rolling bearings

technical handbook

10/2001-TP-VL-A-Rev.1

PREFACE

PSL, situated in Považská Bystrica, is a bearing producer with many years tradition dating back to 1948.
The present production assortment contains more than 200 types of standard and special rolling bearings in following design groups:
- single row, double row and multi-row cylindrical roller bearings
- single row, double row and four-row tapered roller bearings
- double row spherical roller bearings
- thrust ball bearings
- thrust cylindrical roller bearings
- thrust tapered roller bearings

This publication contains basic technical data and procedures when designing and dimensioning of a common arrangement using bearings from the PSL production programme. Their boundary dimensions and basic parameters are shown in the publication “Rolling Bearings PSL - Production Programme“, publication No. 4/98- VLO-A.
Solutions to complex applications of standard and special bearings can be provided on request by the experts of the PSL Technical Consultancy Department.

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Quality of the PSL products and the quality management system are approved to the international Quality Standard ISO 9001, ISO 14001and other standards as follows:

Survey of Standards Used by Design and Production of Rolling Bearings

Parameter		Standard
Boundary dimension	- radial bearings	ISO 15
	- thrust bearings	ISO 104
	- tapered roller bearings in metric dimensions	ISO 355
Abutment and fillet dimensions		ISO 582
Limiting values for dimension and operation accuracy	- radial bearings	ISO 492
	- thrust bearings	ISO 199
Basic dynamic load rating		ISO 281
Basic static load rating		ISO 76
Radial and axial clearance		ISO 5753
Quality system.Model of quality assurance by design, production, putting into operation and service		ISO 9001

1. SELECTION OF THE TYPE AND DETERMINING OF THE BEARING SIZE

1.1 Basic Criteria for Common Arrangement Design

When selecting the type and size of the bearing, it is necessary to evaluate the arrangement as a unit according to the following criteria:
- requirements on space,
- size, direction and type of load,
- rotational speed,
- operational accuracy and arrangement rigidity,
- axial displacement and permissible misalignment,
- requirements on mounting, dismounting and maintenace of bearings in operation,
- economic requirements on the arrangement.

Priority of individual criteria is various and depends on the requirements on the arrangement.

Requirements on Space

The development of the machinery equipment is oriented on smaller and lighter designs by growing output of the equipment.The space filled by the arrangement should be as small as possible, but the bearing efficiency should secure compliance between the technical life of the arrangement and the technical life of the equipment.

Size, Direction and Type of Load

Data about the load are most decisive for determining the type and size of the bearing. Bearings with line contact (cylindrical roller, spherical roller, tapered roller) have a higher basic load rating than bearings with the point contact (ball) of the same dimensions.
The ability of bearings to accommodate forces in the radial or axial direction depends especially on the contact angle, i.e. the angle, which is formed by the connecting line of the contact points of the rolling elements with the perpendicular line to the rotational axis of the bearing. The selection of the bearing with a suitable contact angle depends on the ratio of the axial and radial load, see Figure 1.
The rolling bearings can be affected by the dynamic or static load. By the dynamic load the loaded bearing rotates. By the static load the bearing is loaded at rest, or it moves in a slow swinging way, or rotates very slowly ($n<10\ min^{-1}$).
By the dynamic load the bearing life due to material fatigue is decisive for the calculation, by the static load it is the rise of permanent deformations of the functional surfaces in the contact surface of the rolling elements with the raceways and corresponding bearing safety by the static load.

Rotational Speed

Permissible rotational speed depends on more operational factors which determine the temperature development in the arrangement. It is limited especially by the operational temeprature of the lubricant. For a high rotational speed those bearings are more suitable which develop less heat and have lower friction, and thus lighter operation.

Operational Accuracy and Arrangement Rigidity

Operational accuracy is important especially in spindle arrangements of machine tools, but also in other equipment, as e.g. positioners, gauging systems, etc. Tolerances of the dimension and operation accuracy for the PSL standard bearings are shown in chapter 2.3 of this publication. In some cases besides the operational accuracy also the arrangement rigidity is important. Bearings with the line contact have a higher rigidity than the bearings with the point contact. The rigidity can be changed by a suitable bearing arrangement (“O“, “X“ arrangements, etc.), or by the adjustment of a suitable preload.

