21/18

Grease

Table 5.8

Operating conditions	Contamination level
Very clean installation, very good packing relative to operating conditions, continuous lubrication or lubrication in short intervals (Bearings with integrated bearing glands)	Highly clean
Clean installation, good packing, additional lubrication per manufacturer specifications (Bearings with integrated bearing glands)	Normally clean
Clean installation, average sealing capacity relative to operating conditions	Mild contamination
On-site-installation, bearing and housing insufficiently washed following installation, poor sealing capacity relative to operating conditions, re-lubrication intervals longer than recommended	Strong contamination
Installation in a contaminated environment, insufficient gland packaging, long re-lubrication intervals	Very strong contamination

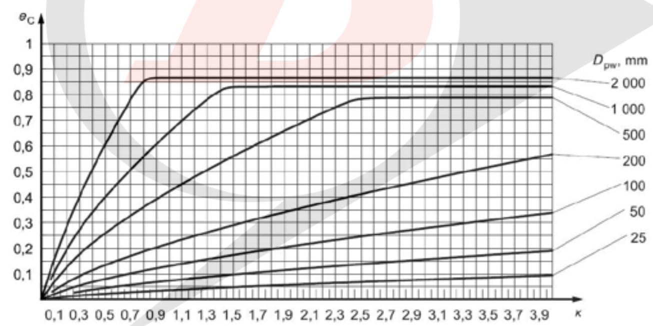

Fig. 5.16Fouling factor for grease lubrication – moderate pollution

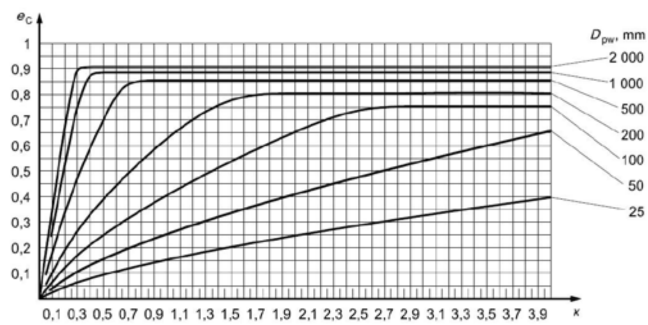

Fig. 5.17Fouling factor for grease lubrication– usual purity

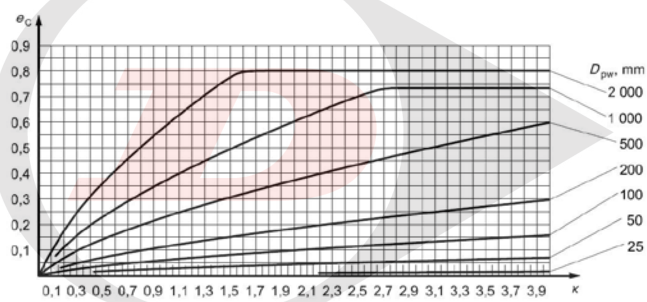

Fig. 5.18Fouling factor for grease lubrication - strong contamination

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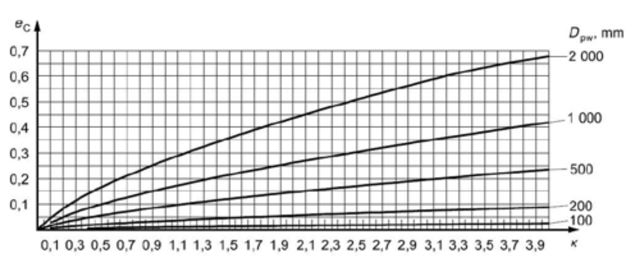

Fig. 5.19Fouling factor for grease lubrication – very strong contamination

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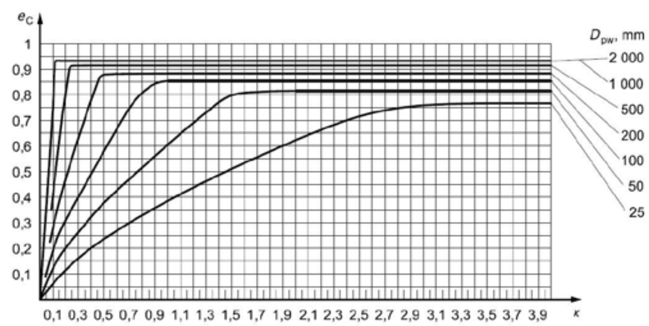

Fig. 5.20Fouling factor for grease lubrication – high purity

5.7.4 Viscosity ratio

The effectiveness of the lubricant is primarily given by the degree of separation of contact elements. The for-mation of adequate lubricating film is subject to the given minimal viscosity that the lubricant must possess, when the application achieves its operating temperature. A requirement for the formation of lubricating film is specified by the viscosity ratio κ, which is defined as the ratio between the real (actual) kinematic viscosity ν and the reference kinematic viscosity ν_1. The kinematic viscosity ν is the viscosity of the lubricant, when the given lubricant achieves its operating temperature.

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

In order to create sufficient lubricating film, the lubricant must maintain a certain minimal viscosity at opera-ting temperature. The bearing life may be increased by increasing the operating viscosity ν.

The reference kinematic viscosity can be determined from figure 5.4 or by using the following equations:

$$\nu_1 = 45\ 000 \cdot n^{-0.8} \cdot D_{pw}^{-0.5} \qquad \text{for } n < 1\ 000 \text{ min}^{-1}$$

$$\nu_1 = 45\ 000 \cdot n^{-0.5} \cdot D_{pw}^{-0.5} \qquad \text{for } n \geqq 1\ 000 \text{ min}^{-1}$$

$D_{pw} = 0.5 \cdot (d + D)$ is the bearing mean diameter

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5.7.5 Calculating the modified durability coefficient

The modified durability coefficient $a_{Dunlop\ BTL}$ may be easily determined from the following graphs:

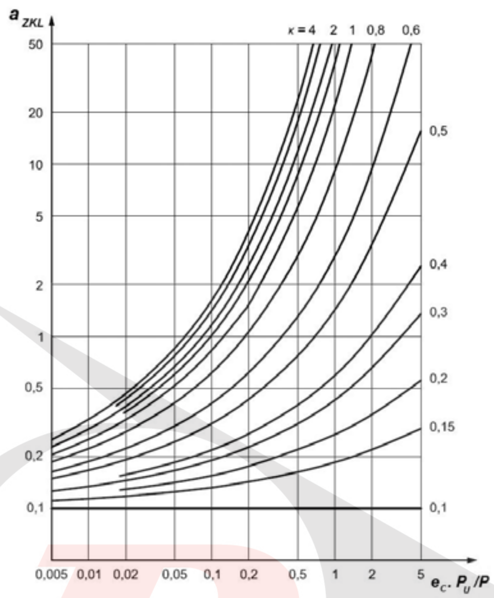


Fig. 5.21Coefficient of life modification factor for thrust ball bearings

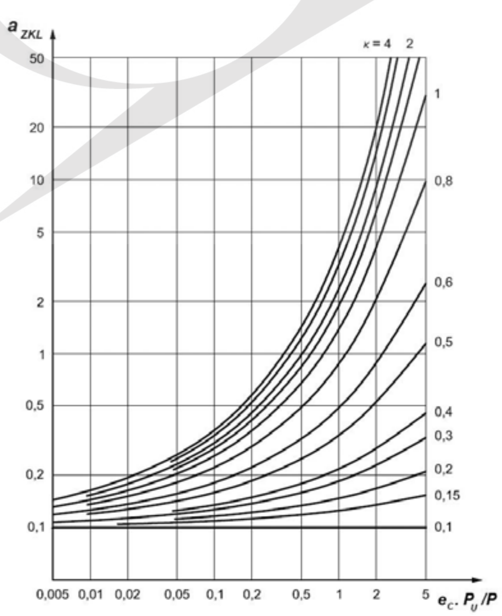


Fig. 5.22Coefficient of life modification factor for thrust rolling bearings

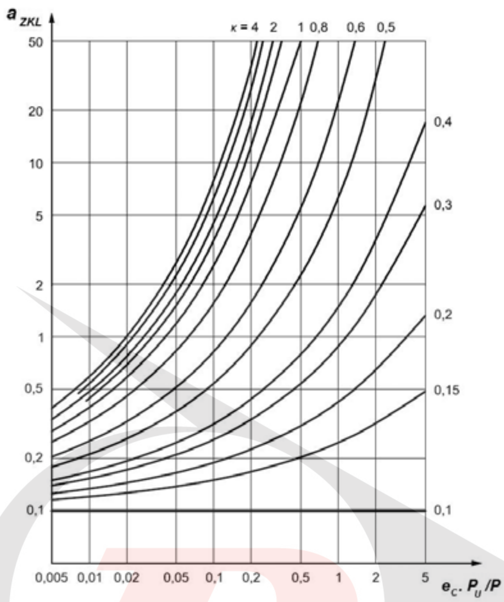


Fig. 5.23Coefficient of life modification factor for radial ball bearings

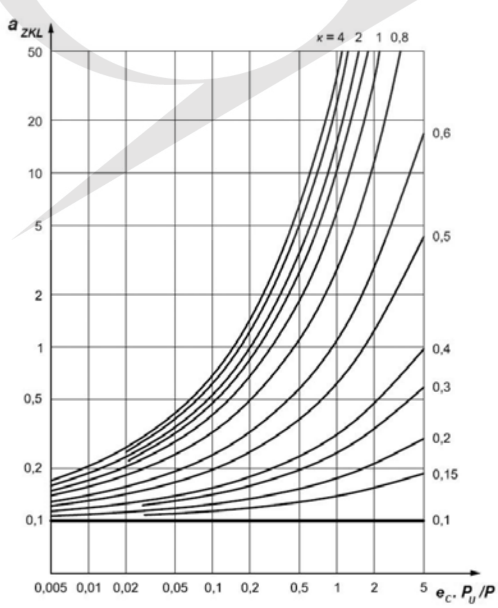


Fig. 5.24Coefficient of life modification factor for radial rolling bearing

5.8 Equivalent dynamic load

The bearing in the structural node is exposed generally to acting forces of various magnitudes at various revolution speeds and with various periods of action. In terms of the calculation method, the applied forces must be recalculated at constant load, during which the bearing has the same durability as achieved under actual load. This recalculated constant radial or axial load is called equivalent load P, or P_r (radial) or P_a (axial), resp.

5.8.1 Combined loads

Constant load method

The external forces applied on the bearing do not change in size or in relation to time.

Radial bearings

If constant radial or axial forces simultaneously act on a radial bearing, the following equation for calculating the radial dynamic load applies:

$$P_r = X \cdot F_r + Y \cdot F_a \qquad [kN]$$

P_r . . . . . . radial equivalent dynamic load [kN]

F_r . . . . radial force acting on the bearing [kN]

F_a . . . . axial force acting on the bearing [kN]

X . . . . radial load coefficient

Y . . . . axial load coefficient

Coefficients X and Y are dependent on the ratio F_a / F_r. The values X and Y are provided in the table or in the commentary preceding each structural group, where further information is provided for bearing calculations of the respective structural group.

Thrust bearings

Thrust ball bearings can only transfer forces acting axially and the following equation applied for calculating the axial equivalent dynamic load:

$$P_a = F_a \qquad [kN]$$

P_a. . . . . . . axial equivalent dynamic load [kN]

F_a . . . . axial bearing load [kN]

Spherical-roller thrust bearings can also transfer certain radial loads, however, only when a simultaneous axial load is applied, while observing the following condition:

$$P_a = F_a + 1.2 \cdot F_r \qquad [kN]$$

Variable loading method

A real variable load, whose time course is known, is replaced by a mean intended load to enable calculation. This intended load has the same effect on the bearing as an actual variable load.

5.8.2 Change in load magnitude at constant revolution speed

If a load acts on a bearing in a constant direction, whose size changes in relation to time, while the revolution speed is constant (fig. 5.25), we calculate the mean intended load F_s according to the equation

$$F_s = \sum_{i=1}^{n} F^3 \cdot \left[\frac{q_i}{100} \right]^{\frac{1}{3}} \qquad [kN]$$

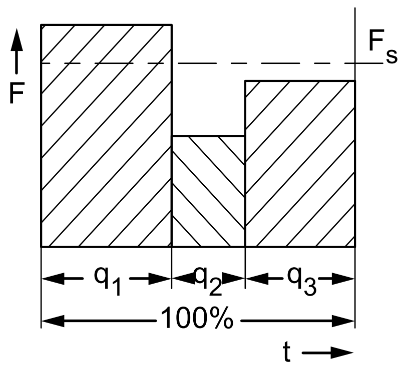
F_s	intended mean constant load	[kN]
$F_s = F_1, ..., F_n$	constant partial actual load	[kN]
$q_i = q_1, ..., q_n$	proportion of partially acting loads	[%]

If a variable load acts on a bearing, while the rotation speed meanwhile changes (fig. 5.26), we calculate the mean intended load using the equation

$$F_s = \frac{F_{min} + 2 \cdot F_{max}}{3} \qquad [kN]$$

Provided that the actual load has a sinusoid shape (fig. 5.27), the mean intended load is given by

$$F_s = 0.75 \cdot F_{max} \qquad [kN]$$



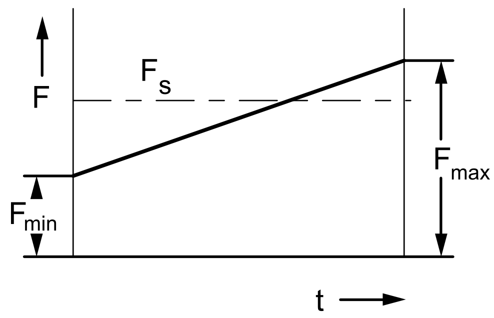


Fig. 5.25

Fig. 5.26

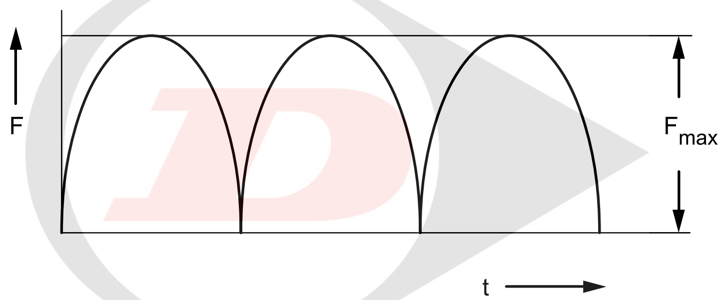


Fig. 5.27

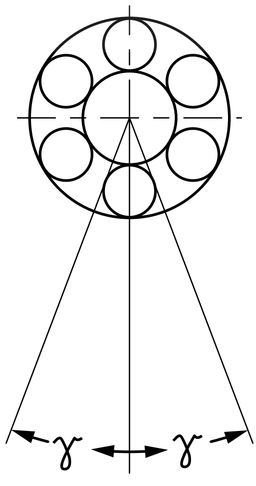


Fig. 5.28

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5.8.3 Change in load magnitude when rotation speed changes

If a variable load acts on a bearing along with variable rotation speed, the intended mean load is derived from the equation

$$F_s = \left[\frac{\sum_{i=1}^{n} F^3 \cdot q_i \cdot n_i}{\sum_{i=1}^{n} q_i \cdot n_i}\right]^{\frac{1}{3}} \qquad [kN]$$

$n_i = n_1, \ldots, n_n$ . . . constaion of partially acting loads and frequencies [min^{-1}]

[%]

If the rotation speed only changes in relation to time, the intended mean rotation speed is calculated using the equation

$$n_s = \left[\frac{\sum_{i=1}^{n} q_i \cdot n_i}{100}\right] \qquad [min^{-1}]$$

n_s . . . . mean rotation speed [min^{-1}]

5.8.4 Oscillating motion of the bearing

During oscillating motion with oscillating amplitude γ (fig. 5.28), it is easiest to substitute the oscillating motion by the notion of rotation, provided that the frequency of rotation is equal to the oscillating frequency. For radial bearings, we calculate the mean intended load using the equation

$$F_s = F_r \cdot \left[\frac{\gamma}{90}\right]^{\frac{1}{P}} \qquad [kN]$$

F_s ....mean intended load [kN]

F_r ....actual radial load [kN]

γ .... amplitude of oscillation [°]

p .... ball bearing exponent p = 3

For roller, needle roller, spherical-roller, and tapered-roller bearings $p = \frac{10}{3}$

5.9 Effect of temperature

The supplied range of bearings is designated for use in environments with a temperature of up to 120 °C. Larger spherical roller bearings are manufactured, by default, for operation in temperatures up to 200 °C. The exception are particular double row spherical roller bearings with polyamide races and single row ball bearings equipped with seals (RS, 2RS, RSR, 2RSR), which may be used short-term in temperatures up to 150 °C. More information about these bearings is available in chapter 12 "Manufacturer data".

Rolling bearings designed for higher operating temperatures are manufactured to ensure their required physical and mechanical properties and dimensional stability. Housing solutions at higher operating tempe-ratures should be consulted with the supplier.

The and dynamic load rating values C_r and C_a provided within the tables of the publication must, in the case of higher operating temperatures, be multiplied by the coefficient f_t, as specified in table 5.9.

Table 5.9

f_t Coefficient values				
operating temperature up to [°C]	150	200	250	300
f_t coefficient	0,95	0,9	0,75	0,6

5.10 Static Load Rating

The radial static load rating C_{or} and axial static load rating C_{oa} for each bearing is specified in the table section of the publication. The values C_{or} and C_{oa} were determined by calculation according to international standard ISO 76.

The static load rating is the load that corresponds to the calculated contact stress in the roller element and raceway contact zone, under the greatest load.

- 4600 MPa for double row self-aligning ball bearings
- 4200 MPa for other ball bearings
- 4000 MPa For roller, needle roller, spherical roller, and tapered roller bearings

This stress permanently deforms the rolling elements and raceways by approximately 0.0001 the diameter of the rolling element. The load is purely radial for radial bearings and purely axial within the bearing axis for thrust bearings.

The static load rating C_{or} is used for calculations, if the bearings

- rotate at very low speeds (n < 10 min^{-1})
- perform very slow oscillating motions
- under load do not move for a particular, extended period.

It is equally very important to check the safety in short-acting loads, such as e.g. shock loads and peak loads that act on a rotating bearing (dynamic load) or on a stationary bearing.

The maximum load that can act on a bearing should be used when calculating the equivalent static load of a bearing.

5.10.1 Equivalent static load

The equivalent static load is the recalculated radial load P_{or} for radial bearings and the axial load P_{oa} for thrust bearings.

$$P_{or} = X_0 \cdot F_r + Y_0 \cdot F_a \qquad [kN]$$

$$P_{oa} = Y_0 \cdot F_a \qquad [kN]$$

P_{or} . . . . . .radial equivalent static load [kN]

P_{oa} . . . . . .axial equivalent static load [kN]

F_r . . . . . . .radial load [kN]

F_a . . . . . . .axial load [kN]

X_0 . . . . . .radial load coefficient

Y_0 . . . . . .axial load coefficient

Table 5.10

s_0 Coefficient			
Bearing motion	Load bearing method, bearing operation requirements	s_0 Ball bearings	s_0 Cylindrical roller, needle-roller, spherical-roller, and tapered-roller bearings
rotational	significant impact loads, high demands on quite operation	2	4
	after static loading, bearing turns at lower loads	1,5	3
	normal demands for quiet operation		
	normal operating conditions and normal operating requirements	1	1,5
	quiet operation without vibration(s)	0,5	1
Oscillating	small oscillating angle with large frequency with occasional uneven loads	2	3,5
	large oscillating angle with small frequency with relatively constant periodical loads	1,5	2,5
non-rotating (at rest)	considerable impact loads	1,5 to 1	3 to 2
	normal and low loads, bearing operation unburdened by increased demands	1 to 0,4	2 to 0,8
	spherical-roller thrust bearings during all types of motion and loading	-	4

Coefficients X_0 and Y_0 are specified in the table section of the publication. Detailed information is also provi-ded here for determining the equivalent static load of bearings of a particular structural group.

5.10.2 Bearing safety during static loading

In practice, the bearing safety under static load is determined from the ratio C_{or}/P_{or} or C_{oa}/P_{oa} and compared with the data in table 5.10, where the smallest permissible coefficient values S_0 are specified for various operating conditions.

$$S_0 = \frac{C_{or}}{P_{or}} \quad \text{and/or} \quad \frac{C_{oa}}{P_{oa}}$$

S_0. . . . . . .safety coefficient under static load [kN]

C_{or} . . . . . .radial dynamic load capacity [kN]

C_{oa} . . . . . .axial dynamic load capacity [kN]

P_{or} . . . . . .radial equivalent static load or max. acting force F_{rmax} (fig. 5.29) under significant impact load, resp. [kN]

P_{oa} . . . . . .axial equivalent static load or max. acting force F_{rmax} (fig. 5.29) under significant impact load, resp. [kN]

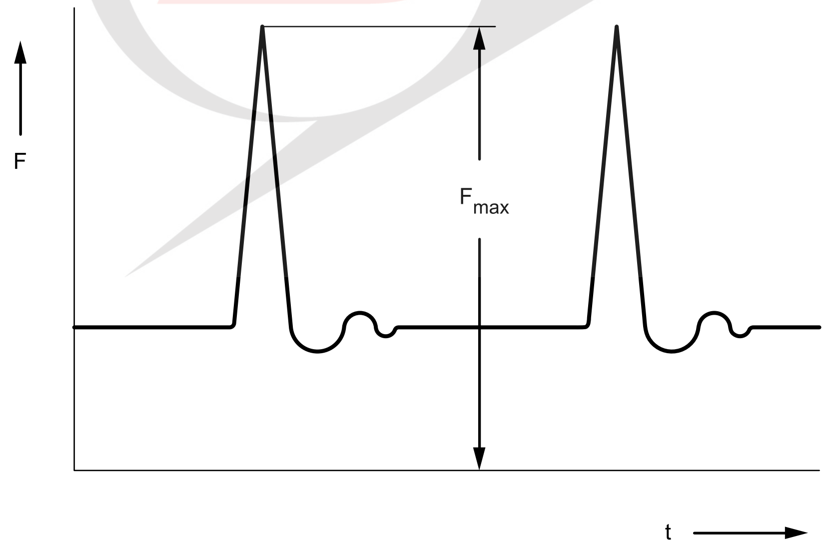

Fig. 5.29

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6. CRITICAL SPEED AND VIBRATIONS

The operating speeds at which bearings can operate are limited by the operating temperature of the lubricant used or by the material of individual bearing components, resp.

The critical operating temperature then depends on the head induced by friction in the bearing and the amount of heat that can be dissipated from the bearing.

6.1 Bearing friction

The friction in the bearing depends on the load, the operating speed, the lubricant, and bearing type and size. Friction then significantly affects the generation of heat in the bearing and hence its operating temperature.

The total rolling resistance in the bearing is given by the sum of:

- the rolling and sliding friction at all contact points (rolling contact, contact between rolling elements and cage or guiding surfaces, resp.)
- friction in the lubricant
- the sliding friction of the friction seal, as applicable

6.1.1 Torque friction estimate

The friction torque can be determined, e.g. using the following relationship:

$$M = 0.5 \cdot \mu \cdot P \cdot d$$

M ...... bearing friction torque [N·mm]

μ .... constant bearing friction coefficient (see table 6.1) [-]

P .... equivalent dynamic bearing load [N]

d .... bearing bore diameter [mm]

The given relationship applies with sufficient accuracy assuming proper lubrication, normal operating condi-tions and bearing load P = 0.1·C.

6.1.2 Calculating frictional torque

Total frictional torque M [N · mm] consists of hydrodynamic frictional torque M_0 [N · mm] of an unloaded bearing, which arises when rotating parts wade in a viscous environment and from rolling friction torque M_1 [N · mm]:

$$M = M_0 + M_1$$

Table 6.1

Bearing type	Coefficient of friction μ
Ball bearings	0,0015
Angular-contact ball bearings	
- single-row	0,0020
- double-row	0,0024
- four-point	0,0024
Self-aligning ball bearings	0,0010
Cylindrical roller bearings	
- with cage while $F_a = 0$	0,0011
- complete with rollers while $F_a = 0$	0,0020
Tapered-roller bearings	0,0018
Spherical-roller bearings	0,0018
Thrust ball bearings	0,0013
Thrust cylindrical roller bearings	0,0050
Spherical-roller thrust bearings	0,0018

Hydrodynamic frictional torque depends on lubrication, bearing size and speed:

$$M_0 = f_0 \cdot d_m^3 \cdot (\nu \cdot n)^{k_0}$$

f_0 . . . . . . constant lubrication for bearings of same series, design, and precision [-]

d_m . . . . . . bearing mean diameter [mm]

ν . . . . kinematic viscosity of lubricant [$mm^2 \cdot s^{-1}$]

n . . . . revolutions [min^{-1}]

k_0 . . . . . . constant equal to 2/3 [-]

The rolling friction torque depends on load, the static load, and bearing size:

$$M_1 = f_\alpha' \cdot F \cdot d_m \cdot (F/C_0)^c$$

f_α' . . . . . . function of the load bearing direction for bearings of same series, design, and precision [-]

F . . . . load [N]

d_m . . . . . . bearing mean diameter [mm]

C_0 . . . . . . static load rating of bearing [N]

c . . . . experimentally determined exponent [-]

A more accurate computational model takes into account four sources of friction:

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

M	...... total frictional torque	[N·mm]
M_{rr}	...... rolling friction torque	[N·mm]
M_{sl}	...... sliding friction torque	[N·mm]
M_{seal}	.... frictional torque within the bearing	[N·mm]
M_{drag}	.... frictional torque caused by wading	[N·mm]

The calculation using this model, however, is considerably complicated.

6.2 Limiting speed

Bearing operating speeds are limited by the bearing internal design, their precision and size, bearing clearan-ce, method of lubrication and loading design, which affect the dissipation of heat, generated by the bearing. Due to the specified influences, proper attention should be given when designing a suitable bearing.

By limiting speed, we mean the revolutions during which, under given operating conditions, a thermal equilibrium is created between the heat generated in the bearing and the head released from the bearing.

We are able to state, on the basis of experimental tests and practical applications, that there is a maxi-mum speed that should not be exceeded for technical or economic reasons that are required to maintain the operating temperature at an acceptable level.

If the bearing is to operate at speeds that exceed the limiting speed, the lubrication, method of heat dissi-pation, the cage design, or the entire bearing design, resp. need to be modified. Manufacturers, for example, recommend that high speed bearings be designed with advanced precision or with the use of a sturdy cage guided on one of the bearing rings and with the use of oil or oil-mist lubrication.

6.2.1 Definition of Dunlop BTL limiting speed

The catalogue tables specify the limiting speeds that are defined as the thermal reference speeds in accor-ding with ISO 15312:2003. The reference conditions that determine the thermal equilibrium are: A tempera-ture increase by 50 °C above the ambient temperature and a 5 % bearing static load range. These conditions apply for opened bearings with normal radial clearance.

Limiting speeds of rolling bearings, as specified in the catalogue tables, are reference speeds for oil lub-rication without EP additives with a kinematic viscosity at a temperature of 70 °C as follows: 12 mm^2/s or 24 mm^2/s, resp. for line-contact thrust bearings.

Limiting speeds for grease lubrication are approximately 20 % lower.

The limiting speed is calculated using the following conditions of thermal equilibrium:

$$n_{mez} = \frac{(W_s (T_{(D,max)} - T_o) - \Sigma Q_i)}{j \cdot M}$$

W_s . . . . . cooling coefficient

$T_{D,max}$ . . . . max. temperature on outer ring

T_o . . . . . . ambient temperature

Q . . . . . . heat

j . . . . mechanical equivalent

M . . . . . . total frictional torque

After modification and substitution, we arrive at the limiting speed equation:

$$n_{mez}^{\frac{5}{3}} + n \cdot \frac{f_{\alpha}' \cdot F \cdot \left(\frac{F}{C_0}\right)^c}{f_0' \cdot d_m^2 \cdot \nu^{\frac{2}{3}}} - \frac{W_s (T_{(D,max)} - T_o) - \Sigma Q_i}{j \cdot f_0' \cdot d_m^3 \cdot \nu^{\frac{2}{3}}} = 0$$

f_0' . . . . . . function of bearing lubrication effect of same series, design, and precision [-]

The given equation has only one real root, while this root physically corresponds to the value of the limiting speed.

The limiting speed values can be approximately determined according to the following relationships:

- for radial bearings:

$$n_{mez} = \frac{(A \cdot f)}{d_m}$$

A . . . . coefficient dependent on the bearing series and lubricant [-]

f . . . . bearing loading and size effect function [-]

- for thrust bearings:

$$n_{mez} = \frac{(A \cdot f)}{\sqrt{(D \cdot H)}}$$

D . . . . . . bearing external diameter [mm]

H . . . . . . bearing height [mm]

Experimentally, the limiting revolution speed is then determined during radial loading, which corresponds to the durability L_h = 104 ÷ 105 hours such that the speed gradually changes and the steady temperature on the bearing outer ring is recorded. The limiting speed is then determined as the intersection point of the linear estimate of measured values and the limiting reference values (fig. 6.1).

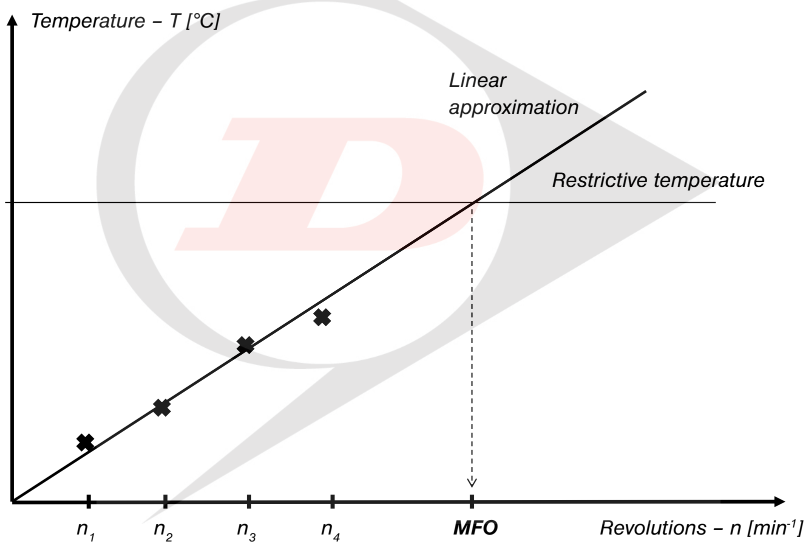

Fig. 6.1

Coefficient A for determining the approximate limiting speed is specified in table 6.2

6.2.2 Special operating speeds

When operating speeds are very low, the elastohydrodynamic lubrication film is not formed at the rolling contact site. Such loadings subsequently require the use of lubricant with EP additives.

Table 6.2

Bearing type	Coefficient A
Single-row ball	500 000
Single-row ball with RS and 2RS glands	300 000
Single-row angular-contact ball	
$\alpha \leq 15°$	500 000
$\alpha = 26°$	420 000
$\alpha = 40°$	400 000
Double-row angular-contact ball	320 000
Single-row cylindrical roller	500 000
Double-row cylindrical roller	500 000
Double-row spherical-roller, except for series 232	250 000
Double-row spherical-roller series 232	200 000
Tapered-roller, except for series 313	250 000
Tapered-roller series 313	200 000
Thrust ball	100 000
Thrust cylindrical roller	100 000
Spherical-roller thrust	200 000

Oscillation motions are another special case. In this type of motion, the direction of rotation changes before the bearing completes one revolution. The speed is zero at the moment the direction of rotation changes and, as such, the hydrodynamic lubricating film is not preserved. The lubricating film is formed, in such case, in the area of mixed lubrication. The limiting speed cannot be determined for oscillations, because the upper threshold is not determined by thermal equilibrium, by non-inertial forces. There is a risk that inertia may cause short-term slippage of rolling elements and damage to orbits each time the direction of rotation changes. Permissible acceleration or deceleration, resp. depends on the mass of the rolling elements and the cage, the lubrication, and the bearing loading.

6.3 Vibrations in the bearing

Sensing of vibrations is generally related to the propagation of noise. The bearing, however, is usually not the source of noise. Noise is just an audible effect of vibrations that are caused either directly or indirectly by the bearing on related components. It is the reason why the majority of noise-related issues are associated with vibrations of the bearing itself or the entire housing.

The number of rolling elements, which carry the load, changes during operation in radially loaded bea-rings. This effect causes a displacement in the direction of the load. While the resulting vibrations cannot be prevented, they may be reduced by introducing an axial preload that ensures loading of all rolling elements.

Roll-over of damaged bearing components occurs in cases of local damage to raceways or rolling elements, resp., which occurs during improper handling or incorrect installation, and it leads to vibrations. The source of vibrations (damaged component) can be determined using vibration frequency analysis.

Penetration of contaminants into the bearing may occur in bearings that operate in contaminated envi-ronments when rolling elements roll over the contaminants. The size of induced vibrations depends on the quantity, size, and structure of the contaminants. This does not generate typical frequencies, but an audible noise may be heard.

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6.3.1 Frequency characteristics of bearings

The frequency of vibration impulses created by toss-over of damaged bearing components has a simple relationship to the internal bearing geometry and to the frequency of shaft revolutions. These relationships can be described using equations that define the frequency of defects of individual bearing components. The specified equations assume optimal conditions, because they do not account for slippage of rolling elements. The equation for ball defects presupposes that the defect touches both the inner and outer ring per revolution of the rolling element.

The frequency during a defect on the outer ring (BPFO)

$$BPFO = z/2 \cdot n \cdot (1 - D_w/d_m \cdot \cos \alpha)$$

The frequency during a defect on the inner ring (BPFI)

$$BPFI = z/2 \cdot n \cdot (1 + D_w/d_m \cdot \cos \alpha)$$

The frequency during a ball- or roller defect (BSF).

$$BSF = d_m / 2D_w \cdot n \cdot (1 - (D_w/d_m)^2)$$

Frequency during a cage defect (FTF)

$$FTF = n/2 \cdot (1 - D_w/d_m \cdot \cos \alpha)$$

D_w . . . . roller element diameter (mm) [mm]

d_m . . . . . bearing pitch diameter (mm) [mm]

z . . . . . number of rolling elements

n . . . . . shaft rotation frequency [s^{-1}]

α . . . . . contact angle

Vibration frequency analyses help determine, which bearing component is damaged. We recommend that the customer coordinates with Dunlop BTL Technical and Consulting Services Department when calculating frequency characteristics.

6.3.2 Influence of the bearing on housing vibrations

The rigidity of the bearing is, in many housings, of the same order as the rigidity of related components. Housing vibrations can be reduced by the proper selection of the bearing, the arrangement of bearings in the housing, and by using a suitable preload or clearance. If the vibrations cannot be eliminated by the selective use of the bearing, its arrangement within the housing, the vibrations may also be reduced by additional modifications of the housing, e.g. by inserting a rubber spacer that will dampen the vibrations or any other structural modification that will eliminate the source of critical vibrations.

7. BEARINGS – GENERAL DATA

7.1 Bearing design data

Besides the suitable type of bearing and the size of it, additional design characteristics that define the bea-ring in location design have to be defined. The location designed is the one usually responsible for the bearing design. This person has to consider the requirements for accuracy of run, service temperature and lubrication, as well as the assembly and disassembly method. In order to meet all different requirements for proper run of bearing, bearings are produced in many versions that are characterized with an additional identification of bearings. Thus, bearings with required tolerances, clearances, materials, cage design or sealing can be selec-ted. Also, accordingly with the identification system, bearings can be specified for certain service conditions that may be characteristic with high revolutions or high temperature, or alternatives of bearings for certain locations can be selected by the knowledge of identification of other bearing manufacturers.

7.2 Main dimensions

Rolling bearings are supplied as a final machine part, and the designer has at disposal fixed dimensions that ensure easy exchangeability. Standardisation applies to outer dimensions important in the assembly point of view. It is convenient for manufacturers and users of bearings for technological and thus also economic reasons. It however does not state inner dimensions, such as the quantity and dimensions of rolling bodies, or designs of cages. Despite that, due to the long-term development and various design and production technology optimisations even the inner design of bearings becomes united to a significant extent.

The ISO international organization came up with dimension plans for roller bearings of metric dimensions that are defined in the below listed documents:

- ISO 15:1998 applies to radial roller bearings of metric dimensions, with the exception of tapered bearings;
- ISO 355:1997 applies to radial tapered bearings of metric dimensions;
- ISO 104:2002 applies to thrust roller bearings of metric dimensions;
- ISO 582:1995 applies to maximum values of bevelling the assembly edges of bearings.

7.2.1 ISO dimension plans

ISO dimension plan allocates to each bearing hole diameter d multiple outer diameters D, and to those dif-ferent widths B – or – more precisely - T for radial and H for thrust bearings. Bearings with the same hole diameter and same outer diameter belong in one diameter row identified by ascending outer diameter with figures 7, 8, 9, 0, 1, 2, 3, 4. Every diameter row contains bearings of different width rows by ascending width: 8, 0, 1, 2, 3, 4, 5, 6 and 7 for radial bearings. Width rows of radial bearings correspond with height rows of thrust bearings (height rows by ascending height 7, 9, 1 and 2).

Combining the diameter and width row creates dimension rows that are identified by double figure where the first figure identified the width row, and the second figure identifies the diameter row. This system is clearly indicated in fig. 7.1.

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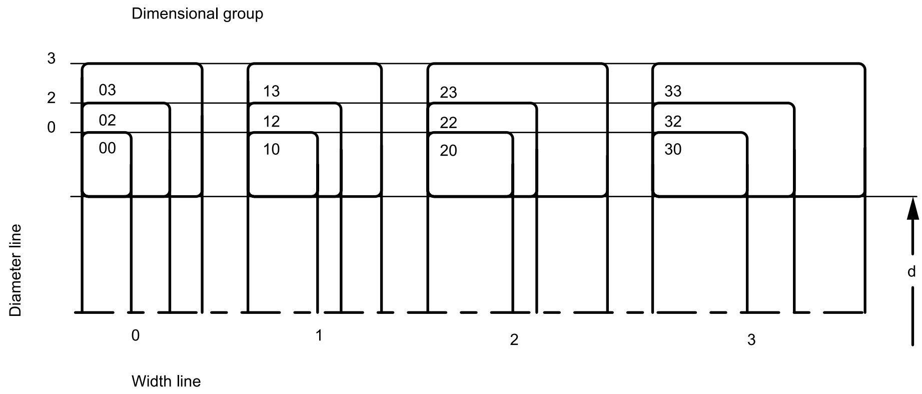

Fig. 7.1

The ISO dimension plan also contains dimensions of bearing ring edge fillet, the so-called installation fillet (fig. 7.2). The chart section of the catalogue indicates minimum installation fillet values for individual bearing types that you need to know when designing radiuses of transmission of components forming the bearing location.

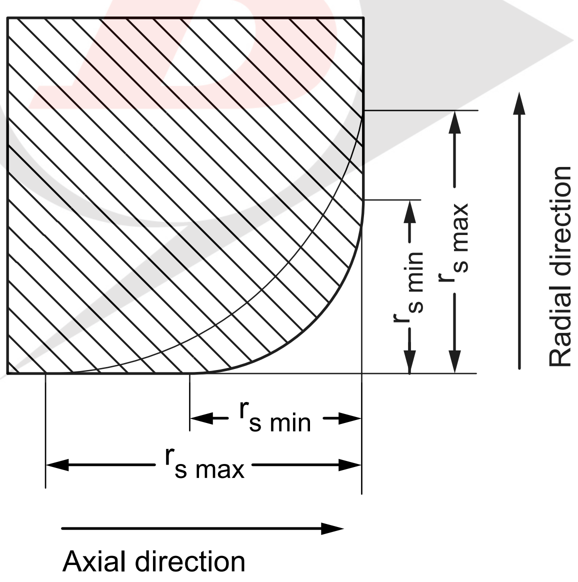

Fig. 7.2

See Chart 7.1 for an overview of the installation fillet complying with the international standard ISO 582.

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Table 7.1

Limit dimensions of installation fillet									
	Radial bearings except tapered				Tapered bearings				Thrust bearings
$r_{s\ min}$	d or D		$r_{s\ min}$		d or D		$r_{s\ min}$		$r_{s\ min}$
mm	over	to	in radial direction	in axial direction	over	to	in radial direction	in axial direction	in radial and axial direction
0,15	-	-	0,3	0,6	-	-	-	-	0,3
0,2	-	-	0,5	0,8	-	-	-	-	0,5
0,3	-	40	0,6	1	-	40	0,7	1,4	0,8
	40	-	0,8	1	40	-	0,9	1,6	0,8
0,6	-	40	1	2	-	40	1,1	1,7	1,5
	40	-	1,3	2	40	-	1,3	2	1,5
1	-	50	1,5	3	-	50	1,6	2,5	2,2
	50	-	1,9	3	50	-	1,9	3	2,2
1,1	-	120	2	3,5	-	-	-	-	2,7
	120	-	2,5	4	-	-	-	-	2,7
1,5	-	120	2,3	4	-	120	2,3	3	3,5
	120	-	3	5	120	250	2,8	3,5	3,5
	-	-	-	-	250	-	3,5	4	3,5
2	-	80	3	4,5	-	120	2,8	4	4
	80	220	3,5	5	120	250	3,5	4,5	4
	220	-	3,8	6	250	-	4	5	4
2,1	-	280	4	6,5	-	-	-	-	4,5
	280	-	4,5	7	-	-	-	-	4,5
2,5	-	100	3,8	6	-	120	3,5	5	-
	100	280	4,5	6	120	250	4	5,5	-
	280	-	5	7	250	-	4,5	6	-
3	-	280	5	8	-	120	4	5,5	5,5
	280	-	5,5	8	120	250	4,5	6,5	5,5
	-	-	-	-	250	400	5	7	5,5
	-	-	-	-	400	-	5,5	7,5	5,5
4	-	-	6,5	9	-	120	5	7	6,5
	-	-	-	-	120	250	5,5	7,5	6,5
	-	–	–	-	250	400	6	8	6,5
	-	-	-	-	400	-	6,5	8,5	6,5
5	-	-	8	10	-	180	6,5	8	8
	-	-	-	-	180	-	7,5	9	8
6	-	-	10	13	-	180	7,5	10	10
	-	-	-	-	180	-	9	11	10
7,5	-	-	12,5	17	-	-	-	-	12,5
9,5	-	-	15	19	-	-	-	-	15
12	-	-	18	24	-	-	-	-	18
15	-	-	21	30	-	-	-	-	21

7.2.2 Accuracy of bearings

Accuracy of bearings means accuracy of bearing dimensions and run. Bearings are made in the accuracy classes P0, P6, P5, P5A, P4, P4A, P2, SP and UP. The P0 accuracy is general, and is not stated in the bearing identification. Descending number in the identification indicates higher bearing accuracy.

Majority locations can utilise roller bearings of normal accuracy level. Bearings with higher accuracy level are used in locations that require higher running accuracy, such as location of machine tool spindles, and where bearings exceed their limit revolutions.

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The limit dimension and run accuracy values are stated in charts 7.2 to 7.12. These values comply with inter-national standards ISO 492 a ISO 199. The P5A and P4A designation is used for bearings made in relevant accuracy level P5 and P4 but selected parameters feature higher accuracy level than is P5 and P4.

Symbols of quantities and their meaning

d . . . . nominal bore diameter

d_1. . . . . . . nominal diameter of bigger theoretical tapered bore diameter

d_2. . . . . . . nominal diameter of shaft ring of double direction thrust bearings

Δd_s. . . . . . deviation of individual bore diameter from nominal dimension

Δd_{mp}. . . . . deviation of mean diameter of cylindrical bore in individual radial plane (for

tapered bore applies Δdmp for theoretical bore diameter)

Δd_{1mp} . . . . deviation of mean theoretical tapered bore diameter

Δd_{2mp} . . . . deviation of mean shaft ring bore diameter of double direction thrust bearings in

individual radial plane radial plane

V_{dp} . . . . . . dispersion of individual bore diameter in individual radial plane

V_{dmp} . . . . . dispersion of mean cylindrical bore diameter

V_{d2p} . . . . .dispersion of shaft ring bore diameter of double direction thrust bearings in individual

radial plane

D . . . . nominal external diameter

ΔD_s . . . . . deviation of individual outer diameter from nominal dimension

ΔD_{mp}. . . . . deviation of mean diameter of cylindrical surface in individual radial plane

VD_p . . . . . dispersion of individual outer cylindrical surface diameter in individual radial plane

VD_{mp}. . . . . dispersion of mean outer cylindrical bore diameter

B . . . . nominal inner ring width

T . . . . nominal total width of tapered bearings

T_1. . . . . . . nominal effective width of inner semi-unit

T_2. . . . . . . nominal effective width of outer semi-unit

ΔB_s . . . . . deviation of individual inner ring width

ΔC_s. . . . . . deviation of individual outer ring width

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ΔT_s . . . . . . deviation of (total) individual bearing width

ΔT_{1s} . . . . . deviation of effective width of inner semi-unit

ΔT_{2s} . . . . . deviation of effective width of outer semi-unit

C . . . . nominal outer ring width

V_{Bs} . . . . . . dispersion of individual inner ring width

V_{Cs} . . . . . . dispersion of individual outer ring width

K_{ia} . . . . . . radial runout of assembled bearing inner ring

K_{ea} . . . . . . radial runout of assembled bearing outer ring

S_i . . . . axial runout of shaft ring raceway

S_e. . . . . . . axial runout of body ring raceway

S_{ia} . . . . . . axial runout of basic front of assembled bearing inner ring

S_{ea} . . . . . . axial runout of basic front of assembled bearing outer ring

S_d. . . . . . . axial runout of basic front

S_D. . . . . . .runout of outer surface against ring front

S_s. . . . . . . runout of inner ring support front against basic front for single row tapered bearings

Limit values of individual parameters for different accuracy levels are stated in the below charts.

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Table 7.2

Accuracy of dimensions and run of radial bearings (except tapered) – Accuracy level P0																
Inner ring																
d		Δ_{dmp}		V_{dp} diameter rows			V_{dmp}	K_{ia}	Δ_{Bs}		V_{Bs}	Δ_{dmp}		Δ_{d1mp}	$-\Delta_{dmp}$	$V^{1)}_{dp}$
				7,8,9	0,1	2,3,4										
over	to	max	min	max	max	max	max	max	max	min	max	max	min	max	min	max
mm		µm														
2,5	10	0	-8	10	8	6	6	10	0	-120	15	-	-	-	-	-
10	18	0	-8	10	8	6	6	10	0	-120	20	-	-	-	-	-
18	30	0	-10	13	10	8	8	13	0	-120	20	21	0	21	0	13
30	50	0	-12	15	12	9	9	15	0	-120	20	25	0	25	0	15
50	80	0	-15	19	19	11	11	20	0	-150	25	30	0	30	0	19
80	120	0	-20	25	25	15	15	25	0	200	25	35	0	35	0	25
120	180	0	-25	31	31	19	19	30	0	-250	30	40	0	40	0	31
180	250	0	-30	38	38	23	23	40	0	-300	30	46	0	46	0	38
250	315	0	-35	44	44	26	26	50	0	-350	35	52	0	52	0	44
315	400	0	-40	50	50	30	30	60	0	-400	40	57	0	57	0	50
400	500	0	-45	56	56	34	34	65	0	-450	50	63	0	63	0	56
500	630	0	-50	63	63	38	38	70	0	-500	60	-	-	-	-	-
630	800	0	-75	-	-	-	-	80	0	-750	70	-	-	-	-	-
800	1000	0	-100	-	-	-	-	90	0	-1000	80	-	-	-	-	-
1000	1250	0	-125	-	-	-	-	100	0	-1250	100	-	-	-	-	-

Table 7.3

Outer ring										
D		ΔD_{mp}		V_{DP} Diameter rows				V_{Dmp}	K_{ea}	Δ_{Cs}, Δ_{Cs}
				7,8,9	0,1	2,3,4	bearings 2) with shields			
přes	do	max	min	max	max	max	max	max	max	
mm		µm								
6	18	0	-8	10	8	6	10	6	15	
18	30	0	-9	12	9	7	12	7	15	
30	50	0	-11	14	11	8	16	8	20	
50	80	0	-13	16	13	10	20	10	25	
80	120	0	-15	19	19	11	26	11	35	
120	150	0	-18	23	23	14	30	14	40	
150	180	0	-25	31	31	19	38	19	45	
180	250	0	-30	38	38	23	-	23	50	
250	315	0	-35	44	44	26	-	26	60	
315	400	0	-40	50	50	30	-	30	70	
400	500	0	-45	56	56	34	-	34	80	
500	630	0	-50	63	63	38	-	38	100	
630	800	0	-75	94	94	55	-	55	120	
800	1000	0	-100	125	125	75	-	75	140	
1000	1250	0	-125	-	-	-	-	-	160	
1250	1600	0	-160	-	-	-	-	-	190	

1) Applies in optional radial bore plane
2) Applies only to bearings of diameter rows 2, 3 and 4
3) Corresponds with Δ_{Bs}, V_{Bs} of inner race of the same bearing

Table 7.4a

Accuracy of dimensions and run of radial bearings (except tapered) – Accuracy level P6											
Inner ring											
d		Δ_{dmp}		V_{dp} Diameter rows			V_{dmp}	K_{ia}	Δ_{Bs}		V_{Bs}
				7,8,9	0,1	2,3,4					
over	to	max	min	max	max	max	max	max	max	min	max
mm		µm									
2,5	10	0	-7	9	7	5	5	6	0	-120	15
10	18	0	-7	9	7	5	5	7	0	-120	20
18	30	0	-8	10	8	6	6	8	0	-120	20
30	50	0	-10	13	10	8	8	10	0	-120	20
50	80	0	-12	15	15	9	9	10	0	-150	25
80	120	0	-15	19	19	11	11	13	0	-200	25
120	180	0	-18	23	23	14	14	18	0	-250	30
180	250	0	-22	28	28	17	17	20	0	-300	30
250	315	0	-25	31	31	19	19	25	0	-350	35
315	400	0	-30	38	38	23	23	30	0	-400	40
400	500	0	-35	44	44	26	26	35	0	-450	45
500	630	0	-40	50	50	30	30	40	0	-500	50

Table 7.4b

Outer ring										
D		ΔD_{mp}		V_{DP} Diameter rows				V_{Dmp}	K_{ea}	Δ_{Cs}, V_{Cs}
				7,8,9	0,1	2,3,4	bearings[1] with shields			
over	to	max	min	max	max	max	max	max	max	
mm		µm								
6	18	0	-7	9	7	5	9	5	8	
18	30	0	-8	10	8	6	10	6	9	
30	50	0	-9	11	9	7	13	7	10	
50	80	0	-11	14	11	8	16	8	13	
80	120	0	-13	16	16	10	20	10	18	
120	150	0	-15	19	19	11	25	11	20	
150	180	0	-18	23	23	14	30	14	23	
180	250	0	-20	25	25	15	-	15	25	
250	315	0	-25	31	31	19	-	19	30	
315	400	0	-28	35	35	21	-	21	35	
400	500	0	-33	41	41	25	-	25	40	
500	630	0	-38	48	48	29	-	29	50	
630	800	0	-45	56	56	34	-	34	60	
800	1000	0	-50	75	75	45	-	45	75	

[1] Applies only to bearings of diameter rows 0, 1, 2, 3 and 4
[2] Corresponds with Δ_{Bs}, V_{Bs} of the inner race of the same bearing

Table 7.5

Accuracy of dimensions and run of radial bearings (except tapered)												
Accuracy level P5												
Inner ring												
d		Δ_{dmp}		V_{dp} Diameter rows 7,8,9	V_{dp} Diameter rows 0,1,2,3,4	V_{dmp}	K_{ia}	S_d	S_{ia} [1]	Δ_{Bs}		V_{Bs}
over	to	max	min	max	max	max	max	max	max	max	min	max
mm		µm										
2,5	10	0	-5	5	4	3	4	7	7	0	-40	5
10	18	0	-5	5	4	3	4	7	7	0	-80	5
18	30	0	-6	6	5	3	4	8	8	0	-120	5
30	50	0	-8	8	6	4	5	8	8	0	-120	5
50	80	0	-9	9	7	5	5	8	8	0	-150	6
80	120	0	-10	10	8	5	6	9	9	0	-200	7
120	180	0	-13	13	10	7	8	10	10	0	-250	8
180	250	0	-15	15	12	8	10	11	13	0	-300	10
250	315	0	-18	18	14	9	13	13	15	0	-350	13
315	400	0	-23	23	18	12	15	15	20	0	-400	15

Table 7.6

Outer ring											
D		ΔD_{mp}		V_{DP} Diameter rows 7,8,9	V_{DP} Diameter rows 0,1 2,3,4	V_{Dmp}	K_{ea}	S_D	S_{ea} [1]	Δ_{Cs}	V_{Cs}
over	to	max	min	max	max	max	max	max	max		max
mm		µm									
6	18	0	-5	5	4	3	5	8	8	It corresponds to the inner ring of the same bearing	5
18	30	0	-6	6	5	3	6	8	8		5
30	50	0	-7	7	5	4	7	8	8		5
50	80	0	-9	9	8	5	8	8	10		6
80	120	0	-10	10	8	5	10	9	11		8
120	150	0	-11	11	8	6	11	10	13		8
150	180	0	-13	13	10	7	13	10	14		8
180	250	0	-15	15	11	8	15	11	15		10
250	315	0	-18	18	14	9	18	13	18		11
315	400	0	-20	20	15	10	20	13	20		13
400	500	0	-23	23	17	12	23	15	23		15
500	630	0	-28	28	21	14	25	18	25		18
630	800	0	-35	35	26	18	30	20	30		20

[1] Applies to ball bearings only
[2] Does not apply to shielded bearings
[3] Corresponds with Δ_{Bc} of the inner ring of the same bearing

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Table 7.7

Accuracy of dimensions and run of radial bearings (except tapered)

Accuracy level P4

Inner ring

d		Δ_{dmp}		Δ_{ds} [1]		V_{dp} Diameter rows 7,8,9	V_{dp} Diameter rows 0,1,2,3,4	V_{dmp}	K_{ia}	S_d	S_{ia} [2]	Δ_{Bs}		V_{Bs}
over	to	max	min	max	min	max	max	max	max	max	max	max	min	max
mm		µm												
2,5	10	0	-4	0	-4	4	3	2	2,5	3	3	0	-40	2,5
10	18	0	-4	0	-4	4	3	2	2,5	3	3	0	-80	2,5
18	30	0	-5	0	-5	5	4	2,5	3	4	4	0	-120	2,5
30	50	0	-6	0	-6	6	5	3	4	4	4	0	-120	3
50	80	0	-7	0	-7	7	5	3,5	4	5	5	0	-150	4
80	120	0	-8	0	-8	8	6	4	5	5	5	0	-200	4
120	180	0	-10	0	-10	10	8	5	6	6	7	0	-250	5
180	250	0	-12	0	-12	12	9	6	8	7	8	0	-300	6

Table 7.8

Outer ring

D		ΔD_{mp}		V_{Ds} [1]		V_{DP} Diameter rows [3] 7,8,9	V_{DP} Diameter rows [3] 0,1 2,3,4	V_{Dmp}	K_{ea}	S_D	S_{ea} [2]	Δ_{Cs}	V_{Cs}
over	to	max	min	max	min	max	max	max	max	max	max		max
mm		µm											
6	18	0	-4	0	-4	4	3	2	3	4	5		2,5
18	30	0	-5	0	-5	5	4	2,5	4	4	5		2,5
30	50	0	-6	0	-6	6	5	3	5	4	5		2,5
50	80	0	-7	0	-7	7	5	3,5	5	4	5		3
80	120	0	-8	0	-8	8	6	4	6	5	6		4
120	150	0	-9	0	-9	9	7	5	7	5	7		5
150	180	0	-10	0	-10	10	8	5	8	5	8		5
180	250	0	-11	0	-11	11	8	6	10	7	10		7
250	315	0	-13	0	-13	13	10	7	11	8	10		7
315	400	0	-15	0	-15	15	11	8	13	10	13		8

[1] Applies only to bearings of diameter rows 0, 1, 2, 3 and 4
[2] Applies to ball bearings only
[3] Does not apply to shielded bearings
[4] Corresponds with Δ_{Bs} of the inner ring of the same bearing

Table 7.9

Accuracy of dimensions and run of roller bearings with tapered hole											
Accuracy level SP											
Inner ring											
d		Δ_{dmp}		Δ_{d1mp}	$-\Delta_{dmp}$	V_{dp}	K_{ia}	S_d	Δ_{Bs}		V_{Bs}
over	to	max	min	max	min	max	max	max	max	min	max
mm		µm									
18	30	10	0	4	0	3	3	8	0	-100	5
30	50	12	0	4	0	4	4	8	0	-120	5
50	80	15	0	5	0	5	4	8	0	-150	6
80	120	20	0	6	0	5	5	9	0	-200	7
120	180	25	0	8	0	7	6	10	0	-250	8
180	250	30	0	10	0	8	8	11	0	-300	10
250	315	35	0	12	0	9	10	13	0	-350	13
315	400	40	0	13	0	12	12	15	0	-400	15
400	500	45	0	15	0	14	12	18	0	-450	25

Table 7.10

Outer ring							
D		ΔD_{mp}		V_{Dp}	K_{ea}	S_D	Δ_{Cs}, V_{Cs}
over	to	max	min	max	max	max	
mm		µm					
50	80	0	-9	5	5	8	It corresponds to the inner ring of the same bearing
80	120	0	-10	5	6	9	
120	150	0	-11	6	7	10	
150	180	0	-13	7	8	10	
180	250	0	-15	8	10	11	
250	315	0	-18	9	11	13	
315	400	0	-20	10	13	13	
400	500	0	-23	12	15	15	
500	630	0	-28	14	17	18	
630	800	0	-35	18	20	20	

[1)] Corresponds with Δ_{Bs} and V_{Bs} of inner ring of the same bearing

Table 7.11a

Accuracy of dimensions and run of roller bearings with tapered hole Accuracy level UP											
Inner ring											
d		Δ_{dmp}		Δ_{d1mp}	$-\Delta_{dmp}$	V_{dp}	K_{ia}	S_d	Δ_{Bs}		V_{Bs}
over	to	max	min	max	min	max	max	max	max	min	max
mm		µm									
18	30	6	0	2	0	3	1,5	3	0	-25	1,5
30	50	7	0	3	0	3	2	3	0	-30	2
50	80	8	0	3	0	4	2	4	0	-40	3
80	120	10	0	4	0	4	3	4	0	-50	3
120	180	12	0	5	0	5	3	5	0	-60	4
180	250	14	0	6	0	6	4	6	0	-75	5
250	315	17	0	8	0	8	5	6	0	-90	6

Table 7.11b

Outer ring							
D		Δ_{Dmp}		V_{Dp}	K_{ea}	S_D	Δ_{Cs}, V_{Cs}
over	to	max	min	max	max	max	
mm		µm					
50	80	0	-6	3	3	2	It corresponds to the inner ring of the same bearing
80	120	0	-7	4	3	3	
120	150	0	-8	4	4	3	
150	180	0	-9	5	4	3	
180	250	0	-10	5	5	4	
250	315	0	-12	6	6	4	
315	400	0	-14	7	7	5	

[1] Corresponds with Δ_{Bs} and V_{Bs} of inner ring of the same bearing

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Table 7.12a

Accuracy of dimensions and run of tapered bearings															
Accuracy level P0															
Inner ring and total bearing width															
d		Δ_{dmp}		V_{dp}	V_{dmp}	K_{ia}	Δ_{Bs}		Δ_{Ts}		Δ_{T1s}		Δ_{T2s}		
over	to	max	min	max	max	max	max	min	max	min	max	min	max	min	
mm		µm													
10	18	0	-12	12	9	15	0	-120	200	0	100	0	100	0	
18	30	0	-12	12	9	18	0	-120	200	0	100	0	100	0	
30	50	0	-12	12	9	20	0	-120	200	0	100	0	100	0	
50	80	0	-15	15	11	25	0	-150	200	0	100	0	100	0	
80	120	0	-20	20	15	30	0	-200	200	-200	100	-100	100	-100	
120	180	0	-25	25	19	35	0	-250	350	-250	150	-150	200	-100	
180	250	0	-30	30	23	50	0	-300	350	-250	150	-150	200	-100	

Table 7.12b

Outer ring								
D		Δ_{Dmp}		V_{Dp}	V_{Dmp}	K_{ea}	Δ_{Cs}	
over	to	max	min	max	max	max	max	min
mm		µm						
18	30	0	-12	12	9	18	0	-120
30	50	0	-14	14	11	20	0	-120
50	80	0	-16	16	12	25	0	-150
80	120	0	-18	18	14	35	0	-200
120	150	0	-20	20	15	40	0	-250
150	180	0	-25	25	19	45	0	-250
180	250	0	-30	30	23	50	0	-300
250	315	0	-35	35	26	60	0	-350
315	400	0	-40	40	30	70	0	-400

Table 7.13a

Accuracy of dimensions and run of tapered bearings Accuracy level P6X															
Inner ring and total bearing width															
d		Δ_{dmp}		V_{dp}	V_{dmp}	K_{ia}	Δ_{Bs}		Δ_{Ts}		Δ_{T1s}		Δ_{T2s}		
over	to	max	min	max	max	max	max	min	max	min	max	min	max	min	
mm		μm													
10	18	0	-12	12	9	15	0	-50	100	0	50	0	50	0	
18	30	0	-12	12	9	18	0	-50	100	0	50	0	50	0	
30	50	0	-12	12	9	20	0	-50	100	0	50	0	50	0	
50	80	0	-15	15	11	25	0	-50	100	0	50	0	50	0	
80	120	0	-20	20	15	30	0	-50	100	0	50	0	50	0	
120	180	0	-25	25	19	35	0	-50	150	0	50	0	100	0	

Table 7.13b

Outer ring								
D		Δ_{Dmp}		V_{Dp}	V_{Dmp}	K_{ea}	Δ_{Cs}	
over	to	max	min	max	max	max	max	min
mm		μm						
18	30	0	-12	12	9	18	0	-100
30	50	0	-14	14	11	20	0	-100
50	80	0	-16	16	12	25	0	-100
80	120	0	-18	18	14	35	0	-100
120	150	0	-20	20	15	40	0	-100
150	180	0	-25	25	19	45	0	-100
180	250	0	-30	30	23	50	0	-100
250	315	0	-35	35	26	60	0	-100

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Table 7.14a

Accuracy of dimensions and run of tapered bearings Accuracy level P6								
Inner ring and total bearing width								
d		Δ_{dmp}		K_{ia}	Δ_{Bs}		Δ_{Ts}	
over	to	max	min	max	max	min	max	min
mm		µm						
10	18	0	-7	7	0	-200	200	0
18	30	0	-8	8	0	-200	200	0
30	50	0	-10	10	0	-240	200	0
50	80	0	-12	10	0	-300	200	0
80	120	0	-15	13	0	-400	200	-200
120	180	0	-18	18	0	-500	350	-250

Table 7.14b

Outer ring					
D		Δ_{Dmp}		K_{ea}	Δ_{Cs}
over	to	max	min	max	
mm		µm			
18	30	0	-8	9	It corresponds to the inner ring of the same bearing
30	50	0	-9	10	
50	80	0	-11	13	
80	120	0	-13	18	
120	150	0	-15	20	
150	180	0	-18	23	
180	250	0	-20	25	
250	315	0	-25	30	

[1)] Corresponds with Δ_{Bs} of inner ring of the same bearing

Table 7.15a

Accuracy of dimensions and run of tapered bearings											
Accuracy level P5											
Inner ring and total bearing width											
d		Δ_{dmp}		V_{dp}	V_{dmp}	K_{ia}	$Ś_d$	Δ_{Bs}		Δ_{Ts}	
over	to	max	min	max	max	max	max	max	min	max	min
mm		µm									
10	18	0	-7	5	5	5	7	0	-200	200	-200
18	30	0	-8	6	5	5	8	0	-200	200	-200
30	50	0	-10	8	5	5	8	0	-240	200	-200
50	80	0	-12	9	6	7	8	0	-300	200	-200
80	120	0	-15	11	8	8	9	0	-400	200	-200
120	180	0	-18	14	9	11	10	0	-500	350	-250

Table 7.15b

Outer ring								
D		Δ_{Dmp}		V_{Dp}	V_D	K_{ea}	$Ś_D$	Δ_{Cs}
over	to	max	min	max	max	max	max	
mm		µm						
18	30	0	-8	6	5	6	8	It corresponds to the inner ring of the same bearing
30	50	0	-9	7	5	7	8	
50	80	0	-11	8	6	8	8	
80	120	0	-13	10	7	10	9	
120	150	0	-15	11	8	11	10	
150	180	0	-18	14	9	13	10	
180	250	0	-20	15	10	15	11	
250	315	0	-25	19	13	18	13	

[1] Corresponds with Δ_{Bs} of the inner ring of the same bearing

Table 7.16a

Accuracy of dimensions and run of axial bearings							
Accuracy level P0, P6 and P5							
Shaft ring							
d		Δ_{dmp}		V_{dp}	S_i		[1]
d_2		Δ_{d2mp}		V_{d2p}	P0	P6	P5
over	to	max	min	max	max	max	max
mm		µm					
-	18	0	-8	6	10	5	3
18	30	0	-10	8	10	5	3
30	50	0	-12	9	10	6	3
50	80	0	-15	11	10	7	4
80	120	0	-20	15	15	8	4
120	180	0	-25	19	15	9	5
180	250	0	-30	23	20	10	5
250	315	0	-35	26	25	13	7
315	400	0	-40	30	30	15	7
400	500	0	-45	34	30	18	9
500	630	0	-50	38	35	21	11
630	800	0	-75	-	40	25	13

[1] Does not apply to thrust spherical roller bearings

Table 7.16b

Housing ring					
D		Δ_{Dmp}		V_{Dp}	S_e [1]
over	to	max	min	max	
mm		µm			
18	30	0	-13	10	Corresponds with S_i of shaft ring of the same bearing
30	50	0	-16	12	
50	80	0	-19	14	
80	120	0	-22	17	
120	180	0	-25	19	
180	250	0	-30	23	
250	315	0	-35	26	
315	400	0	-40	30	
400	500	0	-45	34	
500	630	0	-50	38	
630	800	0	-75	55	
800	1000	0	-100	75	
1000	1250	0	-125	-	
1250	1600	0	-160	-	

[1] Does not apply to thrust spherical roller bearings

7.2.3 Inner clearances of bearings

Clearance in bearing is the value of length of displacement of one assembled bearing ring towards the second ring from one marginal position to another (see fig. 7.3). The displacement can be in radial direction (radial clearance), or in axial direction (axial clearance).

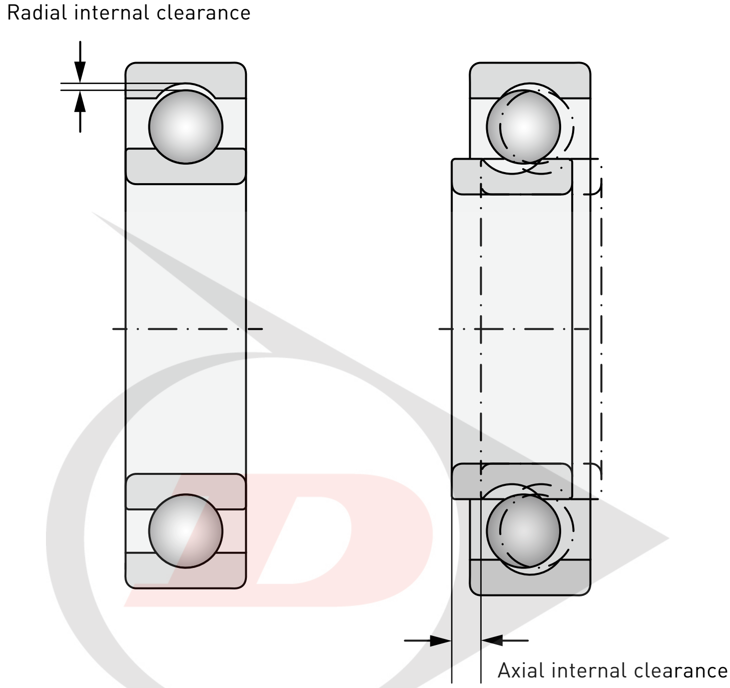

Fig. 7.3

In an in-built bearing we usually detect lower radial clearance than has the same bearing in unassembled state. Reduction of radial clearance is caused by the overlap sizes of bearing rings on the journal and in the body bore, and is therefore dependant on the selected tolerances of location surface diameters for the bearing. Further change of radial clearance, particularly its reduction, takes place during the operation due to temperature induced by the bearing operation itself, and by external sources, and also due to flexible deformations caused by load. Decisive is for bearing in stabilised service effects. Small prestress between the balls and raceways usually does not have negative effect.

Cylindrical roller, tapered roller, sphecical roller bearings feature higher rigidity, and therefore they are supposed to have smaller service clearance that is necessary to ensure safe and reliable run, mainly in heavy service conditions. If extremely high rigidity of location is required, e.g. for machine tools, prestressed bearings are mounted.

For normal design bearings the clearance is adjusted so that one of the bearing rings could be located firmly which is sufficient for majority of service ratios in location. Special cases of location with other requi-rements for radial clearance require bearings with radial clearance designated C1 to C5.

Values of different inner clearance levels according to ISO 5753 standard are for individual design bearing groups stated in charts 7.17 to 7.23 whilst these values apply to non-mounted bearings in zero load during measuring.

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Table 7.17a

Radial clearance of single row ball bearings															
Bore diameter d		Radial clearance												Single row ball bearings separable of E and BO type	Radial clearance	
		C2		Normal		C3		C4		C5				
over	to	min	max	min	max	min	max	min	max	min	max		min	max
mm		µm											µm	
2,5	10	0	7	2	13	8	23	14	29	20	37	E10, E12	15	30
10	18	0	9	3	18	11	25	18	33	25	45	E15	15	30
18	24	0	10	5	20	13	28	20	36	28	48	BO17, E17	25	45
24	30	1	11	5	20	13	28	23	41	30	53	E20	20	40
30	40	1	11	6	20	15	33	28	46	40	64			
40	50	1	11	6	23	18	36	30	51	45	73			
50	65	1	15	8	28	23	43	38	61	55	90			
65	80	1	15	10	30	25	51	46	71	65	105			
80	100	1	18	12	36	30	58	53	84	75	120			
100	120	2	20	15	41	36	66	61	97	90	140			
120	140	2	23	18	48	41	81	71	114	105	160			
140	160	2	23	18	53	46	91	81	130	120	180			
160	180	2	25	20	61	53	102	91	147	135	200			
180	200	2	30	25	71	63	117	107	163	150	215			
200	225	2	35	25	85	75	140	125	195	175	265			
225	250	2	40	30	95	85	160	145	225	205	300			
250	280	2	45	35	105	90	170	155	245	225	340			
280	315	2	55	40	115	100	190	175	270	245	370			
315	355	3	60	45	125	110	210	195	300	275	410			
355	400	3	70	55	145	130	240	225	340	315	460			
400	450	3	80	60	170	150	270	250	380	350	520			
450	500	3	90	70	190	170	300	280	420	390	570			
500	560	10	100	80	210	190	330	310	470	440	630			
560	630	10	110	90	230	210	360	340	520	490	700			
630	710	20	130	110	260	240	400	380	570	540	780			
710	800	20	140	120	290	270	450	430	630	600	860			
800	900	20	160	140	320	300	500	480	700	670	960			
900	1000	20	170	150	350	330	550	530	770	740	1040			
1000	1120	20	180	160	380	360	600	580	850	820	1150			

Table 7.17b

Axial clearance of double row angular-contact ball bearings									
Bore diameter d		Axial clearance							
		C2		Normal		C3		C4	
over	to	min	max	min	max	min	max	min	max
mm		µm							
6	10	1	11	5	21	12	28	25	45
10	18	1	12	6	23	13	31	27	47
18	24	2	14	7	25	16	34	28	48
24	30	2	15	8	27	18	37	30	50
30	40	2	16	9	29	21	40	33	54
40	50	2	19	11	33	23	44	36	58
50	65	3	22	13	36	26	48	40	63
65	80	3	24	15	40	30	54	46	71
80	100	3	26	18	46	35	63	-	-
100	110	4	30	22	53	42	73	-	-

Table 7.18

Radial clearance of double row self aligning ball bearings																					
Bore diameter d		Cylindrical bore										Tapered bore									
		C2		Normal		C3		C4		C5		C2		Normal		C3		C4		C5	
over	to	min	max	min	max	min	max	min	max	min	max	min	max	min	max	min	max	min	max	min	max
mm		µm										µm									
2,5	6	1	8	5	15	10	20	15	25	21	33	-	-	-	-	-	-	-	-	-	-
6	10	2	9	6	17	12	25	19	33	27	42	-	-	-	-	-	-	-	-	-	-
10	14	2	10	6	19	13	26	21	35	30	48	-	-	-	-	-	-	-	-	-	-
14	18	3	12	8	21	15	28	23	37	32	50	-	-	-	-	-	-	-	-	-	-
18	24	4	14	10	23	18	30	25	39	34	52	7	17	13	26	20	33	28	42	37	55
24	30	5	16	11	24	19	35	29	46	40	58	9	20	15	28	23	39	33	50	44	62
30	40	6	18	13	29	23	40	34	53	46	66	12	24	19	35	29	46	40	59	52	72
40	50	6	19	14	31	25	44	37	57	50	71	14	27	22	39	33	52	45	65	58	79
50	65	7	21	16	36	30	50	45	69	62	88	18	32	27	47	41	61	56	80	73	99
65	80	8	24	18	40	35	60	54	83	76	108	23	39	35	57	50	75	69	98	91	123
80	100	9	27	22	48	42	70	64	96	89	124	29	47	42	68	62	90	84	116	109	144
100	120	10	31	25	56	50	83	75	114	105	145	35	56	50	81	75	108	100	139	130	170
120	140	10	38	30	68	60	100	90	135	125	175	-	-	-	-	-	-	-	-	-	-
140	160	15	44	35	80	70	120	110	161	150	210	-	-	-	-	-	-	-	-	-	-

Table 7.19

Radial clearance of single row cylindrical roller bearings											
Bore diameter d		Radial clearance									
		C2		normal		C3		C4		C5	
over	to	min	max	min	max	min	max	min	max	min	max
mm		µm									
10	24	0	25	20	45	35	60	50	75	65	90
24	30	0	25	20	45	35	60	50	75	70	95
30	40	5	30	25	50	45	70	60	85	80	105
40	50	5	35	30	60	50	80	70	100	95	125
50	65	10	40	40	70	60	90	80	110	110	140
65	80	10	45	40	75	65	100	90	125	130	165
80	100	15	50	50	85	75	110	105	140	155	190
100	120	15	55	50	90	85	125	125	165	180	220
120	140	15	60	60	105	100	145	145	190	200	245
140	160	20	70	70	120	115	165	165	215	225	275
160	180	25	75	75	125	120	170	170	220	250	300
180	200	35	90	90	145	140	195	195	250	275	330
200	225	45	105	105	165	160	220	220	280	305	365
225	250	45	110	110	175	170	235	235	300	330	395
250	280	55	125	125	195	190	260	260	330	370	440
280	315	55	130	130	205	200	275	275	350	410	485
315	355	65	145	145	225	225	305	305	385	455	535
355	400	100	190	190	280	280	370	370	460	510	600
400	450	110	210	210	310	310	410	410	510	565	665
450	500	110	220	220	330	330	440	440	550	625	735
500	560	120	240	240	360	360	480	480	600	695	815
560	630	140	260	260	380	380	500	500	620	780	900
630	710	145	285	285	425	425	565	565	705	870	1010
710	800	150	310	310	470	470	630	630	790	980	1140
800	900	180	350	350	520	520	690	690	860	1100	1270
900	1000	200	390	390	580	580	770	770	960	1220	1410
1000	1120	220	430	430	640	640	850	850	1060	1360	1570
1120	1250	230	470	470	710	710	950	950	1190	1520	1760

Table 7.20

Radial clearance of double row cylindrical roller bearings with tapered bore					
Bearings with incommutable rings designed for work spindles of machine tools					
Bore diameter		Radial clearance			
d		C1NA		C2NA	
over	to	min	max	min	max
mm		µm			
24	30	15	25	25	35
30	40	15	25	25	40
40	50	17	30	30	45
50	65	20	35	35	50
65	80	25	40	40	60
80	100	35	55	45	70
100	120	40	60	50	80
120	140	45	70	60	90
140	160	50	75	65	100
160	180	55	85	75	110
180	200	60	90	80	120
200	225	60	95	90	135
225	250	65	100	100	150
250	280	75	110	110	165
280	315	80	120	120	180
315	355	90	135	135	200
355	400	100	150	150	225
400	450	110	170	170	255
450	500	120	190	190	285
500	560	130	210	210	315
560	630	140	230	230	345
630	710	160	260	260	390
710	800	180	290	290	435
800	900	200	320	320	480
900	1000	-	-	355	540

Table 7.21

Radial clearance of single row cageless needle roller bearings with interchangeable rings					
Bore diameter		Radial clearance			
d		normal		C2NA	
over	to	min	max	min	max
mm		µm			
10	14	10	50	25	70
14	18	15	55	35	75
18	24	25	65	40	80
24	30	30	65	50	80
30	40	40	75	60	95
40	50	40	85	65	100
50	65	45	90	70	120
65	80	50	110	75	135
80	100	60	115	95	150
100	120	70	125	115	70
120	140	80	155	130	205
140	160	80	160	140	210

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Table 7.22

Radial clearance of double row spherical-roller bearings											
Bore diameter d		Cylindrical bore									
		C2		normal		C3		C4		C5	
over	to	min	max	min	max	min	max	min	max	min	max
mm		µm									
30	40	15	30	30	45	45	60	60	80	80	100
40	50	20	35	35	55	55	75	75	100	100	125
50	65	20	40	40	65	65	90	90	120	120	150
65	80	30	50	50	80	80	110	110	145	145	180
80	100	35	60	60	100	100	135	135	180	180	225
100	120	40	75	75	120	120	160	160	210	210	260
120	140	50	95	95	145	145	190	190	240	240	300
140	160	60	110	110	170	170	220	220	280	280	350
160	180	65	120	120	180	180	240	240	310	310	390
180	200	70	130	130	200	200	260	260	340	340	430
200	225	80	140	140	220	220	290	290	380	380	470
225	250	90	150	150	240	240	320	320	420	420	520
250	280	100	170	170	260	260	350	350	460	460	570
280	315	110	190	190	280	280	370	370	500	500	630
315	355	120	200	200	310	310	410	410	550	550	690
355	400	130	220	220	340	340	450	450	600	600	760
400	450	140	240	240	370	370	500	500	660	660	820
450	500	140	260	260	410	410	550	550	720	720	900
500	560	150	280	280	440	440	600	600	780	780	1000
560	630	170	310	310	480	480	650	650	850	850	1100
630	710	190	350	350	530	530	700	700	920	920	1190
710	800	210	390	390	580	580	770	770	1010	1010	1300
800	900	230	430	430	650	650	860	860	1120	1120	1440
900	1000	260	480	480	710	710	930	930	1220	1220	1570
1000	1120	290	530	530	780	780	1020	1020	1330	1330	1720

Table 7.23

Radial clearance of double row spherical-roller bearings											
Bore diameter d		Tapered bore									
		C2		normal		C3		C4		C5	
over	to	min	max	min	max	min	max	min	max	min	max
mm		µm									
30	40	25	35	35	50	50	65	65	85	85	105
40	50	30	45	45	60	60	80	80	100	100	130
50	65	40	55	55	75	75	95	95	120	120	160
65	80	50	70	70	95	95	120	120	150	150	200
80	100	55	80	80	110	110	140	140	180	180	230
100	120	65	100	100	135	135	170	170	220	220	280
120	140	80	120	120	160	160	200	200	260	260	330
140	160	90	130	130	180	180	230	230	300	300	380
160	180	100	140	140	200	200	260	260	340	340	430
180	200	110	160	160	220	220	290	290	370	370	470
200	225	120	180	180	250	250	320	320	410	410	520
225	250	140	200	200	270	270	350	350	450	450	570
250	280	150	220	220	300	300	390	390	490	490	620
280	315	170	240	240	330	330	430	430	540	540	680
315	355	190	270	270	360	360	470	470	590	590	740
355	400	210	300	300	400	400	520	520	650	650	820
400	450	230	330	330	440	440	570	570	720	720	910
450	500	260	370	370	490	490	630	630	790	790	1000
500	560	290	410	410	540	540	680	680	870	870	1100
560	630	320	460	460	600	600	760	760	980	980	1230
630	710	350	510	510	670	670	850	850	1090	1090	1360
710	800	390	570	570	750	750	960	960	1220	1220	1500
800	900	440	640	640	840	840	1070	1070	1370	1370	1690
900	1000	490	710	710	930	930	1190	1190	1520	1520	1860
1000	1120	530	770	770	1030	1030	1300	1300	1670	1670	2050

For double row ball bearings with angular contact, axial clearance measured at axial load of 100 N is stated instead of radial clearance.

If different clearance is selected than normal, one needs to process carefully and consider the effect if operating conditions at stabilised state. Radial clearance smaller than normal is selected quite rarely, e.g. in roller bearings for machine tool spindles. More often bearings with radial clearance bigger than normal are needed. This happens mostly in case the limit revolutions are exceeded, or in case of higher temperature gra-dient between the inner an outer ring and, finally, to increase axial load capacity of single row ball bearings. Axial load capacity of these bearings is increased at the clearance of C3 by approx. 10 %, and at clearance C4 by approx. 20 % in normal conditions.

It is understandable that not only too small but also too big radial clearance has negative effect on the operation and life service of roller bearing. As we know from experience, roller bearing is more negatively affected by small radial clearance than by big. If the thermal service conditions in the bearing are unclear, it is safer to select quite bigger radial clearance that might in an extreme case reduce the service life of the bearing which is insignificant.

Single row ball bearings with angular contact and single row tapered roller bearings are usually mounted in pairs in which radial or axial clearance or prestress are adjusted during the assembly. With advantage the property of the so-called combined bearings can be utilised in which the final axial clearance is set by the bearing manufacturer.

Dependence of radial and axial clearance in some bearing types is clear from chart 7.24.

Table 7.24

Dependence of radial clearance V_r an axial clearance V_a	
Type of bearing	**V_a/V_r**
Single Row Ball Bearings	-
Double Row Angular Contact Ball Bearings, type 32, 33	1,4
Self-Aligning Ball Bearings	1,5/e
Tapered Roller Bearings	
Spherical Roller Bearings	

Figure 7.4 shows an informative graph of dependence of radial an axial clearance in bearing, applicable to single row ball bearings.

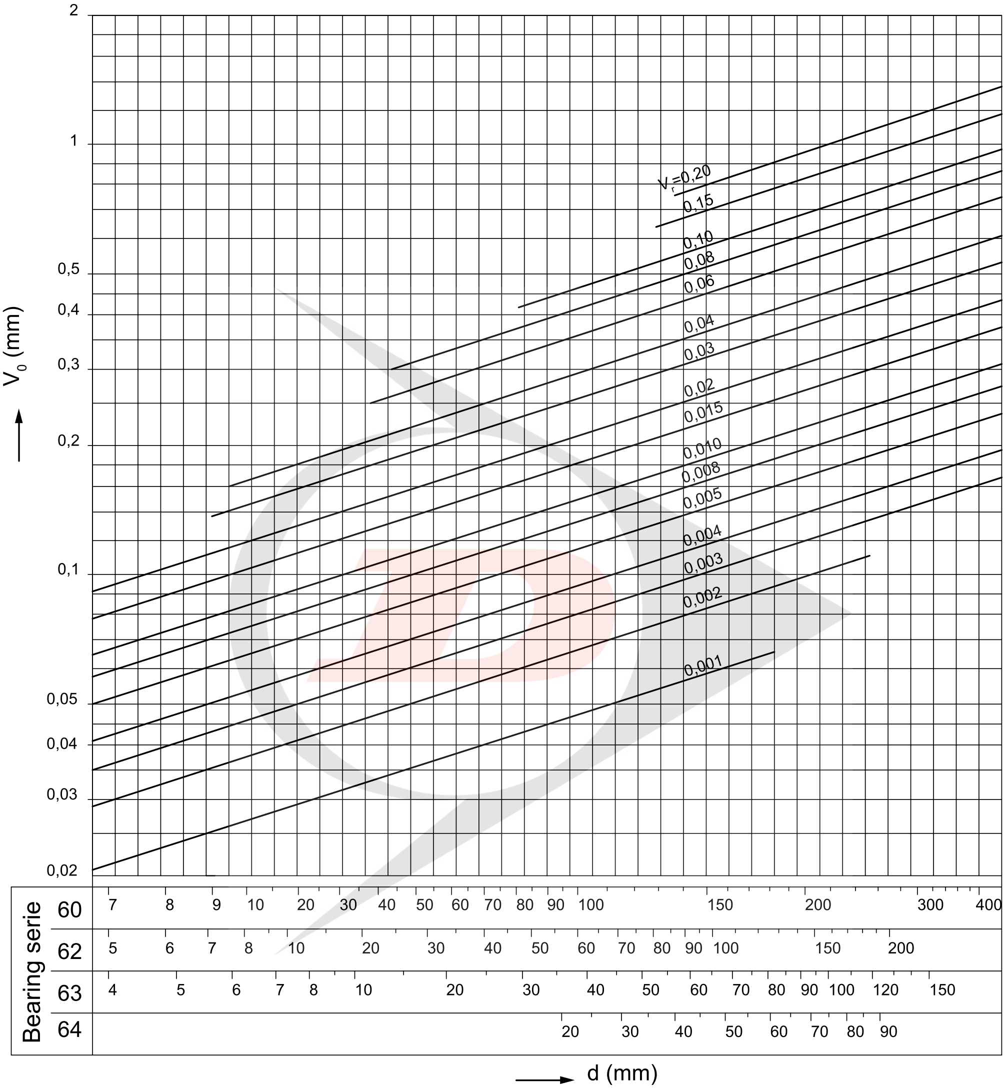

Fig. 7.4

7.3 Roller bearings materials

7.3.1 Materials of bearing rings and rolling bodies

In terms of materials used for production of roller bearings, durability and reliability of roller bearings is specifically increased by using more accurate metallurgical technologies based on recent surveys. Previous studies already demonstrated a direct connection between micropurity of the bearing steel used, and the occurrence of subsurface fatigue damage in the rolling contact. With regard to high pressures in the area of the rolling contact, strict requirements for micropurity and uniformity of distribution of carbidic phases are reasonable. The requirement of continuous durability increase can be satisfied by highly accurate and quality production combined with using materials with low content of oxygen and non-metal intrusions, and technologically correct thermal processing of rings and bearing rolling bodies when specified hardness, microstructure and dimensional stability is achieved. This provides resistance to wear and necessary load capacity of rolling contact. Chemical composition and maximum contents of undesired elements are defined in the international standard for bearing steels ISO 683-17.

For locations with a risk of damage in the area of rolling contact due to passage of electric current, bea-rings with ceramic insulation coating of the outer ring can be supplied.

If there are special requirements for material, design or use of bearings, information is available at the Dunlop BTL´s technical an consultancy centre.

Semiproducts

Besides economic criteria, a semiproduct for production of roller bearings and rolling elements has to comply with technological requirements in terms of proper course of fibres and proper distribution of carbidic phases. For the economic reason and also due to convenient passage of fibres, the most convenient is using a tube semiproduct that is cold rolled to final shape prior to thermal processing. In this way, the majority of the bearing assortment with increased basic durability is produced with the identification "NEW FORCE".

Through-hardening steels

Majority of standard produced Dunlop BTL roller bearings are made of through-hardening steels designed for pro-duction of roller bearings. Those are carbon – chromium steels with an approximate content of 1 % carbon and 1.5 % chromium, complying with the international standard ISO 683-17 "Heat-treated steels, alloy steels and free-cutting steels, Part17: Steels for rolling bearings". After heat treatment, material has the same structure and hardness throughout the component section. After performed martensitic or bainite hardening and subsequent tempering, the hardness of final surfaces is 58 to 65 HRC.

Depending on the type, the highest service temperature of 120 °C to 200 °C is recommended for standard Dunlop BTL roller bearings. The maximum temperature for using the bearings depends on heat treatment of bearing components. For operation at temperatures to 250 °C, bearing components can stabilize in a special heat treatment process. In case of thermal stabilization for operation at higher temperatures, the hardness of com-ponents reduces significantly, and thus also the dynamic load capacity of the bearings. If long-term operation above 250 °C is required, we recommend bearings from high alloy steels designed for high temperatures.

Case hardening steels

After saturation with carbon and hardening, bearing components feature hard surface and simultaneously also tough core. They are used for production of bearings that are loadable with big strokes, locations with big overlap or alternatively for locations with a possibility of contaminated lubrication.

Corrosion-proof steels

These steels are used for bearings intended for operation in oxidizing environment, for instance for aviation technology or food processing industry.

Steels for high temperatures

These materials are used for bearings operating permanently at temperatures over 250 °C whilst maintaining hardness and standard service properties, e.g. in aircraft engines.

Steels for surface hardening

These steels offer convenient combination of hardened tough raceway with tough section core. They are used mainly in large bearings, or bearings with clamp flanges which are contained in bearing rings.

7.3.2 Materials for production of cages

Materials used for production of cages are selected with regard to the service temperature of the bearing, whether the bearing will operate in standard or vibrating environment, alternatively upon the requirements for chemical or corrosion resistance.

The basic quality of materials used for production of cages is good abrasion resistance and slip properties along with sufficient ductility.

Pressed steel cages

They are pressed from low carbon steels that ensure accuracy of final cage shape, as well as sufficient ductility. To improve slip properties and abrasion resistance, the surface of pressed cages is chemically and thermally treated. They suit typical temperature regimen of bearing operation up to 300°C.

In smaller bearings sizes, pressed cages are even made of brass sheet.

Massive brass cages

They are made in routing from roughened or spun semiproducts. Service temperature should not exceed 250°C.

Massive steel cages

In justified cases they are an alternative to brass massive cages. Service temperature may range up to 300 °C. The surface of the cage can be chemically and thermally treated.

7.3.3 Other materials

Polymers

Polymers, usually of polyamide 66 reinforced with glass fibres, are used mainly for production of cages and cage guide rings of double row spherical roller bearings of CJ design. Service operation of these components should not exceed 120 °C in the long term with the use of common lubricants, 150 °C in the short term (within 10 hours), and 170 °C in peaks (within 20 minutes). Usefulness of bearings with polyamide components at lower temperatures is, with regard to polyamide elasticity loss, up to the temperatures of -40 °C.

Ceramic materials

Are used mostly to prevent bearings from damage by passage of electric current, either in form of thermally layered coats on the surface of the outer or inner ring, alternatively by using rolling ceramic elements. Use of rolling elements from ceramic material is justified even in special high-revolution bearings.

Other

Materials of contact seals are selected so as their thermal and degradation resistance suited the selected use.

7.4 Cages

Cage has the below functions in a roller bearing: Distributes rolling bodies uniformly around the circumference and prevents their mutual contact which reduced friction in the bearing. It prevents slippage of rolling bodies in the bearing and falling rolling bodies out of separable bearings during their assembly.

In terms of design and materials, cages are divided in pressed (fig. 7.5) and massive (fig. 7.6).

Pressed cages are made mostly by pressing from steel or brass sheet, and usually are used in dimensionally smaller up to medium bearings. Comparing to massive cages, their advantage is lower weight.

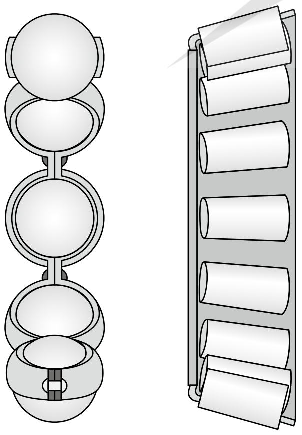

Fig. 7.5

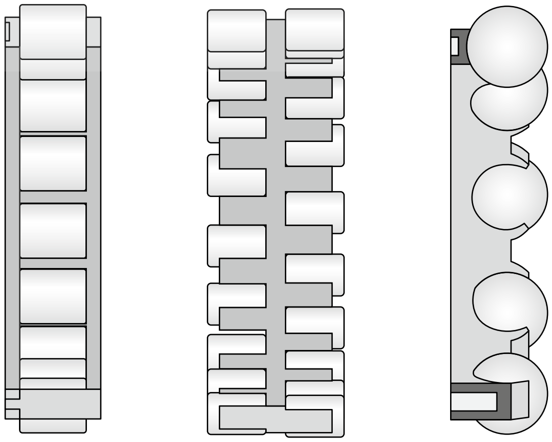

Fig. 7.6

Massive cages are made of steel, brass, bronze, light metals or plastics in various designs. Metal cage materials are used whenever increased requirements are imposed on the rigidity of the cage, and the bearing is designed for higher service temperatures. Cages in bearing run radially on rolling elements which is the most common way, or on flange of one of the bearing rings (fig. 7.7).

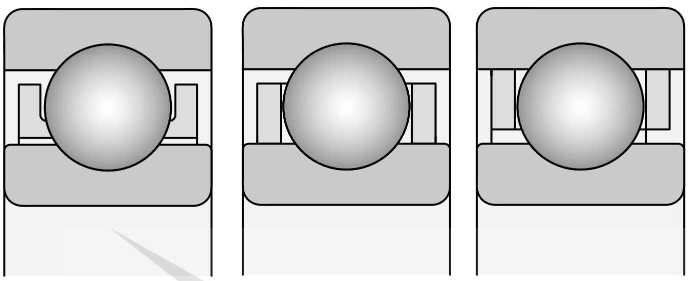

Fig. 7.7

Massive polymer cages are made by injection moulding. The injection moulding technology allows to produc-tion such cage shapes that enable designing bearings with high load capacity. Elasticity and low polyamide weight applies positively in shock stress of bearings, high acceleration and deceleration. Polyamide cages feature good slip properties. During lubrication of bearings with oil, the additives contained in the oil may affect negatively the service life of the cage.

Cages made of phenological resin are light but not suitable to high temperatures. They however feature good resistance to centrifugal forces. They are typically use in accurate ball bearings with angular contact.

Journal cages are made of steel; the condition is use of holy rolling bodies (fig. 7.8). Journal cages are used mainly in large bearings

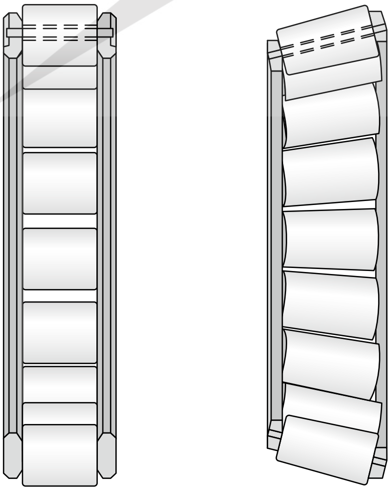

Fig. 7.8

Cageless bearings, i.e. fully complement, are used rarely – only in some types of bearings, e.g. single row cylindrical roller bearings.

In texts to individual design bearing groups the section dedicated to cages always states an overview of cages made in the general design, and delivery option of bearings with cages in different designs.

7.5 Shield and seals

Bearings with covers on one or both sides are made with shields (Z, 2Z, ZR, 2ZR – fig. 7.9), or with contact seal ((RS, 2RS, RSR, 2RSR – fig. 7.10). Shields create create contact-free sealing. In Z or 2Z version, the fitting for shield is on the inner ring; ZR or 2ZR variants have shield adhered to the smooth flange of the inner ring.

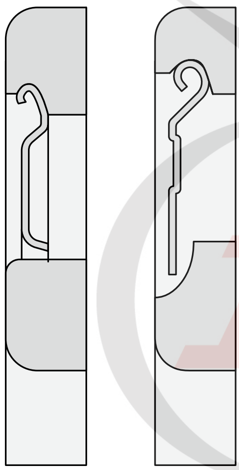

Fig. 7.9

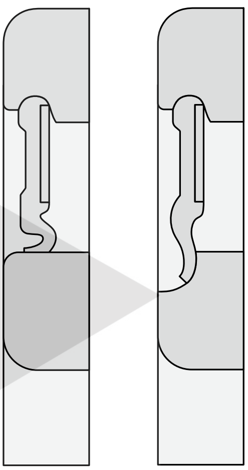

Fig. 7.10

The seal consists of sealing rings of nitrile rubber vulcanized on metal reinforcements that form an efficient contact seal in a design with rounded fitting on the inner ring (RS, 2RS), or in a design with contact on the smooth flange of the inner ring (RSR, 2RSR).

Shields and sealing rings are fastened in the outer ring recess, and are not detachable.

Bearings in basic design are filled with a quality plastic lubricant with temperature range between -30 °C and +100 °C, in the short term even up to +120 °C. Filler of grease usually ensures greasing throughout the service life in normal service conditions. Bearings in this design cannot be additionally greased.

7.6 Designation of roller bearings

Bearing is designated by basic designation and extension expressing the difference between this bearing and the standard version bearing. Designation of bearings contains numerical and literal characters that determine the type, size and design of the bearing. Overview of symbols and their order is based on the scheme shown in figure 7.11.

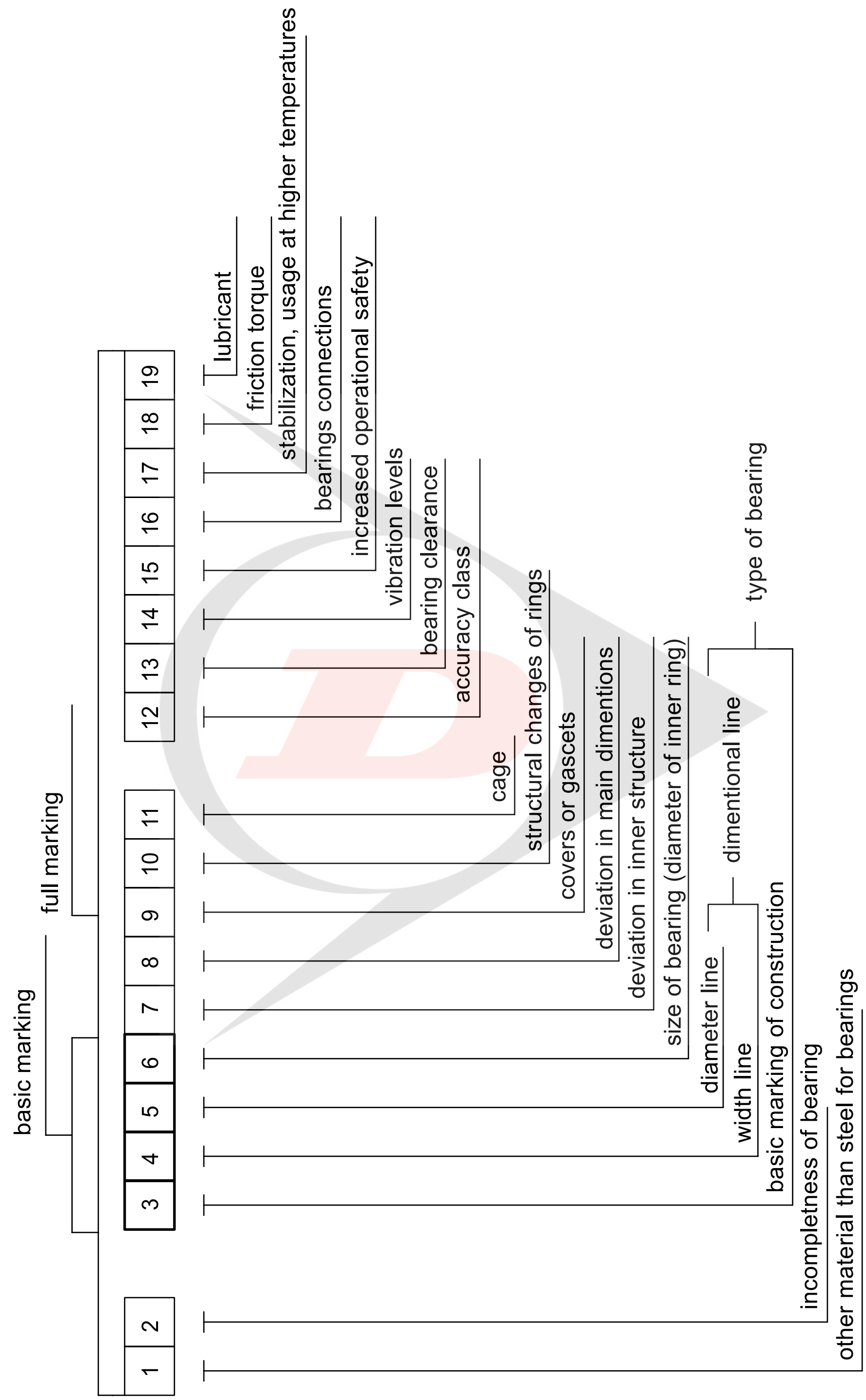
basic marking
full marking
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
19
lubricant
friction torque
stabilization, usage at higher temperatures
bearings connections
increased operational safety
vibration levels
bearing clearance
accuracy class
cage
structural changes of rings
covers or gascets
deviation in main dimentions
deviation in inner structure
size of bearing (diameter of inner ring)
diameter line
width line
dimentional line
type of bearing
basic marking of construction
incompletness of bearing
other material than steel for bearings

Fig. 7.11

7.6.1 Standard bearing version

In standard version, bearings are identified with basic designation consisting of the identification of the type and size of the bearing. The designation usually consists of a symbol expressing the design of the bearing (position 3 of the scheme), and a symbol for the dimensional group or diameter row (positions 4 and 5), e.g. type 223, 302, NJ22, 511, 62, 12 and so on. Designation of the bearing size contains characters for nominal bearing bore diameter d (position 6).

Bearings with bore diameter d < 10 mm:

Figures separate with fraction line or the last digit states directly the nominal bore dimension in mm, e.g. 619/2, 624.

Bearings with bore diameter d = 10 up to 17 mm:

double issue	00	identifies the bore	d = 10 mm, e.g.: 6200
	01		d = 12 mm, e.g.: 51101
	02		d = 15 mm, e.g.: 3202
	03		d = 17 mm, e.g.: 6303

Exception in designation are single row ball bearings of separable type E and BO where the double issue states directly the bore diameter in mm, e.g.: E17.

Bearings with bore diameter d = 20 mm up to 480 mm

Bore diameter is quintuple of the last double issue, e.g. bearing 1320 features bore diameter d = 20 x 5 = 100 mm.

Exceptions are bearings with bore diameter d = 22, 28 and 32 mm where the double issue separated with fraction line stated directly the diameter of bore in mm, e.g. 320/32AX, and some bearing types, such as e.g. separable single row ball bearings of E type, and single row ball bearings of NG type where the double or triple issue states directly the bore diameter in mm, e.g.: E20, NG160.

Bearings with bore diameter d > 500 mm:

The last double issue or triple digit separated with fraction line states directly the bore dimension in mm, e.g. 30/530M, NU29/1060.

7.6.2 Full designation of bearings

Bearing produced in designs different from the standard are identified by the so-called designation, as is shown in the scheme in fig. 7.11. It consists of the basic designation and supplementary characters that express the difference from the basic version.

Meaning of supplementary characters

The following part states, in accordance with full designation, an overview and meaning of supplementary characters used. The digit in the bracket stated with individual groups corresponds with the position number in the scheme. The scheme also states positions in full designation of the bearing that us separated with a gap.

Other characters are written together without a gap. Characters for extension of designation that mean a digit are separated with a dash from the basic designation, e.g. 6305-2Z.

The meaning of supplementary characters for design variances of different bearing types is described in relevant chapters of the chart section of the catalogue.

Supplementary characters before basic designation

Other material than common steel for roller bearings (1)

C . . . . rolling elements from ceramics – e.g. C B7006CTA HSS . . . . .

high speed steel, e.g.: HSS 6215

X . . . . corrosion resistant steel, e.g.: X 623

T . . . . . . .case hardening steel, e.g.: T 32240

Bearing incompleteness (2)

L . . . . separate detachable ring of separable bearing, e.g. L NU206, in thrust ball bearings without a shaft ring, e.g. L 51215

R . . . . . . separable bearing without detachable ring, e.g. R NU206 nebo R N310

E . . . . separate shaft ring or thrust ball bearing, e.g. E 51314

W . . . . . . separate body ring of thrust ball bearing, e.g. W 51414

K . . . . cage with rolling elements e.g.: K NU320

Supplementary characters behind the basic designation

Difference in inner design (7)

A . . . . single row angular-contact ball bearings with contact angle α = 25°, e.g. B7205ATB P5 . . . .

single row tapered bearings with higher load capacity and higher limit revolution frequency,

e.g. 30206A

. . . . thrust ball bearings with higher limit revolution frequency, e.g. 51,105A

AA . . . . . . single row angular-contact ball bearings with contact angle α = 26°, e.g. B7210AATB P5

B . . . . single row angular-contact ball bearings with contact angle α = 40°, e.g. 7304B

. . . . single row tapered bearings with contact angle α > 17°, e.g. 32315B

BE . . . . . . single row angular-contact ball bearings with contact angle α = 40°, in new design, e.g. 7310BETNG

C . . . . single row angular-contact ball bearings with contact angle α = 15°, e.g. 7220CTB

P4 . . . . double row spherical roller bearings in new design, e.g. 22216C

CA . . . . . . single row angular-contact ball bearings with contact angle α = 12°, e.g. B7202CATB P5

C B . . . single row angular-contact ball bearings with contact angle α = 10°, e.g. B7206CBTB P4

D. . . . single row ball bearing of type 160 with higher load capacity, e.g. 16004D

E . . . . single row cylindrical roller bearings with higher load capacity, e.g. NU209E

. . . . double row spherical roller bearings with higher load capacity, e.g. 22215E

. . . . Spherical roller thrust bearings with higher load capacity, e.g. 29416E

Difference in main dimensions (8)

X . . . . Change in main dimensions, established by new international standards, e.g. 32028AX

Covers (9)

RS . . . . . . seal on one side, e.g. 6304-RS

2RS . . . . . seal on both sides, e.g. 6204-2RS

RSN. . . . . seal on one side and snap ring groove on the outer ring on the opposite side than the

seal, e.g. 6306RSN

RSNB . . . seal on one side and snap ring groove on the outer ring on the same side as the seal, e.g. 6210RSNB

2RSN. . . . seal on both sides and snap ring groove on the outer ring, e.g. 6310-2RSN

RSR. . . . . seal on one side, adhering to the smooth inner ring collar, e.g. 624RSR

2RSR. . . . 2RSR – seals on both sides adhering to the smooth inner ring collar, e.g. 608-2RSR

Z . . . . shield on one side, e.g. 6206-Z

2Z . . . . . . shields on both sides, e.g. 6304-2Z

ZN . . . . . . shield on one side and snap ring groove on the outer ring on the opposite side than the shield, e.g. 6208ZN

ZNB. . . . . shield on one side and snap ring groove on the outer ring on the same side as the shield, e.g. 6306ZNB

2ZN . . . . . shields on both sides and snap ring groove on the outer ring, e.g. 6208-2ZN

ZR . . . . . . shield on one side, adhering to the smooth inner ring flange, e.g. 608ZR

2ZR . . . . . shields on both sides, adhering to the smooth inner ring flanges, e.g. 608-2ZR

Design change of bearing rings (10)

K . . . . Tapered bore, taper ratio 1:12, e.g. 1207K

K30 . . . . . Tapered bore, taper ratio 1:30, e.g. 24064K30M

N . . . . snap ring groove on the outer ring, e.g. 6308N

NR . . . . . . snap ring groove on the outer ring, and inserted snap ring, e.g. 6310NR

NX . . . . . . snap ring groove on the outer ring, dimensions of which do not comply with ČSN 02 4605, e.g. 6210NX

D . . . . split inner ring, e.g. 3309D

W33 . . . . groove and lubrication bores on the outer ring circumference, e.g. 23148W33M

O . . . . lubrication slots on outer ring fillet of the bearing , e.g. NU1014O

Cage (11)

Material of cages for standard design bearings is usually not specified.

J. . . . . cage pressed from steel plate, guided on rolling elements e.g.: 6034J

J2 . . . . . . cage pressed from steel plate, guided on rolling elements. New design of single row tapered bearings, e.g. 30206AJ2

Y . . . . cage pressed from brass sheet, guided on rolling elements e.g.: 6001Y

F . . . . massive steel cage, guided on rolling elements e.g.: 6418F

L . . . . massive light metal cage, guided on rolling elements e.g.: NG180L C3S0

M . . . . massive brass or bronze cage, guided on rolling elements e.g.: NU330M

T . . . . massive textite cage, guided on rolling elements e.g.: 6005T

TN . . . . . . massive cage of polyamide or similar plastic, guided on rolling elements e.g.: 6207TN

TNG . . . . . massive cage of polyamide or similar plastic, reinforced by glass fibres, guided on rolling elements e.g.: 2305TNG

Cage design (stated characters are always used in combination with cage material characters).

A . . . . cage guided on outer ring, e.g. NU226MA

B . . . . . . cage guided on inner ring, e.g. B7204CATB P5

P . . . . massive window cage, e.g.: NU1060MAP

H . . . . . . open single-piece cage, e.g.: 629TNH

S . . . . cage with lubrication slots, e.g.: NJ418MAS

R . . . . . . silver-plated cage, e.g.: 6210MAR

V . . . . bearing without cage with full number of rolling elements, e.g. NU209V

Accuracy level (12)

P0 . . . . . . normal accuracy level (is not designated), e.g. 6204

P6 . . . . . . higher accuracy level than normal, e.g. 6322 P6

P5 . . . . . . higher accuracy level than P6, e.g. 6201 P5

P5A . . . . . higher accuracy level than P5 in some parameters, e.g. 6006TB P5A

P4 . . . . . . higher accuracy level than P5, e.g. B7204CBTB P4

P4A . . . . . higher accuracy level than P4 in some parameters, e.g. B7205CATB P4A

P2 . . . . . . higher accuracy level than P4, e.g. B7200CBTB P2

P6E . . . . . higher accuracy level for rotary electrical machines, e.g. 6204 P6E

P6X . . . . . higher accuracy level for single row tapered bearings, e.g. 30210A P6X

SP . . . . . . higher accuracy level for roller bearings with tapered bore, e.g. NN3022K SPC2NA

UP . . . . . . higher accuracy level such as SP for roller bearings with tapered bore, e.g. N1016K

UPC1NA

Clearance (13)

C2 . . . . . . smaller clearance than normal, e.g. 608 C2

C2 . . . . . normal clearance (is not designated), e.g. 6204 C2

C3 . . . . . . bigger clearance than normal, e.g. 6310 C3

C4 . . . . . . bigger clearance than C3, e.g. NU320M C4

C5 . . . . . . bigger clearance than C4, e.g. 22330M C5

NA . . . . . . radial clearance in bearings with incommutable rings (is indicated always behind the radial clearance group), e.g. NU215 P63NA

R... . . . . . radial clearance in non-standardised range (range in µm) , e.g. 6210 R10-20

A... . . . . . axial clearance in non-standardised range (range in µm) , e.g. 3210 A20-30 **Noise level (14)**

C6 . . . . . . reduced noise level lower than normal (is not designated), e.g. 6304 C6

C06 . . . . . reduced noise level lower than C6, e.g. 6205 C06

C66 . . . . . reduced noise level lower than C06, e.g. 6205 C66

Specific values for C06 and C66 are determined based on an agreement between customer and supplier. Note: Bearings in accuracy level P5 and higher feature noise level within C6.

Increased operational safety (15)

C7, C8, C9 . . . . bearings with increased operational safety designed mainly for use in aviation industry, e.g. 6008MB P68

Combining characters (12-15)

Characters/symbols of accuracy level, clearance in bearing, noise levels and increased operational safety are combined with simultaneous omission of C character and following special property of bearings, e.g.

P6 + C3 = P63 . . . . e.g. 6211 P63

P6 + C8 = P68 . . . . e.g. 16002 P68

C3 + C6 = C36 . . . . e.g. 6303-2RS C36

P5 + C3 + C9 = P539 . . . . e.g. 6205MA P539

P6 + C2NA + C6 = P626NA . . . . . . e.g. P626NA

P626NA **Bearing association (16)**

Designation of associated pair, triplet or quaternion of bearings consists of characters expressing arrange-ment of bearings and of characters defining the inner clearance or prestress of associated bearings.

Apart from characters stated in the chart the U character is used to identify that relevant bearings can be associate universally, example of designation B7003CTA P4UL.

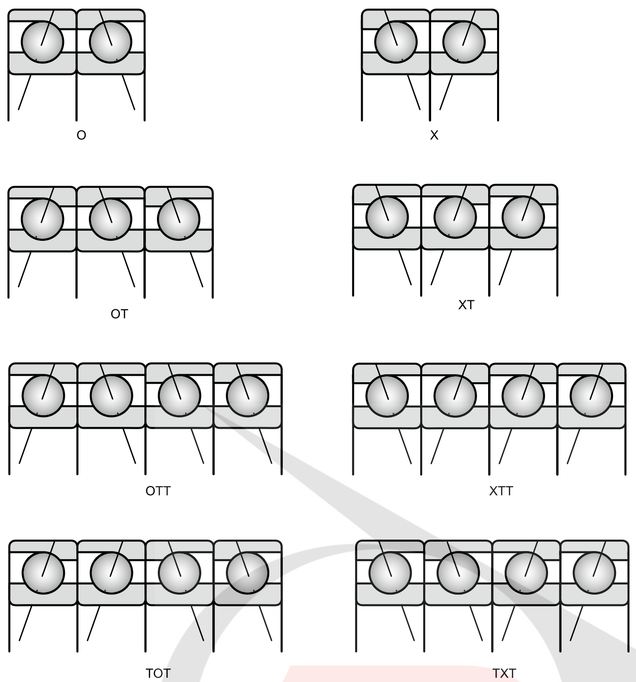

Fig. 7.12

Inner clearance or prestress

Stated characters are always used in combination with association characters.

A . . . . Association of bearings with clearances, e.g. 7305OA

O . . . . Association of bearings without clearances, e.g. 7305 P6XO

L . . . . Association of bearings with small prestress, e.g. B7205CATB P4UL

M . . . . Association of bearings with medium prestress, e.g. B7204CATB P5XM

S . . . . Association of bearings with big prestress, e.g. B7304AATB P4OS **Stabilisation for operation at higher temperature (17)**

Both rings have stabilised dimensions for operation at higher temperature.

S0 – for service temperature . . . . up to 150 °C

S1 . . . . up to 200 °C

S2 . . . . up to 250 °C

S3 . . . . up to 300 °C

S4 . . . . up to 350 °C

S5 . . . . up to 400 °C

Example of designation NG160LB C4S3

Friction torque (18)

JU . . . . . . reduced friction torque, e.g. 619/2 JU

JUA . . . . . bearings with defined friction torque at start-up 632 JUA

JUB . . . . . bearings with defined friction torque at after-running, e.g. 623 JUB **Grease (19)**

For bearings with shield or seal on both sides, the plastic lubrication other than common is designated by means additional characters. The first two characters define the range of service temperature, and the third character (letter) defines the name or type of lubricant according to the manufacturer´s specification, or another character (digit) defines the amount of grease that fills the covered space of the bearing.

TL . . . . . . grease for low service temperatures from -60 °C to +100 °C . . . .

example of designation 6302 2RSTL

TM . . . . . . grease for medium service temperatures from -35 °C to +140 °C . . . .

example of designation 6204 2ZRTM

TH . . . . . . grease for high service temperatures from -30 °C to +200 °C . . . .

example of designation 6202 2ZTH

TW. . . . . . grease for both low and high service temperatures from -40 °C to +150 °

C . . . . example of designation 6310 2ZC4TW

Note: The TM marking need not be stated on bearings and packing.

Bearings by special technical conditions

Single purpose bearings dimensions of which comply with the dimensional plan but the list of all charac-ters of extension expressing their technical characteristics would cause confusion of marking, can be upon agreement between manufacturer and customer replaced with basic designation, attaching the TPF or TPFK marking and a two- or three-digit number behind the basic designation of the bearing, which defines the number of the agreed technical specification determining all technical parameters of bearings.

TPF . . . . . bearings made by special technical conditions agreed with customer,
e.g. bearing 6205MA P66 by technical terms TPF 11142-71 is designated as follows: 6205MA P66 TPF 142.

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TPFK . . . . bearings by special technical terms agreed with customer which have high number of characters stating changes against the basic version. In this case, basic characters are replaced with designation TPFK containing relevant number of technical terms, e.g. bearing NU1015 made by technical terms. TPFK 11137-70 is designated as NU1015 TPFK137.

Bearings by special drawing documentation PLC

Bearings which by some of their dimension do not comply with the dimensional plan or are in line with the next development are marked with PLC by their manufacturer, as well as with other numerical characters. Usually they are single purpose bearings for one customer or a certain application method.

PLC ABC-DE.F (designation structure until 2012)

PLC . . . . . identification of special roller bearing

A . . . . design assembly

0 . . . . single row ball bearings

1 . . . . double row ball bearings:

2 . . . . thurst ball bearings

3 . . . . Not completed.

4 . . . . single row cylindrical roller, spherical-roller and needle roller bearings

5 . . . . double and multirow cylindrical roller, spherical-roller and needle roller bearings

6 . . . . single row, double row and four row tapered roller bearings

7 . . . . special double row bearings

8 . . . . assembly units and separate parts

9 . . . . thrust cylindrical roller, spherical roller, tapered roller and needle roller beariings

BC . . . . . . dimensional assembly – two digit characters

DE . . . . . . ordinal number within dimensional assembly – two digit characters

F . . . . difference in design - one digit or combination of numerical character and letter

Due to extending the assortment of special bearings, it was decided in 2013 to change the structure of designating special bearings: Upon the establishing of a new system, the designation on already produced bearings will not be changed.

PLC AB-CD-EF.G (designation structure since 2013)

PLC . . . . . identification of special roller bearing

A . . . . design assembly

1 . . . . ball bearings

2 . . . . thrust ball bearings

3 . . . . cylindrical roller bearings

4 . . . . thrust cylindrical roller bearings

5 . . . . needle roller bearings

6 . . . . spherical-roller bearings

7 . . . . spherical roller thrust bearings

8 . . . . tapered roller bearings

9 . . . . thrust tapered roller bearings

0 . . . . other bearings and mounting assemblies

B . . . . number of rolling units or bearings in mounting assemblies

CD . . . . . . dimensional assembly – two digit characters

EF . . . . . . ordinal number within dimensional assembly – two digit characters

G . . . . difference in design - one digit or combination of numerical character and letter

7.7 NEW FORCE bearings

In order to satisfy the needs of technically advanced customers, Dunlop BTL pays particular attention to technical development of products and investments in new technologies. The outcome of one of the recent key inno-vations is initiation of successive start up of production of Dunlop BTL bearings with higher quality standard with designation NEW FORCE.

The NEW FORCE bearings represent a new generation of Dunlop BTL bearings. Launching of bearings brings customers higher durability of bearings, enhanced operational safety, prolonged maintenance intervals and thus substantial reduction of operating costs. NEW FORCE bearings are designed for extreme locations of transmissions, railway vehicles, presses, rolling mills, paper machines, pumps, machine tools, power enginee-ring plants, polygraphic machines, etc.

As the first integrated new generation bearings, the radial spherical-roller bearings were launched on the market, double row self-aligning ball bearings, double row angular-contact ball bearings and thrust ball bearings. The next phase of launching bearings of this standard was the production assortment of bearings with outer diameter over 400 mm.

The achieved parameters of NEW FORCE bearings are the result of Dunlop BTL development in the following areas:

- Material of roller bearing components
- Technology of bearing ring flaring
- Optimisation of inner construction
- Surface treatments of bearing components

The achieved results allowed Dunlop BTL to offer NEW FORCE roller bearings with high utility properties to their customers:

- high dynamic load capacity
- low friction
- reliability in the extreme operating conditions

High durability of bearings

Increase of dynamic load capacity by 8% to 25% brings increase of durability of bearings by 30% up to 110%, comparing to the up-to-now designs.

Increase of dynamic load capacity allows customer to design construction with smaller dimensions to transfer the same load. Thus Dunlop BTL brings to their customer an opportunity to reduce total price of the equipment, and achieve power savings during operation.

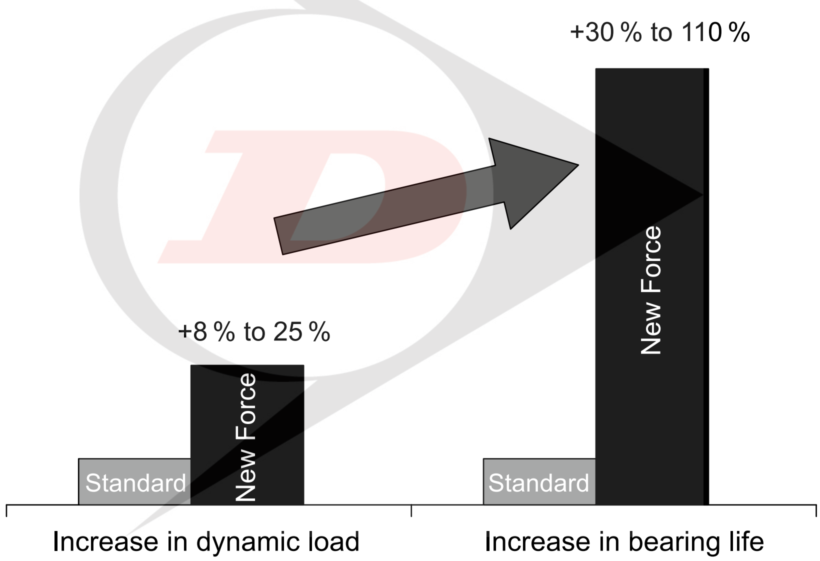

Fig. 7.13

Use of quality bearing material

Steels for production of bearings meet the parameters of international standards defined by ISO 683-17. Production of bearing rings and rolling elements utilised high quality material of selected smelting houses. Long-term cooperation with suppliers ensures continuous process of improving parameters of input material.

Key quality parameters of steel and its processing affect the service properties of bearing, i.e. resistance to fatigue damage, abrasion resistance and dimensional stability. These are:

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- **chemical composition and heat treatment**

Selection of the type of bearing steel and optimisation of heat treatment conditions is conducted by the dimension of the component. The heat treatment processing technology of NEW FORCE bearings ensures stabile hardness values of bearing components in the entire section. Spherical-roller bearing components are heat treated to ideal material structure and hardness that enable using of the bearings at service temperatures to 200 °C. The final material structure ensures dimensional stability of bearing components throughout their service life.

- **Content of non-metal intrusions – micropurity**

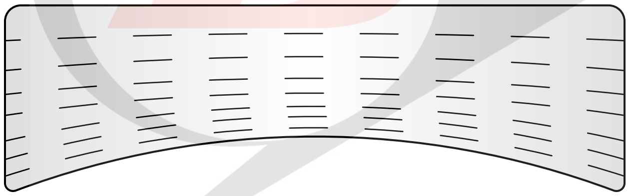
Reduction of content of non-metal intrusions is the key quality parameter in the bearing steel metallurgy development. In production of bearings, Dunlop BTL utilises bearing steel with minimum oxygen content.

- **Type of semiproduct**

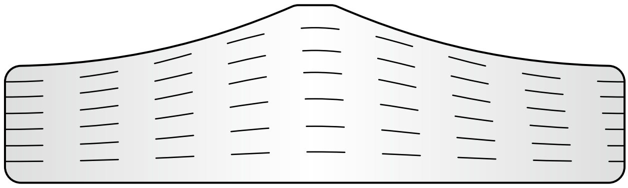
The quality of bearing and production economics are affected also by selection of the semiproduct type. The level of forming and positive angle of forming fibre contact towards the orbit are the parameters that positively increase resistance of the NEW FORCE bearings against fatigue damage,

Technology of bearing ring rolling

Basic research demonstrated effect of material fibre direction towards the contact surface to the durability of bearings. Most convenient is such layout of fibres when their direction is in parallel with the contact surface. With increasing fibre direction angle towards the contact surface the durability decreases. The technology of cold or semi-heating rolling brought an ideal material structure of the NEW FORCE bearings in order to achieve higher durability of bearings.



Threads 1 - after rolling (outer ring)



Threads 2 - after rolling (inner ring)

Fig. 7.14

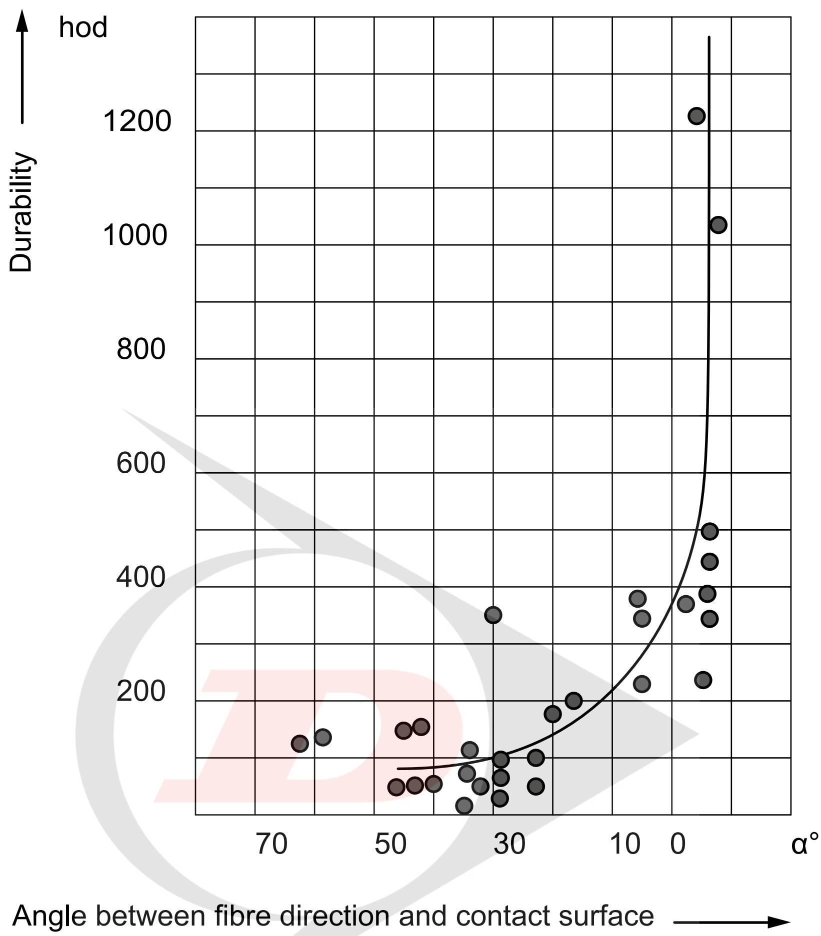


Fig. 7.15

Optimised design and inner geometry

Advanced design and calculation programs, together with new bearing production technologies, enabled opti-misation of inner construction of bearings and improved accuracy of functional areas. Thus the NEW FORCE version bearings achieved better quality of functional surfaces and improved course of discharge voltages in bearing component sections, comparing to the standard bearing designs. This brings reduced noise level and higher accuracy of bearing run, as well as extended durability of bearings.

Special surface treatment

Within innovation programs, a new design of sheet cages for radial and thrust spherical-roller bearings was launched in the production. Cages are made of steel plate with surface treatment in order to improve slip proper-ties and reduce wear of cages. The design of cages allows achieving better lubrication and extended service life of bearings. Surface treatments of bearing components represent a well tested way of improving bearing properties for certain locations. The benefit of surface layers lies in better keeping the lubricant in the rolling contact, reduced friction and enhanced resistance to wear and corrosion. We recommend that suitability of sur-face treatment for special operating condition is discussed with the technical and consultancy services of Dunlop BTL.

Bearings NEW FORCE +

Dunlop BTL bearings with NEW FORCE+ marking represent a brand new generation of Dunlop BTL bearings which is characterised by an innovated modification of the bearing inner structure geometry towards optimum voltage course in the area of rolling contact. This Dunlop BTL bearings´ innovation is associated with further enhancement of accuracy, comparing to the standardly produced bearing assortment, including the NEW FORCE bearings.

Optimisation of the shape of rolling surfaces brings improved dynamic load capacity of bearings and thus also significant extension of bearings´ durability. Development of the NEW FORCE+ generation is associated with the introduction of new calculation methods in the structure of bearings based on FEM and production upgrade by introducing numerically controlled machines that enable achieving final shapes of functional surfaces with modified geometry.

With regard to the fact that the entire design optimisation and production process of modified parts is unique for every bearing application, the NEW FORCE+ bearing generation is not designed to be launched in the standard production program of Dunlop BTL. The bearings will be manufactured upon request for extreme locations for selected OEM customers.

7.8 Technical support

Dunlop BTL operates as bearing manufacturer and supplier already since 1947. Since the beginning, the company has been cooperating with their customers worldwide. This allows continuous expansion of the Dunlop BTL rolling bearing production assortment offered in maximum quality at reasonable price. Experience in operation of bearings obtained in cooperation with customers, along with continuous education of their employees allows ongoing development of technical support to Dunlop BTL customers and extension of services for Dunlop BTL bearing users.

Proposal verification

The Dunlop BTL bearings´ structure and their basic parameters are designed by the Dunlop BTL´s own well tested metho-dologies that adhere to the international ISO standards. Designing new bearings utilises most sophisticated design and calculation CAD systems. Designs of new bearings are optimised and their rigidity checked by means of FEM based numerical calculations. When creating designs, information obtained in achieved test results and experiences from production and operation of Dunlop BTL bearings are utilised.

Verification of quality parameters of Dunlop BTL bearings

Parameters of Dunlop BTL rolling bearings are verified in tests within development, as well as in periodical quality assessment during series production. Tests are conducted according to the company´s own methods in the test stations of the bearing test room. Bearing and input material tests results are analysed and serve as the basis for new design, technological and investment solutions.

Technical support for Dunlop BTL bearing users

Customer needs are solved by fully available workers of Dunlop BTL technical and consultancy services. Expert workers are ready to solve operatively requests and questions of Dunlop BTL bearing users in the area of selection of bearings, design of rolling location and assembly procedures. Dunlop BTL technical support provides users with information in the area of roller bearings, accessories and tribology. Upon user´s request it also provides professional supervision over assembly and disassembly of bearings directly at customer, and organizes professional training course of user employees. It cooperates with manufacturers in development of rolling location. It draws up expert opinions on broken bearings. It determines causes of accidents and proposes measures to prevent them.

8. Bearing applications

8.1 Arrangement of bearings

To locate rotary shaft you need at least two bearings that are located in certain distance from each other. Depending on the application method, location with axially free and axially guiding bearing is selected; prestressed location or floating arrangement of bearings. See figure 4.12 in chapter Bearing type selection for examples of bearing arrangements.

8.1.1 Location with axially free and axially guiding bearing

Axially guiding bearing on one shaft end brings besides radial load element also axial element in both direc-tions. For the above reason, it has to be secured both in the shaft and in the body. Axially free bearing in location compensates production inaccuracies in location and, first of all, changes in dimensions in operation due to increased temperatures. An ideal axially free bearing is roller bearing in N and NU design the rolling bodies of which can move on the raceway of bearing ring without guide flanges. Bearings of the other types, such as ball bearings and spherical-roller bearings, can be used as axially free only if one of bearing races is push-located.

Axially guide bearing guides shaft in axial direction and besides radial forces captures also axial forces. Selection of bearing type to be used as axially guide bearing depends on the size of axial load and on requi-rements for accuracy of shaft location. Double row angular-contact ball bearing ensures more accurate axial guidance than e.g. ball or spherical-roller bearing. Accurate axial guidance can be achieved also by a pair of tapered roller bearings which are used as axially guide bearing. At lower axial load even NUP cylindrical roller bearing can be used as axially guide bearing.

8.1.2 Symmetrical arrangement of bearings

This type of location suits mainly short shafts. It features shaft being guided in one direction by one bearing and in other direction by other bearing. Suitable bearings for this type of arrangement are all radial bearings that allow transfer of axial force at least in one direction. In this arrangement, prestressed bearings can be mounted (fig. 8.1).

8.1.3 Prestressed location

Location of prestressed bearing usually consists of symmetrically placed ball bearings with angular contact, or of tapered roller bearings. Prestress is achieved by use of springs. Such design compensates thermal dila-tation. It is used in case when idle bearings can be exposed to vibrations. Prestressed bearings can reduce noise level, especially in small electric motors.

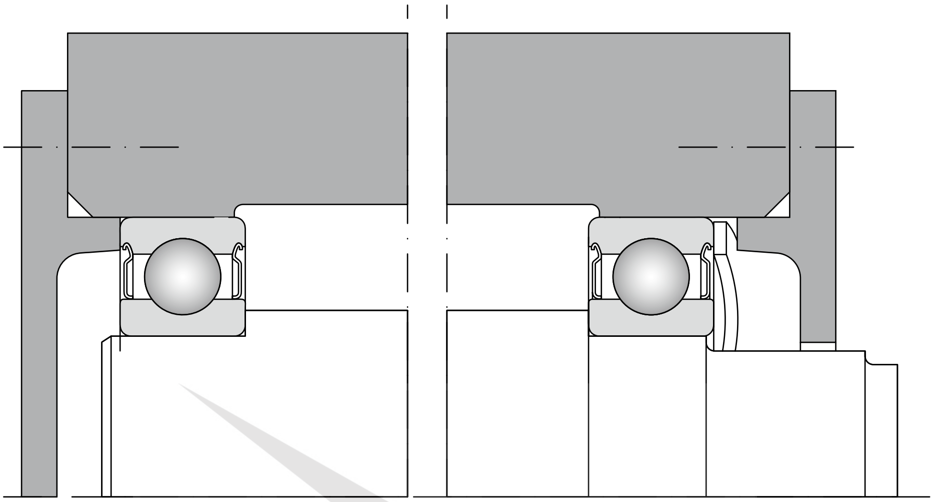


Fig. 8.1

Spring acts on outer race of one of the bearings whilst relevant outer race has to allow axial displacement in the body. Prestress remains practically constant even though the bearing axially moves due to thermal dilatation. Required prestress can be calculated using the below relation:

$$F = k \cdot d$$

F . . . . Prestress force (kN)

k . . . . coefficient, see next

d . . . . bearing hole diameter [mm]

Depending on design of electric motor, the coefficient may reach values of 0.005 up to 0.01.

If prestress is supposed to prevent bearing from getting damaged due to vibrations, it has to be set to higher level.

Then k = 0.02 has to be selected.

This method is however not suitable for locations that must feature high rigidity where the direction of acting load changes, or where shock load acts.

If certain optimum prestress value is exceeded, rigidity increases only insignificantly whilst friction and also service temperature in the bearing grow rapidly. This reduces durability of bearing since additional constant load acts on it. Informative relation between durability and prestress – clearance – is indicated in diagram in fig. 8.2.

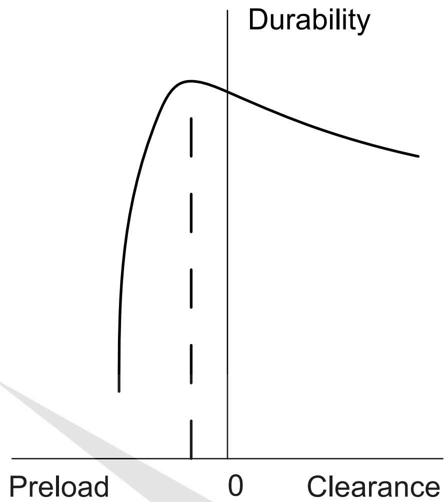


Fig. 8.2

8.2 Location design – General principles

Properties of bearings are fully utilised only when bearing races are supported along the entire circumference and width of raceways. Solid support surface can be of either cylindrical or tapered shape, in thrust bearings the surface is flat. Support surfaces must be manufactured to have adequate accuracy, and must not be provided with grooves, holes, etc. Besides that, bearing races must be reliably secured to prevent them from turning in the body or on the shaft.

Suitable radial security and adequate support can only be achieved if bearing rings are mounted with overlap. If however easy assembly and disassembly are required, alternatively axial transferability of axially free bearing, fixed location of the ring cannot be selected.

Where free location is chose, provisions must be adopted to avoid irrevocable wear during shifting the ring.

Rotating shaft or another component located in roller bearings is guided by them in radial and axial direction so that the principal condition of definiteness of its movement is achieved. If possible, the component should certainly be located, i.e. supported radially on two spots and axially in one spot.

Examples of such location are shown in figures 4.12. Most common location is such where the shaft is located radially in two bearings one of which locks it in axial direction. Guide (fixed) bearing transfers radial load and also axial load in both directions. Radial bearings are mostly used as guide. They are able to transfer combined load, e.g. single row ball bearings, double row angular-contact bearings, double row self--aligning ball bearing, double row spherical-roller bearings or single row angular-contact ball bearings and tapered roller bearings. The lastly mentioned two bearing types must be assembled in pairs. Free bearing only transfers radial load and must allow certain displacement of the shaft in axial direction in order to prevent occurrence of undesired prestress caused by external effects (thermal dilatation, production inaccuracy of connecting location components, etc.).

Axial displacement can be achieved by shifting between one of the body rings and machine components directly associated with the bearing, e.g. between the outer bearing ring and the bore in the body (fig. 4.12a, b), or directly in the bearing (fig. 4.12 c to h).

Locations where higher radial force and axial load in higher revolution frequency act should be solved by the bearings capturing only radial or axial forces, see fig. 8.3.

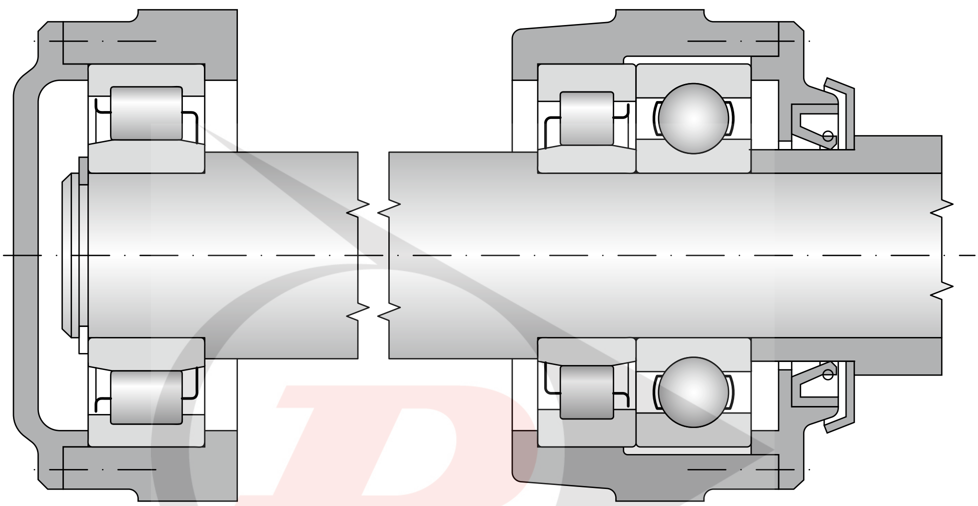


Fig. 8.3

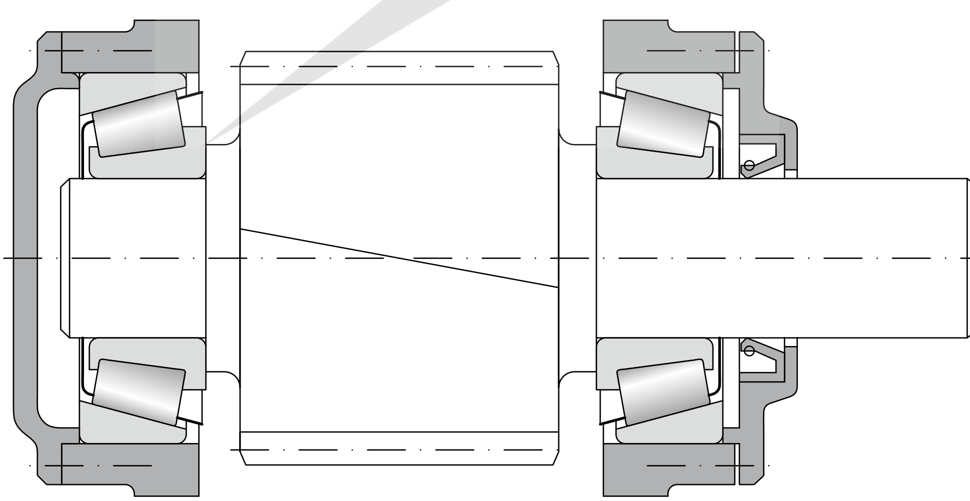


Fig. 8.4

In these cases, any of radial bearings can be used for radial guidance, and those radial bearings for axial guidance that feature the ability to transfer also axial load, alternatively a pair of these bearings or double direction thrust bearings or a pair of single direction thrust bearings. Condition is that axially guide bearings have to be located with radial clearance.

Another frequently used solution is location in two bearings the design of which allows capturing of both radial and axial load in both directions. Axial load is captured in turns by both bearings, always by the direc-tion in which forces act and, at the same time, they transfer also radial load. An example of such location is shown in figure 8.4.

In this case, a pair of single row tapered roller or single row angular-contact ball bearings is used as a well tested construction. Also other types of bearings that are able to transfer load in radial and axial direction at the same time can be used, e.g. single row bearings, alternatively single row cylindrical roller bearings in NJ design, etc.

Radial and axial security of bearing on journal and in body bore or in another part has direct connection with the overall design location arrangement. When selecting the method of fixation, the character and inten-sity of acting forces has to be considered particularly, as well as service temperature at the point of location and the material of connecting components.

When specifying the dimensions of connecting parts, the designer needs to consider also the assembly and disassembly method and maintenance actions, besides the type and dimensions of the bearing.

8.2.1 Radial security of bearings

Bearing is fixed in radial direction on fitted cylindrical surface on the surface of the journal and bore in the body. In some cases of fixation on journal, adapter or withdrawal sleeve is used; alternatively the bearing can be fixed directly on tapered journal.

Proper radial fixation of bearing on journal and in body is very important for utilisation of its loading capa-city and correct location function. In doing so, the following aspects need to be considered:

a) safe fixation and uniform support of rings

b) easy assembly and disassembly

c) displacement of free bearing in axial direction

In principle, both bearing rings should be fixed firmly since only this way their reliable support on the entire circumference and radial fixation against spinning can be achieved. To simplify assembly and disassembly or in order to shift free bearing, one of the rings can be located as sliding.

If proper radial fixation of bearing is selected, one needs to evaluate and consider the effect of the method of rotation and intensity of load.

Circumferential load

Circumferential load occurs when relevant bearing ring turns, and the direction of load does not change, or when the ring does not turn and the load rotates. The bearing ring circumference is loaded successively in one revolution. In this case, loaded ring must be always fixed with necessary overlap.

Spot load

Spot load occurs when the bearing ring stands and outer force is directed still in the same spot of the race-way, or when the ring and force rotate at the same revolution frequency. The ring to which the spot load acts can be located with clearance (mobile), if the conditions require so.

Uncertain way of loading

In case of uncertain way of loading, the ring is acted on by variable external forces the direction and change of load of which cannot be determined (e.g. unbalanced masses, shocks, etc.). Uncertain way of loading requires that both rings are located with overlap (firmly). Under this condition in majority cases of location bearings with increased radial clearance have to be selected.

Load intensity

The load directly affects selection of the size of overlap in location. The bigger the load of the bearing, the bigger overlap in location has to be selected. This particularly applies in cases of shock and vibration load of the bearing. Fixed location on journal or in bore of the body induces deformation of ring, which reduces radial clearance. To ensure the needed radial clearance in cases of fixed location, sometimes bearings with increased radial clearance have to be used. Final clearance after assembly depends on the type and size of the bearing. Therefore the size of needed overlap of fitted ring has to be considered by the type and size of the bearing. For bearings of smaller dimensions smaller overlaps are selected, and vice versa. Relatively smaller overlaps are used e.g. for ball bearings of the same bigness comparing to cylindrical roller, tapered roler or spherical roller bearings.

Material and design of connecting pieces

Designing and determination of tolerances of connecting parts must take into account the materials used, as well as the construction of the connecting pieces. Results of practical experiences reflect in the below stated charts. When bearings are mounted in bodies made of light metal alloys or on journals of hollow shafts, location with higher overlaps has to be selected.

Split bodies are not suitable for locations with big overlaps since they represent a risk of gripping the bearing in the dividing plane of the body.

Heating and warmth

Warmth generated in bearing may lead to release of overlap on the journal which may cause spinning the ring. An opposite case may occur in the body. Heating causes clearance adjustment which will limit up to eliminate axial displacement of the ring of free bearing in the body. Therefore we need to be very attentive to this factor when designing the location.

Accuracy of bearing surfaces

Accuracy of bearing surfaces in terms of tolerances and geometrical shapes is important since it may transfer to raceways of bearing rings. First of all, this has to be reflected in location designs which are highly focused on the running accuracy. Major share of inequality is transferred in thin profiles of bearing rings.

When normal accuracy level bearings are used, usually tolerances within the tolerance level IT6 are selected for the bearing surface on the journal, whilst for the bearing surface in the body the selected tolerance level is IT7.

For ball and cylindrical roller bearings of smaller dimensions, IT5 level can be used for the journal and IT6 for the bore in the body.

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For bearings of higher accuracy levels, for locations with high accuracy requirements, e.g. machine tool spin-dles, the recommended least level is IT5 for the shaft, and at least IT6 for the body.

Table 8.1

Recommended accuracies of the shape of bearing surfaces for bearings			
Accuracy level of bearing	**Location place**	**Admissible deviation of cylindricality**	**Admissible frontal runout of support surfaces towards the axis**
P0, P6	shaft	IT5/2	IT3
	body	IT6/2	IT4
P5, P4	shaft	IT3/2	IT2
	body	IT4/2	IT3

Table 8.2

Basic tolerances IT2 to IT6						
Nominal diameter		**Tolerance level**				
over	**to**	**IT2**	**IT3**	**IT4**	**IT5**	**IT6**
mm		**µm**				
6	10	1,5	2,5	4	6	9
10	18	2	3	5	8	11
18	30	2,5	4	6	9	13
30	50	2,5	4	7	11	16
50	80	3	5	8	13	19
80	120	4	6	10	15	22
120	180	5	8	12	18	25
180	250	7	10	14	20	29
250	315	8	12	16	23	32
315	400	9	13	18	25	36
400	500	10	15	20	27	40

Allowed deviation of roundness and cylindricality and allowed frontal run out of bearing and support surfaces for bearings must be smaller against the axis than the scope of tolerance of the diameters of the journal and the bore. With increasing accuracy of the bearings used, also the requirements for the accuracy of bearing surfaces grow. The recommended accuracy values of the bearing surfaces shape for bearings are stated in chart 8.1, and general tolerances IT2 to IT6 in chart 8.2

Assembly and disassembly of bearing

If any of the rings is located with clearance (mobile), the assembly is easy. If the service conditions require that both rings are located with overlap, a suitable type of bearing has to be chosen, e.g. separable bearing (tapered, cylindrical, needle), or a bearing with tapered bore. Shaft journals for location of sleeves for bearing with tapered bore can be within the h9 or h10 tolerance, geometrical shape must be within the accuracy IT5 or IT7, depending on the complexity of location.

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Axial displacement of free bearing races

At any service conditions the axial displacement of free bearing has to be ensured. If non-separable bearings are used, displacement of spot-loaded ring will be reached by locating with clearance (mobile location). In bodies made of light metal alloys the bore has to be sleeved with a steel sleeve, if outer ring is to be located with clearance. Reliable sliding ability in axial direction will be achieved if cylindrical roller bearings of N and NU designs or radial needle roller bearings are used in the location.

The recommended tolerances of journal and hole diameters of connecting pieces are for radial and axial bearings stated in charts 8.3 to 8.10.

Table 8.3

Tolerances of journal diameters for radial bearings (applies for full steel shafts)

Service conditions	Examples of location	Journal diameter [mm]			Tolerance
		Ball bearings	Cylindrical roller, needle roller [1], tapered roller bearings	Spherical roller bearings	
Inner ring spot load					
Small and normal load Pr ≦ 0.15 Cr	Free wheel, pulleys, belt pulleys		All diameters		g6 [2]
Big impact load Pr > 0.15 Cr	Wheels of conveyance trolleys, tension pulleys				h6
Circumferential load of inner ring or uncertain way of loading					
Small and variable load Pr ≦ 0.07 Cr	Conveyers, fans	(18) to 100 (100) to 200	≦ 40 (40) to 140	- -	j6 k6
Normal and big load Pr > 0.07 Cr	General engineering, pumps, combustion engines transmissions, woodworking machines	≦ 18 (18) to 100 (100) to 140 (140) to 200	- ≦ 40 (40) to 100 (100) to 140 (140) to 200 > 200	- - ≦ 40 (40) to 65 (65) to 100 (100) to 140 > 140	j5 k5 (k6) [3] m5 (m6) [3] m6 n6 p6
Extremely big load, shocks heavy service conditions Pr > 0.15 Cr	Axle bearings of rail vehicles, traction motors rolling mills	- - -	(141) to 140 (140) to 500 > 500	(101) to 100 (100) to 500 > 500	n6 [4] p6 [4] r6 (p6) [4]
High location accuracy at small load Pr ≦ 0.07 Cr	Machine tools	≦ 18 (18) to 100 (100) to 200	- ≦ 40 (40) to 140 (140) to 200	- - - -	h5 [5] j5 [5] k5 [5] m5)
Axial load exclusively			all diameters		j6
Bearings with tapered bore and with adapter or withdrawal sleeve or dismantling sleeve					
All ways of loading	General locations, axle bearings of rail vehicles, Unexacting locations		all diameters		h9/IT5 h10/IT7

[1] Does not apply to needle bearings without rings
[2] For bearings tolerance f6 can be selected to ensure axial shift
[3] Tolerance in brackets is selected usually for single row tapered roller bearings or at low frequency revolutions where clearance diffusion does not have major significance.
[4] Bearings with increased radial clearance have to be used
[5] Tolerances for single row ball bearings of accuracy P5 and P4 are stated in chapter 12.2

Table 8.4

Tolerance of diameters of radial bearing body bores (applies to bodies of steel, alloy and cast steel)				
Service conditions	**Sliding ability of outer racew**	**Body**	**Examples of location**	**Tolerance**
		Circumferential load of outer ring		
Big shock load Pr > 0.15 Cr Thin-walled elements	Does not slide	Single piece	Hubs with roller bearings, crank pin bearings	P7
Normal and big load Pr > 0.07 Cr	Does not slide		Hubs with roller bearings travelling wheels of cranes, crank shaft bearings	N7
Small and variable load Pr ≦ 0.07 Cr	Does not slide		Converyer rollers, tension pulleys	M7
		Uncertain way of loading		
Big shock load Pr > 0.15 Cr	Does not slide		Traction motors	M7
Big and normal load Pr > 0,07 Cr	Usually does not slide	Single piece	Electromotors, pumps, fans, crank shafts	K7
Small and variable load Pr ≦ 0.07 Cr	Usually sliding		Electromotors, pumps, fans, crank shafts	J7
		Accurate locations		
Small load Pr ≦ 0.07 Cr	Usually does not slide	Single piece	Roller bearings for machine tools, ball bearings for machine tools, small electromotors	K6 [1]
	Sliding			J6 [2]
	Slightly pushing			H6
		Spot load of outer ring		
Optional load	Slightly pushing	Single piece or two piece	General engineering axle bearings of rail vehicles	H7 [3]
Small and normal load Pr ≦ 0.15 Cr	Slightly pushing	Single piece or two piece	General engineering less exacting mechanical eingineering	H8
			Paper machine drying cylinders, big electromotors	G7 [4]

1) For big load, stronger M6 or N6 tolerances are selected. For cylindrical roller bearings with tapered bore, tolerances K5 or M5 are selected.
2) Tolerances for single row ball bearings of accuracy P5 and P4 are stated in chapter 12.2
3) For bearings with outer diameter D < 250 mm with thermal difference between outer ring and body above 10 °C, tolerance G7 is selected
4) For bearings with outer diameter D > 250 mm with thermal difference between outer ring and body above 10 °C, tolerance F7 is selected.

Table 8.5

Tolerance of journal diameters for axial bearings				
Bearing type	**Way of loading**		**Journal diameter** [mm]	**Tolerances**
Axial ball	Axial load exclusively		All diameters	j6
Axial spherical-roller				j6
	Current axial and radial load	Spot load of shaft ring	All diameters	j6
		Circumferential load of shaft ring or uncertain way of loading	≦ 200	k6
			(200) to 400	m6
			> 400	n6

Table 8.6

Tolerance of diameters of axial bearing body bores				
Bearing type	**Way of loading**		**Note**	**Tolerances**
Axial ball	Axial load exclusively		In common locations, the casing ring may feature clearance	H8
			Casing ring is mounte with radial clearance	-
Axial spherical-roller	Current axial and radial load	Spot load or uncertain way of loading of casing ring		H7
		Circumferential load		M7
		Circumferential load		

Table 8.7

Limit deviations of journal diameter tolerances																	
Nominal diameter of journal		**f6**		**g5**		**g6**		**h5**		**h6**		**j5**		**j6(js6)**		**k5**	
over	**to**	**upper**	**lower**	**upper**	**lower**	**upper**	**lower**	**upper**	**lower**	**upper**	**lower**	**upper**	**lower**	**upper**	**lower**	**upper**	**lower**
mm		**μm**															
1	3	-6	-12	-2	-6	-2	-8	0	-4	0	-6	2	-2	4	-2	4	0
3	6	-10	-18	-4	-9	-4	-12	0	-5	0	-8	3	-2	6	-2	6	1
6	10	-13	-22	-5	-11	-5	-14	0	-6	0	-9	4	-2	7	-2	7	1
10	18	-16	-27	-6	-14	-6	-17	0	-8	0	-11	5	-3	8	-3	9	1
18	30	-20	-33	-7	-16	-7	-20	0	-9	0	-13	5	-4	9	-4	11	2
30	50	-25	-41	-9	-20	-9	-25	0	-11	0	-16	6	-5	11	-5	13	2
50	80	-30	-49	-10	-23	-10	-29	0	-13	0	-19	6	-7	12	-7	15	2
80	120	-36	-58	-12	-27	-12	-34	0	-15	0	-22	6	-9	13	-9	18	3
120	180	-43	-68	-14	-32	-14	-39	0	-18	0	-25	7	-11	14	-11	21	3
180	250	-50	-79	-15	-35	-15	-44	0	-20	0	-29	7	-13	16	-13	24	4
250	315	-56	-88	-17	-40	-17	-49	0	-23	0	-32	7	-16	16	-16	27	4
315	400	-62	-98	-18	-43	-18	-54	0	-25	0	-36	7	-18	18	-18	29	4
400	500	-68	-108	-20	-47	-20	-60	0	-27	0	-40	7	-20	20	-20	32	5
500	630	-76	-120	-	-	-22	-66	-	-	0	-44	-	-	22	-22	-	-
630	800	-80	-130	-	-	-24	-74	-	-	0	-50	-	-	25	-25	-	-
800	1000	-86	-142	-	-	-26	-82	-	-	0	-56	-	-	28	-28	-	-
1000	1250	-98	-164	-	-	-28	-94	-	-	0	-66	-	-	33	-33	-	-

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Table 8.8

Limit deviations of journal diameter tolerances																	
Nominal diameter of journal		k6		m5		m6		n6		p6		h9[1]		IT5	h10[1]		IT7
over	to	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower		upper	lower	
mm		µm															
1	3	6	0	6	2	8	2	10	4	12	6	0	-25	4	0	-40	10
3	6	9	1	9	4	12	4	16	8	20	12	0	-30	5	0	-48	12
6	10	10	1	12	6	15	6	19	10	24	15	0	-36	6	0	-58	15
10	18	12	1	15	7	18	7	23	12	29	18	0	-43	8	0	-70	18
18	30	15	2	17	8	21	8	28	15	35	22	0	-52	9	0	-84	21
30	50	18	2	20	9	25	9	33	17	42	26	0	-62	11	0	-100	25
50	80	21	2	24	11	30	11	39	20	51	32	0	-74	13	0	-120	30
80	120	25	3	28	13	35	13	45	23	59	37	0	-87	15	0	-140	35
120	180	28	3	33	15	40	15	52	27	68	43	0	-100	18	0	-160	40
180	250	33	4	37	17	46	17	60	31	79	50	0	-115	20	0	-185	46
250	315	36	4	43	20	52	20	66	34	88	56	0	-130	23	0	-210	52
315	400	40	4	46	21	57	21	73	37	98	62	0	-140	25	0	-230	57
400	500	45	5	50	23	63	23	80	40	108	68	0	-155	27	0	-250	63
500	630	44	0	-	-	70	26	88	44	122	78	0	-175	30	0	-280	70
630	800	50	0	-	-	80	30	100	50	138	88	0	-200	35	0	-320	80
800	1000	56	0	-	-	90	34	112	56	156	100	0	-230	40	0	-360	90
1000	1250	66	0	-	-	106	40	132	66	186	120	0	-260	46	0	-420	105

[1] In journals manufactured within tolerances h9 and h10 for bearings with adapter or withdrawal sleeve, the circularity and cylindricity deviations must not exceed the basic tolerance IT5 and IT7.

Table 8.9

Limit deviations of bore diameter tolerances																	
Nominal diameter of bore		F7		G6		G7		H6		H7		H8		J6(Js6)			
over	to	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower		
mm		µm															
6	10	28	13	14	5	20	5	9	0	15	0	22	0	5	-4		
10	18	34	16	17	6	24	6	11	0	18	0	27	0	6	-5		
18	30	41	20	20	7	28	7	13	0	21	0	33	0	8	-5		
30	50	50	25	25	9	34	9	16	0	25	0	39	0	10	-6		
50	80	60	30	29	10	40	10	19	0	30	0	46	0	13	-6		
80	120	71	36	34	12	47	12	22	0	35	0	54	0	16	-6		
120	180	83	43	39	14	54	14	25	0	40	0	63	0	18	-7		
180	250	96	50	44	15	61	15	29	0	46	0	72	0	22	-7		
250	315	108	56	49	17	69	17	32	0	52	0	81	0	25	-7		
315	400	119	62	54	18	75	18	36	0	57	0	89	0	29	-7		
400	500	131	68	60	20	83	20	40	0	63	0	97	0	33	-7		
500	630	146	76	66	22	92	22	44	0	70	0	110	0	22	-22		
630	800	160	80	74	24	104	24	50	0	80	0	125	0	25	-25		
800	1000	176	86	82	26	116	26	56	0	90	0	140	0	28	-28		
1000	1250	203	98	94	28	133	28	66	0	105	0	165	0	33	-33		
1250	1600	235	110	108	30	155	30	78	0	125	0	195	0	39	-39		

Table 8.10

Limit deviations of bore diameter tolerances															
Nominal diameter of bore		J7(Js7)		K6		K7		M6		M7		N7		P7	
over	to	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
mm		µm													
6	10	8	-7	2	-7	5	-10	-3	-12	0	-15	-4	-19	-9	-24
10	18	10	-8	2	-9	6	-12	-4	-15	0	-18	-5	-23	-11	-29
18	30	12	-9	2	-11	6	-15	-4	-17	0	-21	-7	-28	-14	-35
30	50	14	-11	3	-13	7	-18	-4	-20	0	-25	-8	-33	-17	-42
50	80	18	-12	4	-15	9	-21	-5	-24	0	-30	-9	-39	-21	-51
80	120	22	-13	4	-18	10	-25	-6	-28	0	-35	-10	-45	-24	-59
120	180	25	-14	4	-21	12	-28	-8	-33	0	-40	-12	-52	-28	-68
180	250	30	-16	5	-24	13	-33	-8	-37	0	-46	-14	-60	-33	-79
250	315	36	-16	5	-27	16	-36	-9	-41	0	-52	-14	-66	-36	-88
315	400	39	-18	7	-29	17	-40	-10	-46	0	-57	-16	-73	-41	-98
400	500	43	-20	8	-32	18	-45	-10	-50	0	-63	-17	-80	-45	-108
500	630	35	-35	0	-44	0	-70	-26	-70	-26	-96	-44	-114	-78	-148
630	800	40	-40	0	-50	0	-80	-30	-80	-30	-110	-50	-130	-88	-168
800	1000	45	-45	0	-56	0	-90	-34	-90	-34	-124	-56	-146	-100	-190
1000	1250	52	-52	0	-66	0	-105	-40	-106	-40	-145	-66	-171	-120	-225
1250	1600	62	-62	0	-78	0	-125	-48	-126	-48	-173	-78	-203	-140	-265

8.2.2 Axial security of bearings

Inner bearing ring with cylindrical bore seated on journal with overlap (fixed location) is usually locked in axial direction using a adapter nut, terminal plate or snap ring whilst the other face is usually leaned by the shaft fitting. Adjacent components are used as support faces for inner rings and, if needed, spacer rings are inserted between this component and the inner ring of the bearing. Examples of axial fixation of bearing are shown in figure 8.5.

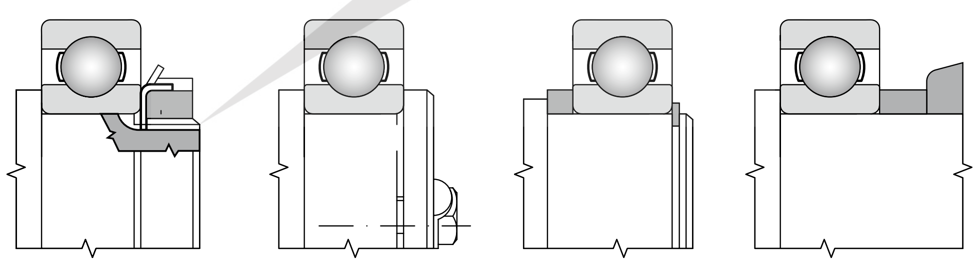

Fig. 8.5

Bearings with tapered bore mounted directly on tapered journal are usually secured with a safety nut screwed onto the thread on the shaft. If bearings are mounted on withdrawal sleeve, the inner ring must be supported, e.g. by a spacer ring. The spacer ring can form a part of labyrinth. The withdrawal sleeve is axially fixed with terminal plate or safety nut.

Examples of axial fixation of bearing with tapered bore directly on tapered journal or by means of adapter or withdrawal sleeve are shown in Fig. 8.6.

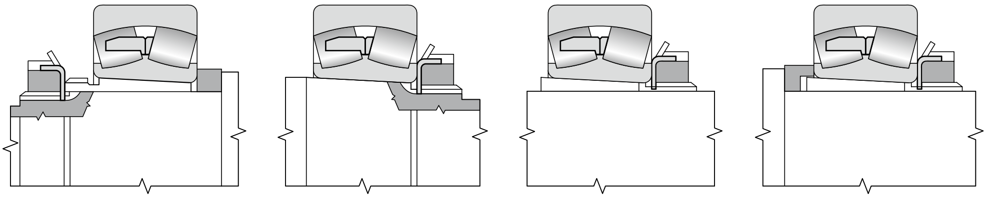

Fig. 8.6

Admissible axial load of bearings fixed by means of adapter sleeve on smooth shafts without the bearing leaning on shaft fitting is calculated by the below equation:

$$\mathbf{F_a = 3B \cdot d} \qquad [N]$$

F_a . . . . . . admissible axial load of bearing [N]

B . . . . . . bearing width [mm]

d . . . . bearing hole diameter [mm]

If axial displacement of outer ring in body is not desirable, we can use a solution utilising the front support surface or seating surface of the bearing lid, nut or snap ring. Bearings with a groove for snap ring (NR) are less demanding in space, and their locking is simple.

Examples of solution are shown in Fig. 8.7.

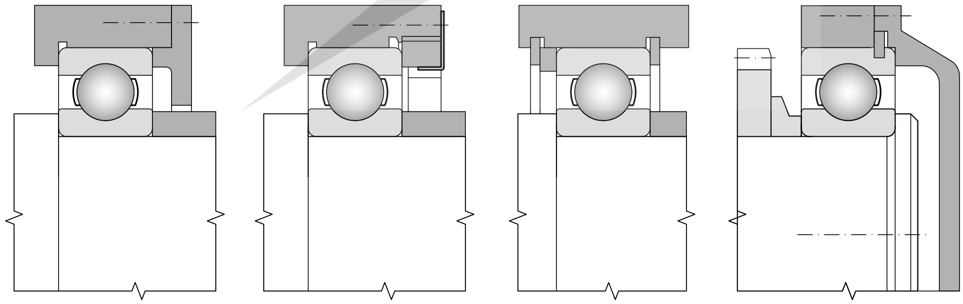

Fig. 8.7

Connecting dimensions for individual bearing types are stated in this publication in the chart section (chapter 12).

8.3 Seal

Sealing the bearing space is very important since harmful substances present in the proximity of the bearing affect it and often even put it out of service. Seal has also an opposite function – it prevents the grease from leaking out of the bearing and from the stowage compartment. For that reason, the seal has always to be designed considering the service conditions of the machine or equipment, lubrication method, maintenance options and economic aspects of production and use.

8.3.1 Contact-free sealing

This type of seal features only a tight gap between the non-rotary and rotary component which is sometimes filled with grease. In this design no wear due to friction occurs, and therefore this seal suits to use for highest circumferential speeds and high service temperatures. Examples of slotted seals are shown in fig. 8.8

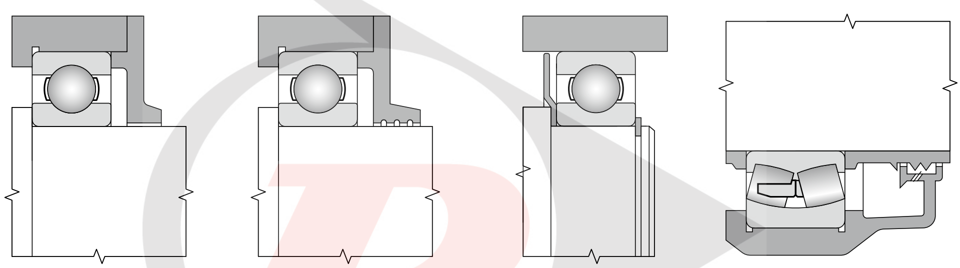

Fig. 8.8

Another very efficient seal is a labyrinth seal which can be used to enhance the packing effect by higher number of labyrinths or extension of sealing slots. See fig. 8.9. for examples of this seal.

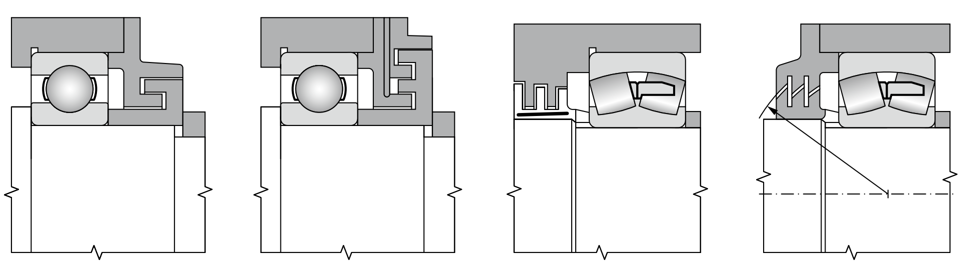

Fig. 8.9

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8.3.2 Friction sealing

Friction sealing is made of elastic or soft but sufficiently solid and impermeable material that is inserted between the rotary and fixed component. Such seal is usually cheap and suits to various constructions. Disadvantage is sliding friction touching the surfaces which limits the use of it for high circumferential speeds.

The simplest is seal with a felt ring (fig. 8.10). It suits to service temperatures within -40°C and +80°C and to circumferential even to 7 $m.s^{-1}$, whilst the maximum required surface roughness of the sliding surface is R_a = 0.16, and minimum hardness 45 HRC or treatment by hard chromium plating. Dimensions of felt rings and grooves are solved by relevant national standards.

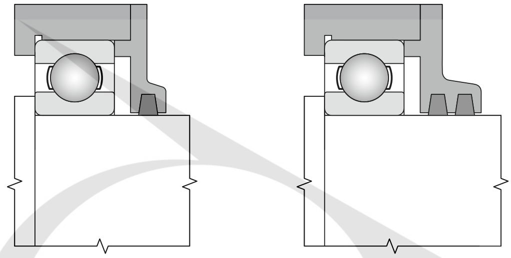

Fig. 8.10

A very frequent sealing method is sealing with shaft rings (fig. 8.11). Shaft rings are made of rubber or other suitable plastics, stiffened by metal stiffener. According by the material used they suit to service temperatures from -30°C to +160°C. Admissible circumferential speed depends on the roughness of the sliding surface roughness.

- to 2 $m.s^{-1}$ the roughness is max R_a = 0.8,
- to 4 $m.s^{-1}$ the roughness is max R_a = 0.4,
- to 12 $m.s^{-1}$ the roughness is max R_a = 0.2.

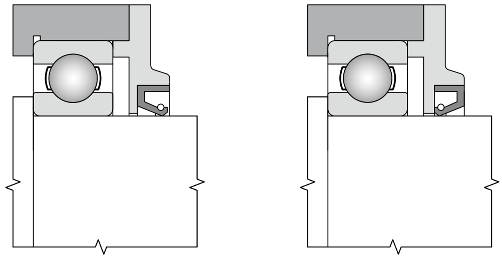

Fig. 8.11

Besides the stated most common sealing rings there are other friction seal designs that utilise specifically shaped sealing rings made of rubber, plastic, etc., or special elastic metal rings. This seal is either selected for locations with high demands on sealing the bearing space (bog contamination of ambient area, high temperature, effect of chemicals), or due to economic reasons in bulk and large lot production. Examples are shown in fig. 8.12.

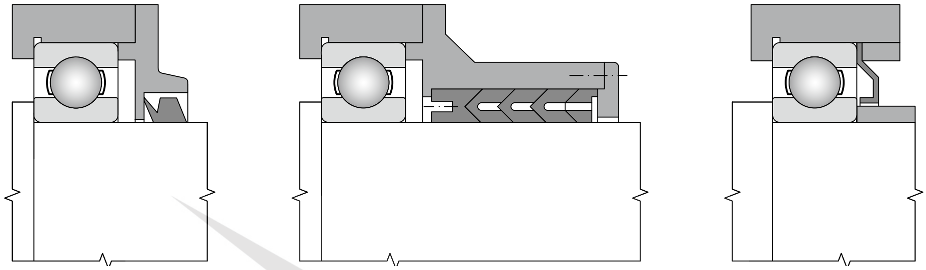

Fig. 8.12

8.3.3 Combined seals

Enhanced sealing effect is achieved by combination of contact-free and friction sealing. Such seals are reco-mmended for humid and contaminated environment. Example is shown in fig. 8.13.

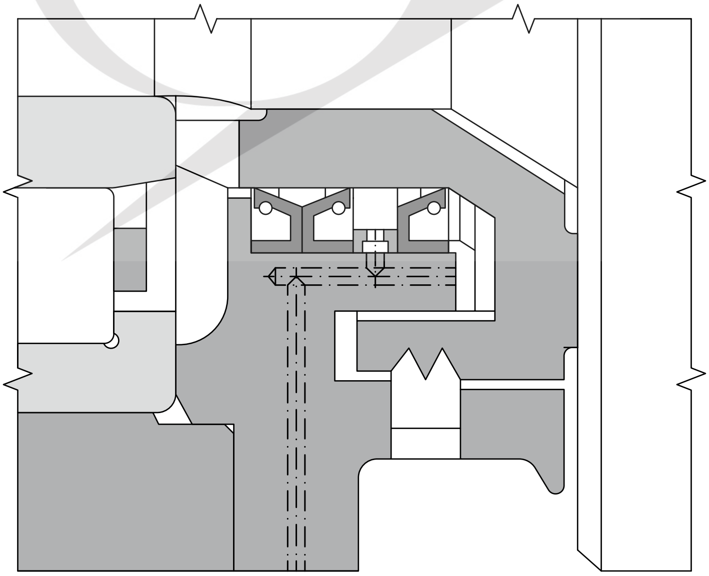

Fig. 8.13

9. BEARING LUBRICATION

The main purpose of lubrication is to reduce friction and wear inside the bearing. Slippage and rolling occur in the contact area between rings and rolling elements. The size of slippage depends on the type of bea-ring used, the load, and mode of lubrication. Elastohydrodynamic lubrication occurs in roller bearings under operating conditions and is characterized by a significant rise in pressure within the lubricating film inside of the contact area.

Main roles of lubricants:

- Decrease friction and wear – direct metal-to-metal contact between bearing rings, rolling elements, and cages is prevented by the use of lubricating film that decreases friction and wear in the contact areas.
- Extend fatigue life – bearing fatigue life depends, in particular, on the viscosity and film thickness of the lubricant between contact surfaces.
- Heat dissipation – oil circulation can dissipate excess frictional heat or heat from the external envi-ronment from the bearing, thereby protecting the bearing against overheating and the oil against degradation.
- Protection of bearing surface against corrosion
- Preventing entry of foreign particles (contaminants) into the bearing, removal of foreign particles from the bearing oil circulation.

9.1 Types of lubrication

Oil or grease are used under normal conditions for bearing lubrication, or in special cases solid lubricant is used, e.g. for extreme temperatures or operation in a vacuum. When deciding on the type and method of lubrication, one must consider the operating conditions, the characteristic properties of applied lubricant, the design of the equipment, and its operating efficiency. Oil lubrication provides better lubrication characteristics, but grease lubricants make for easier use in bearings.

A comparison between oil and grease lubrication is provided in table 9.1.

Table 9.1

grease lubrication	oil lubrication
low temperatures	high and extremely low temperatures
low speeds (65 % to 85 % of revolutions, which can be achieved during oil lubrication)	high rotational speed
protection against entry of contaminants (glands, covers)	oil seals to prevent leakage
long-term maintenance-free operation	bearings are lubricated from a central source, which also serves to lubricate other machine components
weak cooling	heat dissipation via oil circulation
removal of contaminants from grease not possible	easy removal of particles from lubricant using oil filter

9.2 Grease lubrication

Under normal conditions, most of the loadings use grease lubrication. An advantage of grease is that it holds better in the loading, it seals the housing against entry of contaminants, moisture, and water and, in particular, affords easier bearing maintenance.

Fig. 9.1

Bearing grease is most often produced from high-quality mineral or synthetic oils that are thickened using fatty acid metal detergents. Greases need to have good lubricating ability and high chemical, thermal, and mechanical stability. Greases can be enriched with additives that increase the service life of the grease and bearing. When selecting grease, the most important characteristics to consider are the viscosity of the base oil, its consistency, load-bearing ability, and anti-corrosion properties.

9.2.1 Composition of grease lubricants

- Base oil – most frequently a mineral-based or synthetic oil. Lubrication properties of the grease are usually given by the properties of the base oil. Base oil viscosity is the decisive factor when selecting grease. Greases produced from low-viscosity base oil are suitable for high speed and low tempera-ture applications, and lubricants with high-viscosity base oil are preferred for high temperature and heavy load applications.

- Thickening agent – the type of thickening agent, in particular, affects the grease dropping point and determines the application for a particular operating temperature; the higher the dropping point, the higher the temperature resistance of the grease. The maximum operating temperature of gre-ase however is affected by the thermal resistance of the base oil. The water resistance of grease depends solely on the type of thickening agent.

- Additives – greases often contain additives that enhance certain grease characteristics or extend its life. Among the most commonly used are antioxidants (extend life), corrosion inhibitors (improve corrosion resistance), and EP additives (extreme loads).

9.2.2 Basic grease characteristics

- Base oil viscosity – the grease viscosity is given by the base oil; it is the most important factor when selecting a grease and has the most significant effect on the thickness of the lubricating film in the contact area and hence the bearing life. The oil viscosity is defined as the measure of flow resistan-ce during lubricant sheer stress. The viscosity increases exponentially proportionally to the pressure and exponentially decreases proportionally to the temperature.
- Characteristics of captured oil – grease assumes all characteristics of the base oil, such as viscosity, freezing point, and flash point; such characteristics significantly influence the behaviour of grease.
- Consistency – greases are divided into several consistency classes according to the NLGI (National Lubricating Grease Institute) classification. The grease consistency should dramatically change within the temperature range and during mechanical loading. If an unsuitable grease consistency is selec-ted for a given loading, then the grease may leak out of the bearing or may increase the rotation resistance and lead to insufficient oil release in the contact area.

9.2.3 Miscibility

Mixing of greases should generally be avoided. Mixing greases with different types of thickening agents can interfere with the composite and physical characteristics, which can lead to leakage of the lubricant from the bearing and potential bearing failure. Greases manufactured using the same thickener base and similar base oil can generally be mixed without any adverse effects.

An overview of roller bearing grease is provided in table 9.2.

9.2.4 Amount of lubricant

The amount of grease depends on the bearing loading design, the amount of free space, the characteristics of the grease applied, and the operating temperature. An abundant use of grease in the loading causes an increase in operating temperature. Generally, the bearing is filled with grease and the free space in the bea-ring loading is only partially filled. The amount of grease in the free space of the loading can be determined relative to the speed:

- 1/2 to 2/3 free space at speeds below 50% bearing limiting speeds.
- 1/3 up to 1/2 free space at speeds above 50% bearing limiting speeds.

The bearing with grease should be run in, so that the grease can be evenly distributed throughout the bea-ring and so the excess grease can leak out of the bearing; the bearing can then subsequently operating at maximum speeds. When the bearing is properly run in, the bearing temperature decreases and the operating temperature becomes stable.

Bearings operating at very low speeds, as well as the free loading space, should be fully packed with grease to protect the bearing against corrosion and entry of contaminants.

Table 9.2

Grease characteristics for roller bearings				
Grease type		**Characteristics**		
Thickening agent	**Base oil**	**Heat range of use [°C]**	**Water resistance**	**Application**
Lithium soap	mineral	-20 to 130	resistant	multi-purpose lubricant
calcium soap	mineral	-20 to 50	highly resistant	good sealing effect against water
sodium soap	mineral	-20 to 100	non-resistant	emulsifies with water
aluminium soap	mineral	-20 to 70	resistant	good sealing effect against water
lithium complex soap	mineral	-20 to 150	resistant	multi-purpose lubricant
calcium complex soap	mineral	-30 to 130	highly resistant	multi-purpose high temperature, high-load lubricant
sodium complex soap	mineral	-20 to 130	resistant	suitable for high temperatures, high loads
aluminium complex soap	mineral	-20 to 150	resistant	suitable for high temperatures, high loads
barium complex soap	mineral	-30 to 140	resistant	Suitable for high temperatures and loads
bentonite	mineral	-20 to 150	resistant	suitable for high temperatures and low speeds
polycarbamide	mineral	-20 to 160	resistant	suitable for high temperatures and medium speeds
lithium soap	silicone	-40 to 170	highly resistant	suitable for wide temperature ranges and medium rotational speeds
barium complex soap	ester	-60 to 140	resistant	suitable for high temperatures and high speeds

9.2.5 Re-lubrication

Bearings must be re-lubricated if the expected bearing life is longer than the uptime of the applied grease. The re-lubrication interval is significantly influenced by the type and size of the bearing, the operating speed and temperature, and by the type and quality of grease.

The re-lubrication interval is the period during which the grease possesses the required lubricating characteristics. After this period elapses, the bearing must be re-lubricated after thoroughly first removing the old grease from the bearing space. The recommended re-lubrication intervals for individual types of bearings under normal load ($P \leqslant 0.15$ C) and normal operating conditions is provided in the diagrams on figures 9.2 and 9.3. The diagrams apply for common greases for temperatures up to +70 °C. At temperatures above +70 °C, the re-lubrication intervals are reduced to one-half their original values for every increase of 15 °C. At temperatures below 40 °C, the re-lubrication intervals may be increased two-fold.

For small, in particular single-row ball bearings, the re-lubrication intervals are several-fold greater than the expected bearing low; consequently, such bearings are generally not re-lubricated. For the reason specified above, it is preferable to use such bearings designed with shields or with seals on both sides, which are filled with grease at the factory and which never require re-lubrication. After certain speeds, the re-lubrication period falls outside of the curve on the diagram; this means that the permissible grease lubrication threshold has been exceeded. In such cases, we recommend that the loading be designed for oil lubrication.

The grease should be re-filled whenever the re-lubrication interval is longer than 6 months. The re-lubrication intervals may be greater when using extreme performance grease. More information will be provided by the Dunlop BTL Technical and Consultation Services Department.

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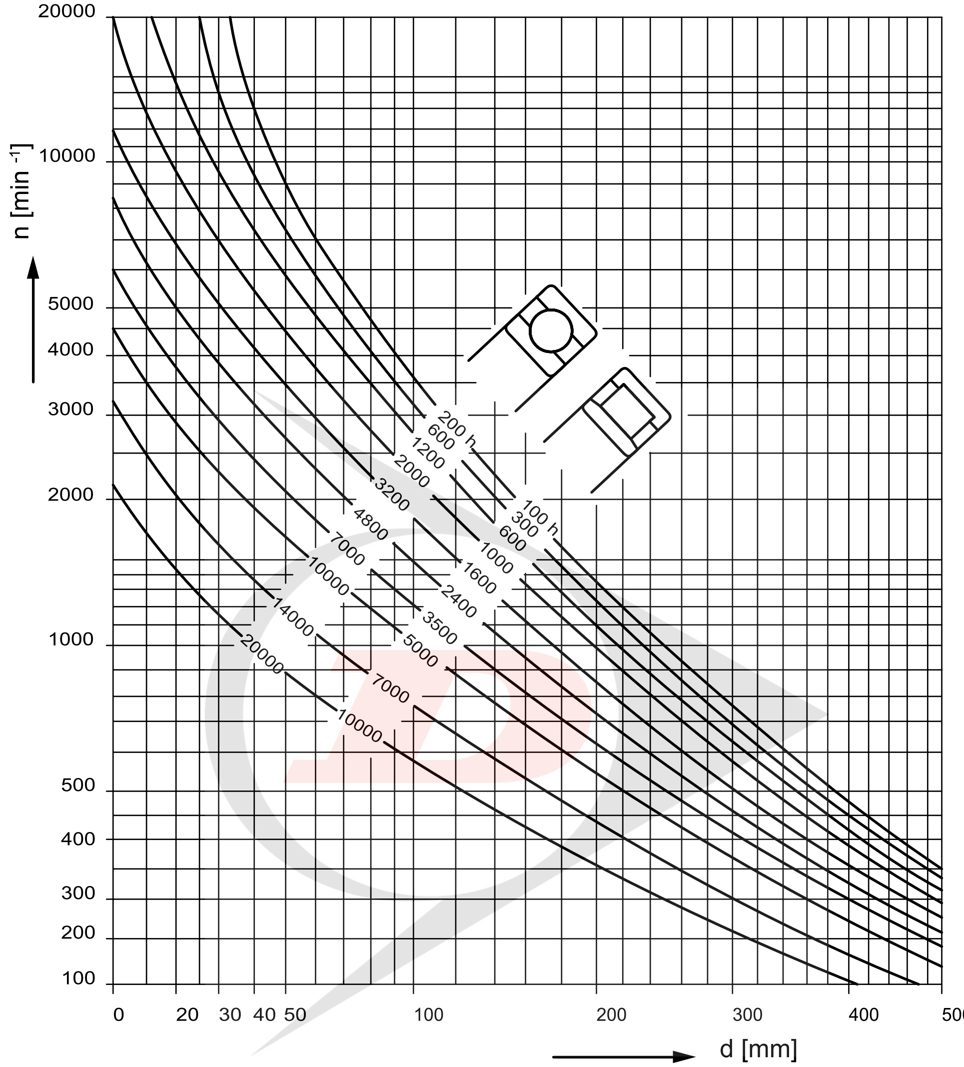

Fig. 9.2

The amount of grease required for re-lubrication can be calculated from the equation

$$Q = 0.005 \cdot D \cdot B \quad [g]$$

Q . . . . . . quantity of grease [g]

D . . . . . . outer bearing diameter [mm]

B . . . . . . bearing width [mm]

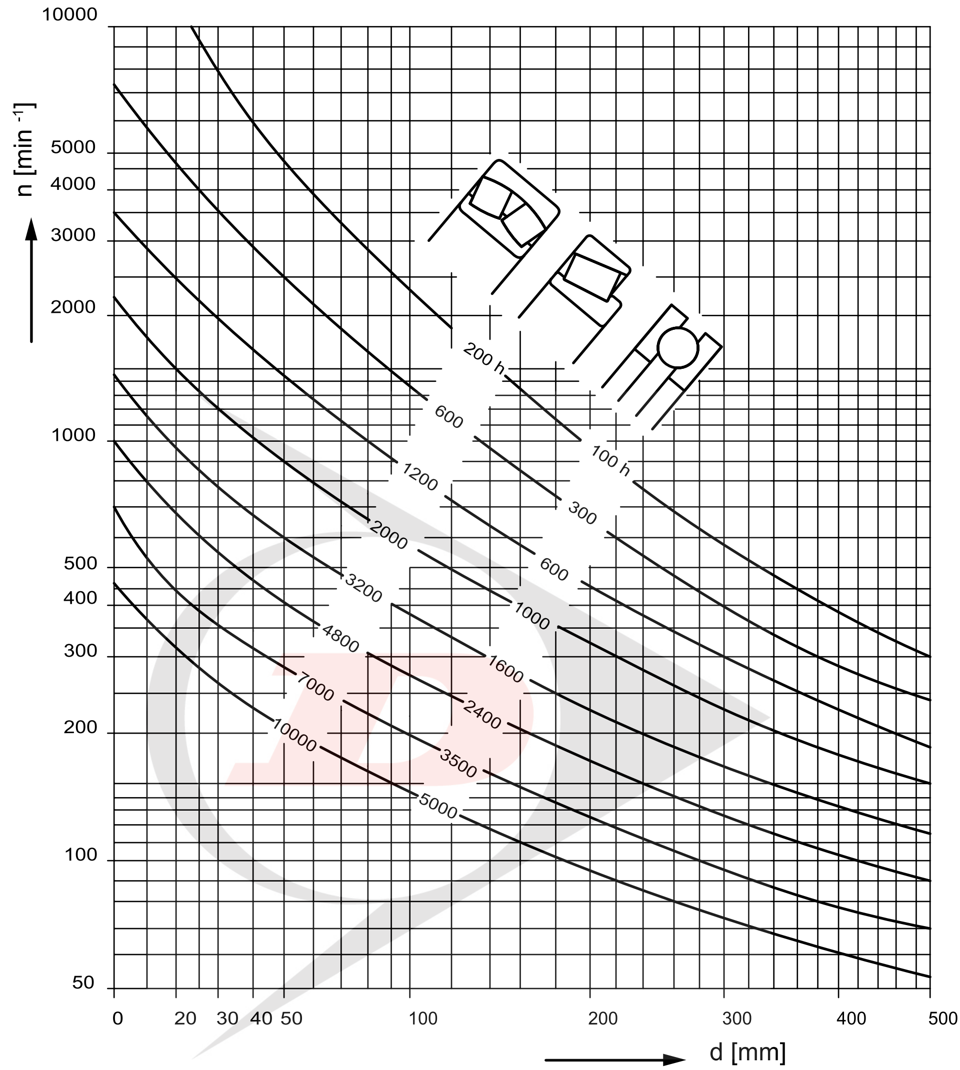

Fig. 9.3

Used grease should first be removed from the bearing space in high-speed bearings, requiring more frequent re-lubrication. This helps to prevent any undesired rise in operating temperatures. A grease slinger can be used to prevent bearing over-lubrication. It comprises a plate, which rotates on a shaft and the centrifugal force pushes out any excess and degraded grease through the slot in the housing out of the bearing (fig. 9.4).

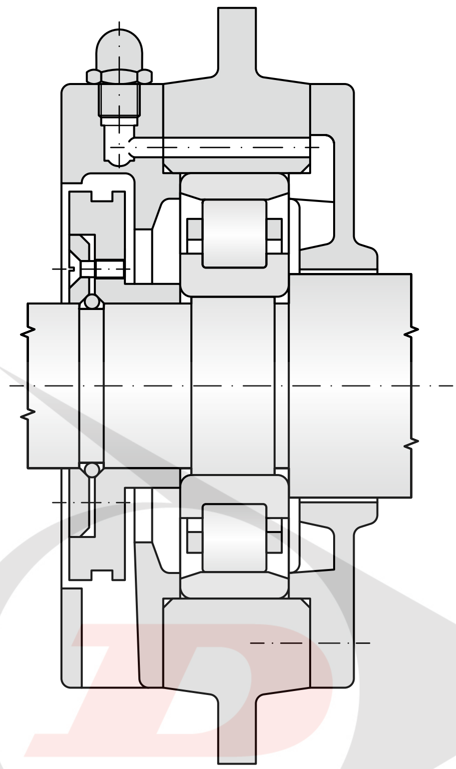

Fig. 9.4

Other factors affecting the re-lubrication interval:

- Vibrations – kneading of grease occurs during large vibrations and shocks, requiring more frequent re-lubrication. If grease becomes soft, grease with higher mechanical stability or stiffer grease must be used.
- Vertical shaft – the re-lubrication interval must be shortened by half and requires the use of glands and covers that prevent the leakage of grease from the loading.
- Contamination of grease – the re-lubrication intervals must be shortened, when the grease contains undesired particles, which can have a negative effect on the bearing life.

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9.3 Oil lubrication

Oil lubrication is used when the rotation speeds are so high that the grease re-lubrication periods are too short. Another reason may be the need to dissipate heat from the bearing or when the temperature of the environment is high, which prevents the use of grease or if adjacent components already use an oil lubrication design (e.g. gearbox gears). With the exception of select spherical-roller bearings, such loadings are always lubricated with oil.

The use of oil lubrication necessitates that lubrication during running in and afterwards, during operation, be ensured. Excessive use of oil increases the oil temperature and thus the bearing temperature. The oil supply to the bearing is secured using various design methods:

- Oil bath lubrication – the most popular and simplest method of oil lubrication for low and medium rotational speeds. The oil level extends to the centre of the bottom rolling element and must be maintained at this level. The oil is carried by the rotating components of the bearing and dispersed in the bearing to return to the oil bath.

- Circulating oil lubrication – used most often in high speed applications, where the bearing needs to be cooled and for high temperature applications. Oiling is achieved by a pump. After the oil passes through the bearing, the oil is fed back into the sump, re-filtered, and cooled, as needed.

- Drop lubrication – is widely used for lubrication small ball bearings used in high speed applications.

- Oil splash lubrication – oil is splashed on the bearing by a rotating gear wheel or by a simple rotor adjacent to the bearing. The bearing does not need to be immersed in the oil bath; this method of lubrication is often used in automobile transmissions.

- Oil injection lubrication – generally used for high-speed bearings. Oil is injected under pressure directly into the bearing. The oil jet velocity must be sufficiently high to ensure that the oil penetra-tes through the swirling air created by the rotating parts of the bearing.

- Oil mist lubrication – injects an oil mist into the bearing. This method of lubrication is often used for lubricating spindle bearings of machining centres.

- Oil-air lubrication system – compressed air is used to supply a very small, precise amount of oil into each bearing to ensure sufficient lubrication and to better achieve lower operating temperatures and higher speeds. This lubrication method is used for lubrication most spindle bearings and for other high-speed applications.

9.3.1 Oil lubricants

Refined oils, with good chemical stability, are generally used for lubricating bearings. Stability can be impro-ved by the use of antioxidant additives. Mineral oil without additives is generally preferred for lubricating roller bearing; additives are used only in special circumstances. Synthetic oils are intended solely for demanding applications at extreme temperatures (high or low).

Certain types of bearings, e.g. spherical-roller bearings, spherical-roller thrust bearings, or tapered roller bearings usually achieve higher operating temperatures than other types such as, e.g. ball bearings or roller bearings under identical operating conditions. This must also be considered when selecting the type of oil.

The decisive characteristic of oil is its kinematic viscosity, which decreases as the temperature increases. We can determine the appropriate oil viscosity from the diagram on fig. 5.4 in relation to the mean bearing diameter d_s = (d+D)/2 and the rotating speed.

If the operating temperature is known or can be identified, a suitable oil and viscosity on fig. 5.5 can be determined using the internationally standardized reference temperature of 40 °C, required for calculating the χ ratio. Figures can be found in chapter 5 Determining the bearing size.

The use of oil with EP additives is recommended when the χ ratio < 1, since they increase the oil film bearing capacity. Oil with EP additives must always be used, whenever the χ value falls below 0.4. Improved reliability of the respective loading design is achieved if χ > 1.

Example:

- bearing: d = 180 mm, D = 320 mm, d_s = 250 mm
- rotation speed n= 500 min^{-1}
- expected operating temperature 60 °C

According to the diagram on fig. 5.4, the minimum kinematic viscosity required to meet these conditions is

ν_1 = 17 mm^2s^{-1}.

Adjusting for an operating temperature of 60 °C, the applied oil, selected according to the diagram on fig. 24 at a standardized temperature of 40 °C, must have a minimum kinematic viscosity of 35 mm^2s^{-1}.

The kinematic viscosity of lubricating oil for spherical-roller thrust bearings is estimated according to table 9.3 relative to the product n*d, where n is the bearing rotation speed in revolutions per minute and d is the bore diameter in mm. Lower viscosity values apply for low-load bearings, for which the relationship $P_a \leq 0.1\ C_a$ applies. Higher values apply for $P_a > 0.1\ C_a$.

Table 9.3

Oil viscosity for spherical-roller thrust bearings	
d*n	kinematic viscosity of oil [mm^2s^{-1} at 40°C]
1 000	250 to 550
10 000	100 to 250
100 000	45 to 100
200 000	30 to 80

9.3.2 Changing oil

The oil change interval depends on operating conditions and the oil quality used. If the operating temperature is less than 50 °C and the oil works in good operating conditions with and in a low dust environment, the oil is regularly changed once annually. If the oil temperature ranges near 100 °C, the oil must be changed approximately once every three months. The more demanding the operating conditions, the more frequent the oil changes to ensure lubricant purity and adequate state of oxidation. The use of specialized types of oils for specific operating conditions may significantly extend their uptime.

9.4 Lubrication using solid lubricants

Sold lubricants are used for lubricating bearings only in cases, when grease or oil are unable to meet the demands for reliable lubrication under limiting friction conditions or when required to provide adequate resi-stance against high operating temperatures, chemicals, and similar other effects.

Graphite, MoS_2, and PTFE, in particular, are used for bearing lubrication. The lubricating mechanism is given by the lattice structure of compounds; the layers of particles easily slide along each other and adhere well to a metal surface, which prevents the displacement of lubricant particles during sliding or rolling motions.

Drawbacks of solid lubricants:

- High coefficient of friction
- Inability to act as a coolant
- Limited uptime
- Low dampening of vibrational instability of rolling elements and cage

10. ASEMBLY AND DISASSEMBLY OF BEARINGS

10.1 General information

Roller bearings are strongly stressed machine components parts of which feature high accuracy. To be able to utilise fully functional properties of bearings and avoid damaging them before the end of their service life, assembly and potentially disassembly procedures have to be correctly specified. To do so, the structure of location has to be well known, suitable workplace and assembly tools made available to simplify the assembly and disassembly of bearings. It is very important that the assembly is performed by workers who are properly qualified and equipped with protective equipment.

10.2 Assembly worksite

Worksite must be equipped with suitable assembly tools and jigs to make the work comfortable and also safe. Equipment varies by the type and size of bearings to be assembled at the worksite. Very important is to make sure that these tools are clean and the work is performed in a clean working environment. In negative sense, impurities have decisive impact on the run of bearing when it is in service. Depending on the size and origin of impurities they may cause increased noise level of bearing and may also cause a bearing failure. The same conditions of cleanliness have to be applied in the preparation of all lubricating agents and com-ponents associated with location. Assembly worksite has to be therefore separated from normal production and only reserved for assembly of bearings. The worksite must be sufficiently spacious, dry and dust-free. No adjustments of components are supposed to be performed there, such as polishing, drilling or welding that could cause impurities to penetrate into the location area, or no air compressing devices shall be used in the proximity. The worksite shall not be exposed to weather effect since bearings are very sensitive to humidity, especially after being washed off preservative agents or old lubricant.

10.3 Work procedures

Prior to the commencement of every assembly the work procedure has to be specified based on drawing documentation to define individual work steps. In special cases that differ from common practice, detailed assembly instructions have to provided, containing all assembly details, such as specification of needed work tools and equipment for assembly and disassembly, measuring instruments, special tooling, way of heating the bearings up, type and amount of lubrication, etc.

10.4 Preparation of bearings for assembly

Prior to the assembly, the fitter has to make sure whether the designation stated on the bearing corresponds with that on the bearing packaging stated on the drawing. The fitter should have basic knowledge of roller bearing identification system.

Dunlop BTL bearings are in original packaging protected with a preservative agent against corrosion for a period of 5 years on condition of proper storage. In order to maintain cleanliness, bearings are taken out of the packaging just before the assembly. Only in exceptional cases the bearing is cleared of preservative agent. Damaged packaging indicated potential contamination of bearings during the storage; so the bearings always have to be washed out prior to the assembly. Various cleaning agents can be used to wash out bearings – organic or inorganic. One can use e.g. benzine with 5 to 10 % addition of oil, petroleum, alcohol or dehydrating fluids. Majority of these agents are flammables – this have to be borne in mind. An alternative are alkaline cleaning agents but these are caustic substances.

Bearings are washed out in a clean suitable tank using a brush or a fibre-free cloth. During the washout one of the bearing rings have to be rotated with. If one bath is not sufficient to wash out the bearing, multiple baths are used depending on the level of contamination. After the washout, the bearing has to be provided with protective oil or grease layer depending on the type of lubrication to be used in run. During preservation, one ring of the bearing is slowly rotated with so that the raceways of both rings as well as the surface of rolling elements come to contact with the preservative agent.

After preservation, the bearing has to be protected from contamination and mounted to respective place as soon as possible. The anticorrosive agent that is used for preservation of Dunlop BTL bearings is compatible with majority of commonly used greases and need not be removed before the assembly. It is only recommended to wipe the surface and hole of the bearing to ensure proper location of the bearing.

No additional mechanical adjustment shall be done on roller bearings, such as making bores for supply of lubricant, slots, recesses, etc., since this might release tension in the rings that would cause early damage to the bearing. Besides that, there is a risk that the bearing can be contaminated with splinters or abrasion dust.

When handling bearings one needs to use gloves and lifting equipment to simplify the operation and enhance work safety. If you need to lift bearings in vertical position, we recommend to suspend them on a steel belt or strap on the outer ring circumference and not in one spot only. To lift bearings in horizontal position we recommend that big bearings are, upon a special request, provided with tapped bores for lifting lugs that will simplify subsequent handling. Suspension screws must however be loaded exclusively in the direction of the shank axis.

10.5 Preparation of location components for assembly

Prior to the assembly, all located parts must be thoroughly clean and cleared of burrs caused during their machining. Unmachined surfaces of the inside of rolling location bodies must be perfectly clean and cleared of the moulding sand residues, and provided with a protective coating. Also, all lubrication holes and threads have to be cleared thoroughly. All sharp edges need to be bevelled.

Prior to the assembly itself you need to check that the defined tolerances, geometrical accuracy and quality of bearing saddle surface and that in the body have been met. The accuracy of rolling bearings' dimensions need not checked prior to the assembly.

To ensure reliable operation of bearings, bearings must not be mounted on shafts which do not guarantee the accuracy of geometrical shape, on bended shafts or on shafts with mechanical damage. Therefore the shaft has to be checked carefully prior to the assembly. Depending on the size of the shaft, the accuracy of shape in tips can be checked on the lathe (fig. 10.1) or in supports by means of pointer indicator or micrometer.

Cylindrical journal can be checked using a snap gauge or micrometer in two planes perpendicular towards the journal axis. Two measurements are to be performed in each plane (fig. 10.2).

Additionally, the fitting and fillet of transit on the shaft have to be checked. It is very important that the perpendicularity of fitting the frontals towards to cylindrical seating surface axis for bearings was as accurate as possible. Bearing ring must seat with the entire surface on the front surfaces of the support. Major devia-tions of frontal surface perpendicularity cause additional tensions in bearings and ring deformation when the rings are pressed on and in axial load. At higher revolution frequencies these strains negatively affect the run of the bearing. The method of measuring the perpendicularity of the fitting forefront is indicated in fig. 10.3.

Tapered journals are checked by taper gauge (mostly taper 1:12) which has to be seated on the entire surface.

Bearing bodies are checked in the same way as journals (fig. 10.4) using an internal micrometer or a gauge. We also check the concentricity of seating surfaces in the body, especially if ball and roller bearings are mounted. Split cases have to be checked for not forming a bore on the body after tightening the conne-cting screws which would result in undesirable gripping and deformation of the outer ring of the bearing.

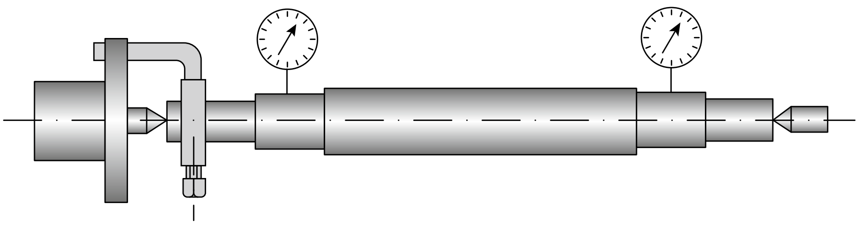

Fig. 10.1

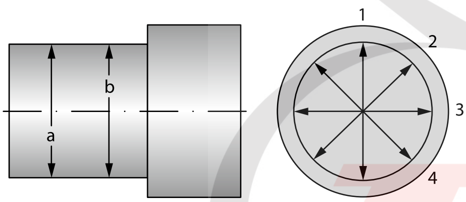

Fig. 10.2

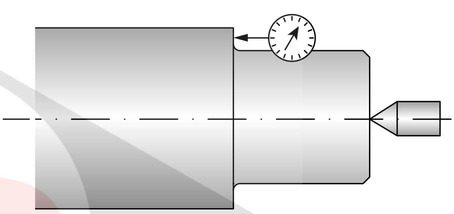

Fig. 10.3

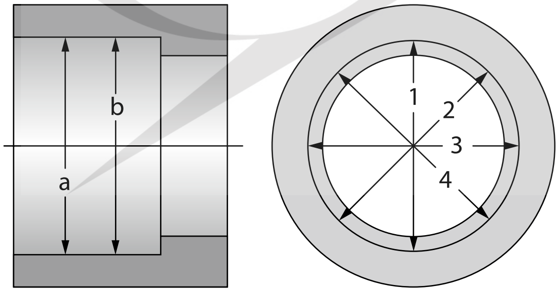

Fig. 10.4

It is recommended to record the results of measuring. During the measuring it has to be made sure that the measured parts and measuring instruments have approximately the same temperature. This is of special importance when big and heavy bearings and related parts are measured.

We also recommend that prior to the assembly the locations of bearings are provided with mounting lubricant. Mounting lubricant can be used for any fixed and sliding locations. It simplifies the assembly itself, prevents occurrence of joint corrosion and makes easier subsequent disassembly of the bearing from location.

10.6 Assembly of bearings with cylindrical bore

Different types and sizes of roller bearings require different assembly procedure. In principle, direct hammer strokes on the ring flanges, on cages or rolling elements have to be avoided during the assembly. When assembling non-separable bearings, the mounting force must act on the ring located with overlap that is mounted as first. In no case shall the mounting force be transferred via the rolling elements of the bearings. Thus the bearing is firstly mounted on journal by loading via the inner ring and then the entire bearing is pushed in the body where the location is usually sliding (fig. 10.5). If a non-separable bearing with overlap on shaft and in body is mounted, the mounting force must act on both rings equally (fig. 10.6). Rings of separable bearings can be assembled separately.

Bearings are mounted in location units either cold or heated.

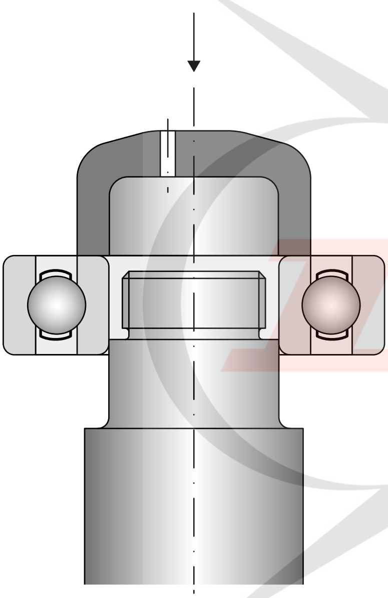

Fig. 10.5

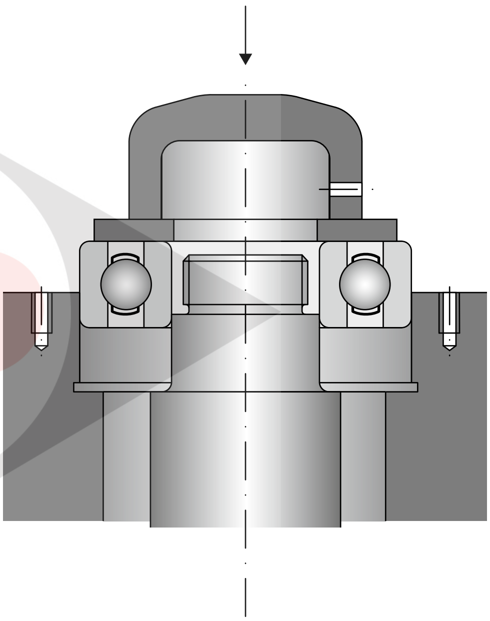

Fig. 10.6

10.6.1 Cold assembly

Bearings with smaller dimensions, up to bore diameter of 80 mm can be cold mounted with common overlap. The force needed for assembly is achieved by means of press. Pressing is recommended to be performed using assembly jigs. If no press is available, smaller bearings can be mounted by means of light hammer strokes via the mounting sleeve leaned on the pressed ring. Hydraulic nuts can also be conveniently used in cold assembly.

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10.6.2 Hot assembly

Hot assembly is used for bigger bearings rings of which are usually located with higher overlap. During the process, inner rings, alternatively entire bearings and bodies in which the bearings are mounted, are heated up prior to the assembly. We recommend that the assembly procedure with bearing temperature heat-up above 100 °C is discussed with the workers of the Dunlop BTL technical and consultancy services.

To ensure fast, safe and clean heat-up of bearings it is recommended to use induction heating equipment to ensure uniform heating of bearings without the risk of local overheating. Individual bearings of smaller dimensions can be heated on electrical hot plate with thermostatic control. Bearings have to be turned seve-ral times during the heating. Medium sized bearings can be heated by hot air reheat case with thermostatic control. The time of heating is however relatively long.

Roller bearings of all types and sizes can be heated in oil bath (fig. 10.7). This way of heating does not suit heating of sealed bearings, bearings with plastic lubricant filler and accurate bearings. Oil filler should be provided with thermostatic control (temperature between 80 and 100 °C) but usual heating is 50 to 60 °C above ambient temperature, i.e. oil is heated up to 70 to 80 °C. In the bath, bearings have to be placed on a grid or suspended in the bath to avoid their direct contact with the heated surface which might lead to overheating. Heating in oil bath however has a number of disadvantages, mostly the risk of injury, pollutant load with oil vapours, risk of hot oil inflammation and risk of bearing contamination.

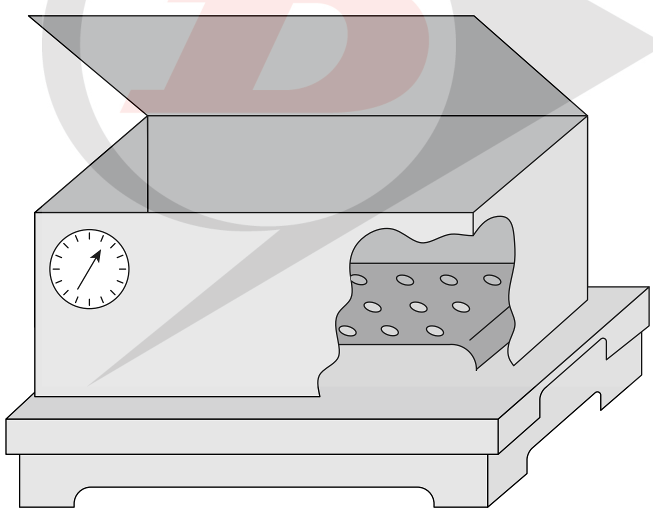

Fig. 10.7

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10.7 Assembly of bearings with tapered bore

Bearings with tapered bore are mounted on shaft by means of adapter or withdrawal sleeves, or alternatively directly on tapered journal. Inner ring with tapered bore is always placed on the journal firmer than a ring with cylindrical bore. Fixed location is achieved either by pressing the inner ring on by means of a nut or a tape-red sleeve. In both cases the inner ring will expand and cause reduction of radial clearance in the bearing. Therefore a method has to be determined that would correctly specify the overlap. This can be achieved by measuring the radial clearance reduction using a feeler gauge. The clearance before and after assembly must be measured between the inner ring and unloaded rolling element. This method suits to medium size and big spherical roller bearings. Other methods are e.g. measuring of the lock nut torque angle or measuring of axial displacement of the inner ring on the tapered journal. In the assembly of double row self-aligning ball bearings, the adapter sleeve nut can be tightened to such extent that the inner ring can be smoothly turned and tilted. The assembly method should be consulted with the manufacturer.

Reliability of fixation of spherical-roller bearings can be checked by measuring of axial displacement of the inner ring on the journal or tapered sleeve. The initial position for measuring of this displacement will be achieved when the contact surfaces (of the ring, sleeve, shaft) abut against each other on the entire bearing surface. The values of axial displacement for the assembly of double row spherical-roller bearings with tapered bore are stated in chart 5, chapter Spherical roller bearings.

Small bearings of bore diameter up to 80 mm can be pressed on a tapered journal, adapted sleeve (fig. 10.8) or the withdrawal sleeve (fig. 10.9) by means of terminal nut that is tightened by a mounting spanner. Prior to the assembly, the contact surfaces have to be coated by oil.

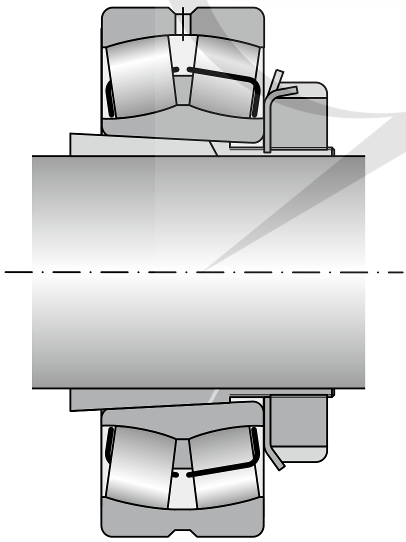

Fig. 10.8

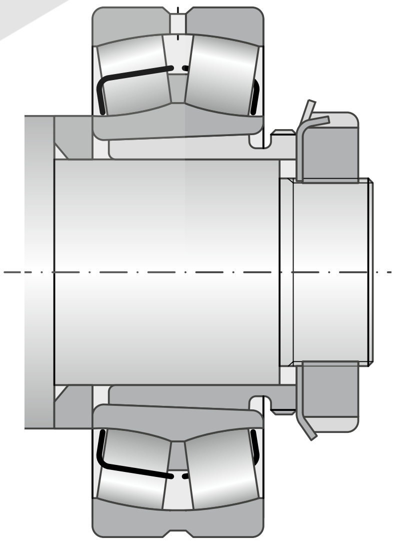

Fig. 10.9

Bigger bearings require much bigger mounting force, and that's why hydraulic nut or pressure oil method should be applied in their assembly, when oil is brought between the contact surfaces of the ring and journal under high pressure (fig. 10.10). This creates an oil film that reduces friction between the bearing surfaces. This method can be used also for the assembly onto adapter sleeves or withdrawal sleeves that are modified to suit this method. Use of oil of 75 mm^2/s viscosity at 20 °C is recommended for the assembly (nominal viscosity at 40 °C is 32 mm^2/s).

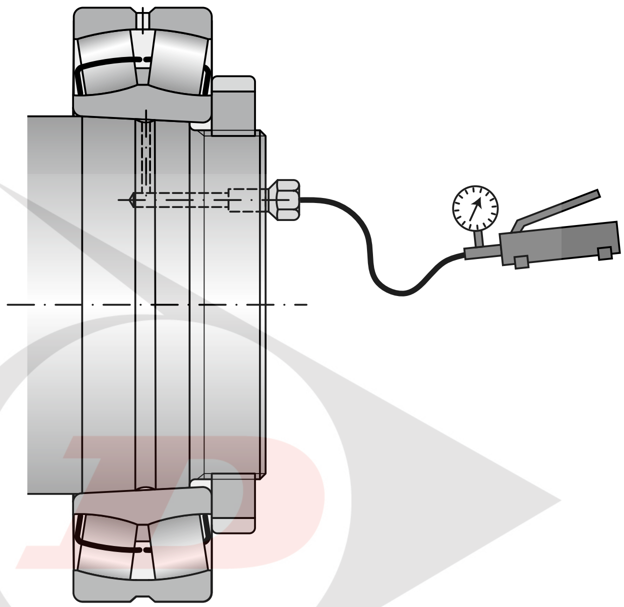

Fig. 10.10

10.8 Disassembly of bearings with cylindrical bore

If bearings and related parts are to be reused, the disassembly has to be paid particular attention. Non-separable bearing is always dismantled by force acting on the ring located with overlap. In separable bearings they are dismantled one by one, analogically with the assembly of these bearings.

For disassembly of smaller bearings mechanical pullers or hydraulic presses (fig. 10.1) should be used. The disassembly can be simplified by means of a groove on the shaft or in the body that will allow engagement of the puller on the ring mounted with overlap. To dismantle inner rings of heat mounted cylindrical roller bearings one should use induction tools.

To dismantle bearings with fixed location on cylindrical journal also the procedure using pressure oil can be applied (fig. 10.12). This method significantly simplifies the disassembly in cases when big pulling force would have to be applied. The use of this method requires provision of a location with canals and distribution grooves for supply of pressure oil in the bearing inner ring location. The supplied oil significantly reduced the force necessary for bearing disassembly that has to be performed with the help of suitable dismantling equipment, even if this method is applied. Once the oil separates the surfaces of the bearing location which becomes obvious when the oil starts infiltrating, we will pull the bearing down rapidly, without an interruption. If the bearing blocks once the oil canal on the shaft gets partially uncovered, we either have to heat it, or pull down applying considerable force by means of hydraulic tooling.

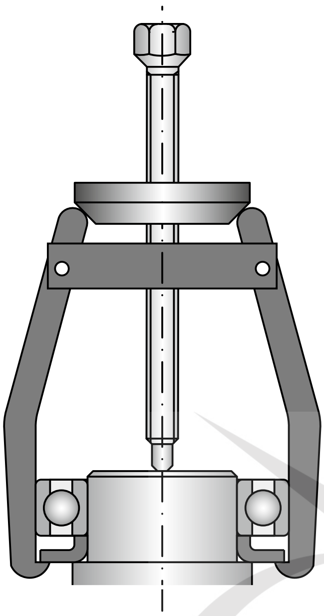

Fig. 10.11

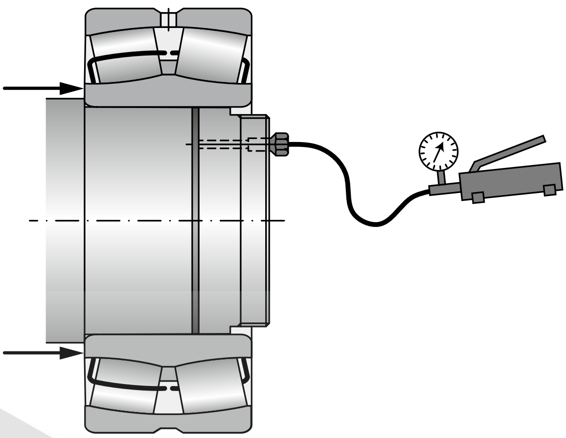

Fig. 10.12

To disassembly inner rings of cylindrical roller bearings without guide flanges or with one guide flange also heating rings can be used (the so-called thermo rings). These are tools made of light alloy, provided with radial grooves (fig. 10.13). This alternative is a cheaper option to induction equipment, mainly for dismant-ling of bearings with bore diameter exceeding 400 mm, or bearings that are dismantled only sometimes. A thermo ring is heated on an electrical hot place to the temperature of 280 °C approximately, slipped over a dismantled bearing ring and clamped in grips. After pulling the cylindrical roller bearing inner ring off the journal, the ring has to be taken off the thermo ring immediately to prevent if from overheating.

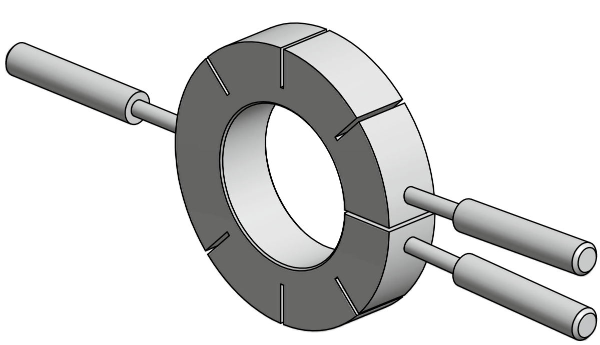

Fig. 10.13

10.9 Disassembly of bearings with tapered bore

If a bearing is mounted on a tapered journal or on a adapter sleeve, first the lock of the terminal nut or the adapter nut has to be removed. The nut is loosened by the distance necessary to release the bearing. Alternatively, another holdback can be used. After pulled down from the tapered journal the bearing will release at a swoop, and without this holdback there is a risk that the bearing will fall down of the shaft.

Disassembly of small and medium size bearings off tapered journal proceeds often by means of pullers that are fastened by the inner ring of the bearing or a support part, such as labyrinth ring. Already when designing the location the layout of suitable bores or grooves for puller arms should be considered. Inner rings of small bearings can be pulled down by means of press or hammer and spine. If press is used, the adapter sleeve has to be leaned and force applied on the inner bearing ring (fig. 10.14).

Bearings that are fastened by means of a adapter sleeve are dismantled by means of terminal nut (fig. 10.15). When big bearings are dismantled and therefore bigger force has to be applied, trust screws guided by nut can be used (fig. 10.16). A washer has to be put between the inner bearing ring and the screws in order to prevent damage of the bearing. Very fast, simple and economic is disassembly of a withdrawal sleeve by means of hydraulic nut. If the bearing is on the edge of the journal, it is recommended that the hydraulic nut is before the disassembly locked with a jig fastened e.g. to the front of the shaft (fig. 10.17).

Big withdrawal sleeves usually have canals and grooves for pressure oil. Oil is thus supplied directly by the pulling nut between the shaft and the sleeve, and between the sleeve and the bearing (fig. 10.18). After pressurised, contact surfaces can be shifted against each other without a risk of damage. Needed pressure is achieved by oil injectors. For the disassembly oil with low viscosity is used, approx. 150 mm^2/s at 20°C is required (nominal viscosity at 40°C is 46 mm^2/s).

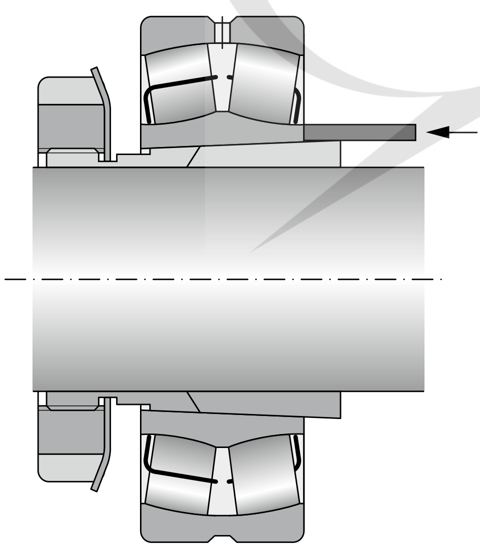

Fig. 10.14

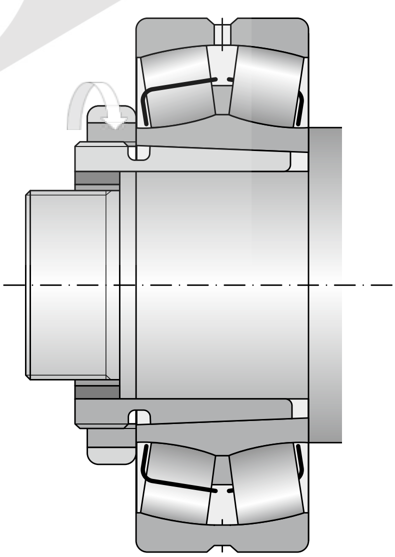

Fig. 10.15

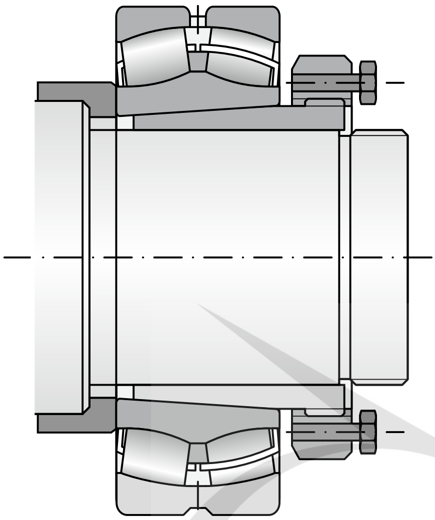

Fig. 10.16

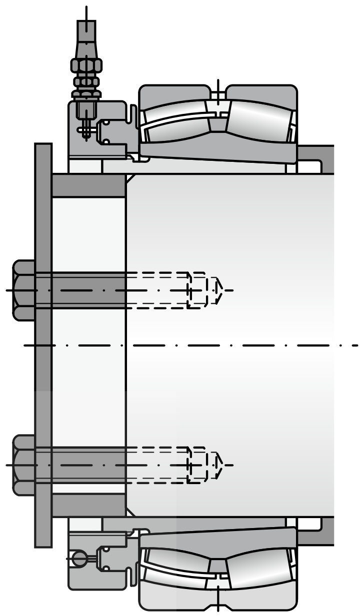

Fig. 10.17

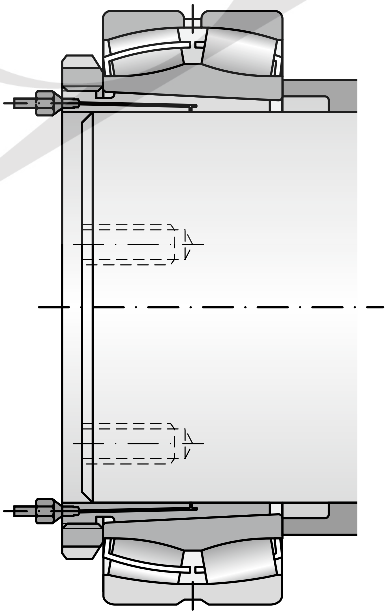

Fig. 10.18

10.10 Storage of bearings

Dunlop BTL bearings are stored and packed in a way that maintains the properties of the bearings as long as possible. The prerequisite for achieving these goals is achieving the conditions for storage of bearings and handling them.

Relative air humidity in the storage should not exceed 60%, and there should be no major temperature oscillations. Most convenient temperature range for storage of bearing is between 15 and 25°C.

Bearings should not be exposed to vibrations and shocks. When stored, bearings must not be exposed to aggressive media, such as gases, fog or aerosols of acids, lyes and salts. Also the effect of direct sunlight has to be prevented since it may cause major temperature oscillations in the container. Big bearings, especially those of light series, must not be stored as standing. They should be placed horizontally to avoid deformation of rings. Bearings must not be stored in racks made of fresh timber or on a stone floor. Bearings must not be placed in the proximity of heating or water piping.

10.10.1 Storage period

If preserved in usual manner, bearings can be stored up to five years as long as the above specified conditi-ons are met. Otherwise shorter storage term has to be counted with.

If the admissible storage terms are exceeded we recommended that bearings are checked in terms of preservation and corrosion.

If possible, both-side shielded (2Z) or sealed (2RS) bearings should not be stored until the end of the storage term. During the storage, grease filler may get old due to chemical and physical processes. Bearings can be functional but the lubricant may be useless. The recommended time of storage of bearings with grease is two years.

11. BEARING DEFECTS AND DAMAGE

Just as other mechanical components, roller bearings can also undergo premature failure or housing defects for various reasons. One must differentiate bearing durability determined by loan fatigue during operating speeds and bearing service life, which determines the bearing uptime, before a bearing is for various reasons decommissioned.

Durability and the systemic approach to calculating fatigue damage is described in chapter 5. Determining bearing size. Bearing durability is affected, e.g. by improper installation, poor selection of bearings, produc-tion errors when manufacturing connecting parts, handling of bearings by unqualified personnel, the entry of contaminants into bearings, or improper lubrication. If bearings show signs of damage or other deficiencies, the cause of such damage must be determined to enable the adoption of measures that would prevent their recurrence.

This often involves more than a simple analysis, especially if there are several concomitant factors or if the damage is so extensive that the initial site of damage cannot be ascertained. Incipient damage is usually demonstrates during operation by increased vibration, temperature, or noise. Sophisticated housing designs should thus be monitored during operation with diagnostic systems and the equipment should be shut down in the initial stages of damage.

11.1 Main types of damage

Examples of main types of roller bearing damage are illustrated in the following figures.

Flaking of the surface

Unacceptable tearing off of material due to thermal overloading of the bearing is shown in fig. 11.1 and 11.2.

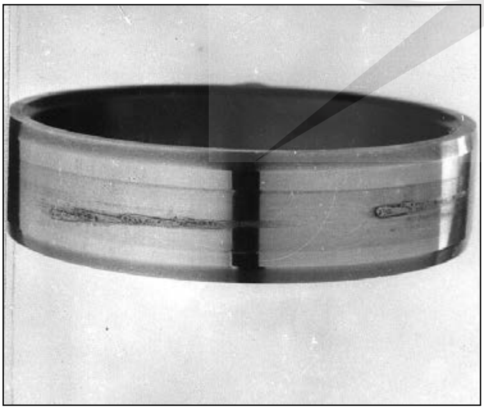

Fig. 11.1

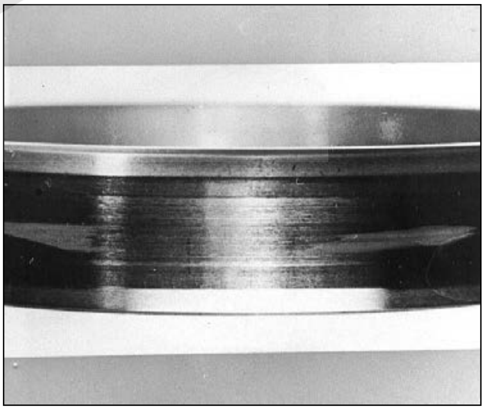

Fig. 11.2

Typical fatigue effect – pitting, which forms on the bearing rings, is shown on fig. 11.3 and 11.4. This damage is the result of cyclical loading of bearing components and is caused by normal fatigue of the material. The first cracks emanate from miniature non-homogeneities in the material at a particular depth below the surface. They are often, however, caused by overloading, insufficient lubrication, or other operating influences. Their timely identification can better help analyse and eliminate the cause. The figures illustrate unacceptable wear.

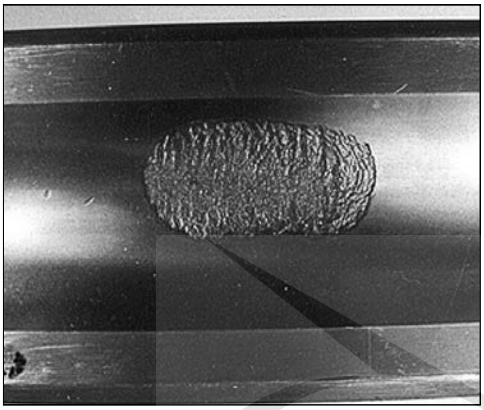

Fig. 11.3

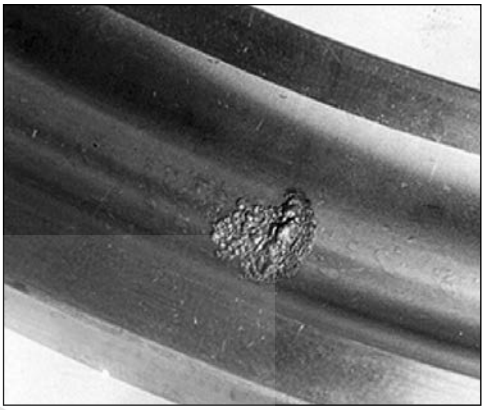

Fig. 11.4

Depressions and pressure damage

Damage to bearing rings caused by indelicate installation (fig. 11.5) and shallow depressions in the race-way caused by beading of solid impurities during bearing operation (fig. 11.6). The extent of damage in both illustrated cases is unacceptable and may form the initial site of progressive fatigue damage – pitting. Damage to raceway caused by improper installation are usually easily discernible because they are located within the pitch of the roller elements. Pressure damage caused by stationary overloading or by equipment vibrations when transporting over long distances , e.g. during shipping, also present a danger.

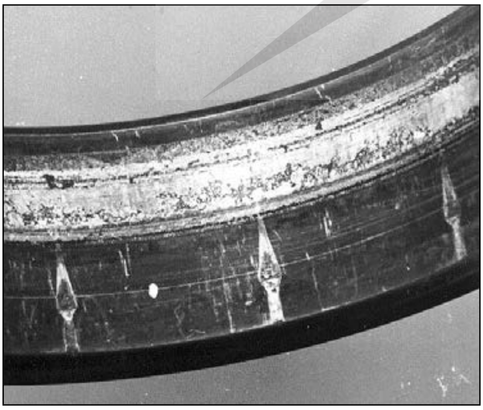

Fig. 11.5

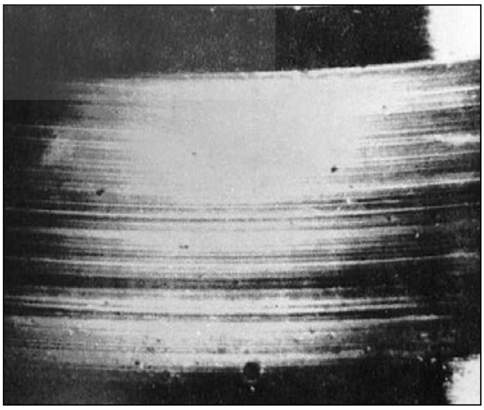

Fig. 11.6

Abrasion

Ball glazing due to overloading and lubrication failures (fig. 11.7) and abrasion of the race due to spinning within the seat (fig. 11.8). The condition in both cases is unacceptable.

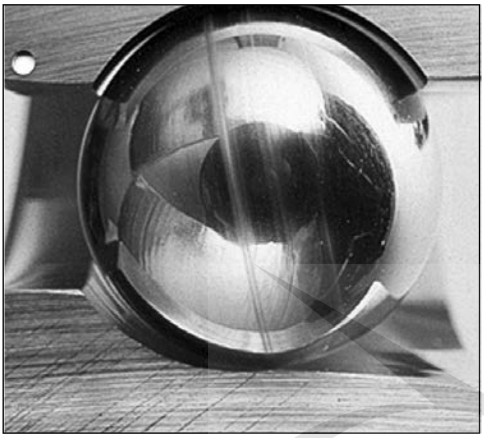

Fig. 11.7

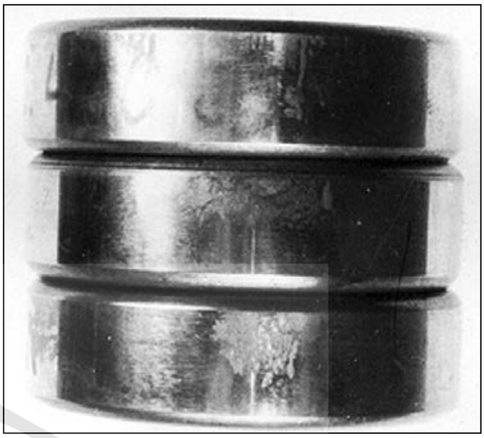

Fig. 11.8

The formation of grooves and craters due to the passage of electric current

Damage to the ball (fig. 11.9) and the raceway (fig. 11.10) by the passage of electric current through the roller contact. This type of damage is unacceptable. This forms when sparking occurs over a thin layer of lubricant. Burned-out cratering forms on such sites and are a source of bearing vibration and increased noise. This type of damage in motor housings and other roller-contact seats of rail vehicles with electrical traction are prevent, for example, by the use of bearings with an insulation layer on one of the rings and by the use of hybrid bearings with ceramic balls.

Fig. 11.9

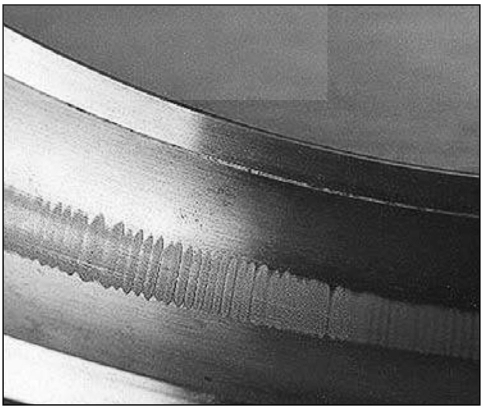

Fig. 11.10

Wear

Wear on the rolling surfaces of cylindrical rollers (fig. 11.11) and races (fig. 11.12) are caused by lubrication failure without flaking of material. Such damage may occur primarily in areas, where maintenance of the lubricating film is hindered, such as bearing ring faces or on roller faces. Undesirable wear may also occur due to slippage of rolling elements towards the bearing rings. Wear is characterized by traces of seizing and slippage, which is often accompanied by brownish spots on the raceway. This is unacceptable wear.

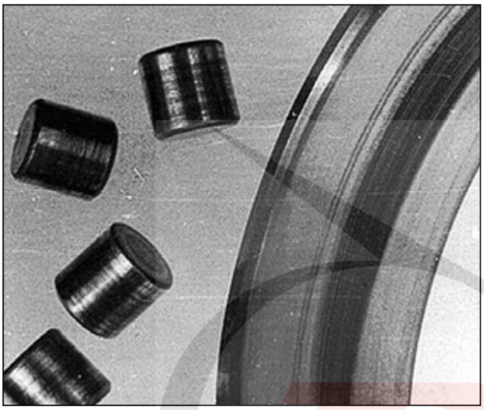

Fig. 11.11

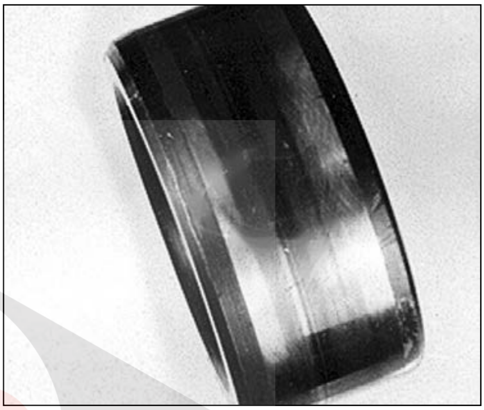

Fig. 11.12

Corrosion

The first picture (fig. 11.13) shows traces of acceptable contact corrosion on the raceway and the second (fig. 11.14) show inner ring corrosion. Corrosion resulting from inadequate protection against moisture or the use of an unsuitable lubricant is always impermissible. Areas affected by rust formation may progressively become initial sites of flaking of operating surfaces, which can lead to deteriorated operating precision and decreased bearing durability. Corrosion occurs when atmospheric moisture condenses, which can occur under improper

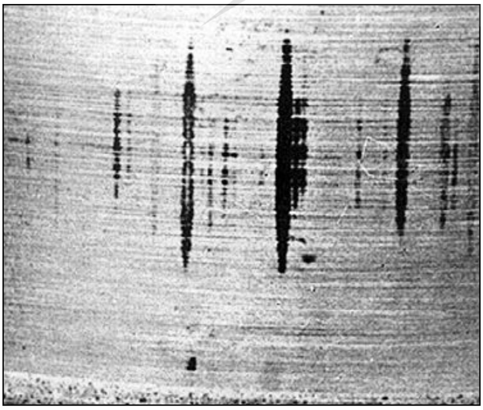

Fig. 11.13

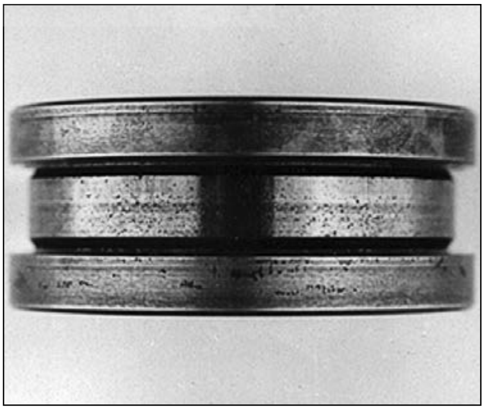

Fig. 11.14

storage conditions. Contact corrosion is caused by very weak oscillations or vibrations of loose components, which can lead to serious bearing damage and thus prevent their further use.

Cage damage

Under normal operating conditions, the roller bearing cage is stressed little. Damage primarily occurs due to poor lubrication. When lubrication is inadequate, cage wear first occurs on the surfaces in contact with rolling elements or with guiding surfaces of bearing rings. The first picture (fig. 11.15) shows deep cage pocket wear from contact with the cylindrical roller with traces of flaked material. This extent of damage is impermissible. The second picture (fig. 11.16) shows permissible glazing of the guide diameter of the solid bronze ball bearing cage.

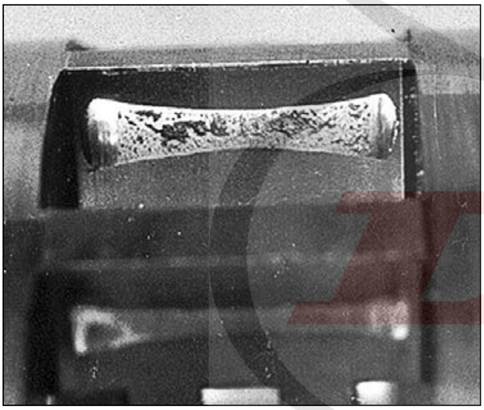

Fig. 11.15

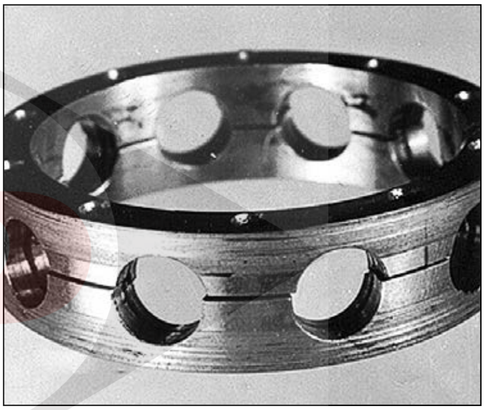

Fig. 11.16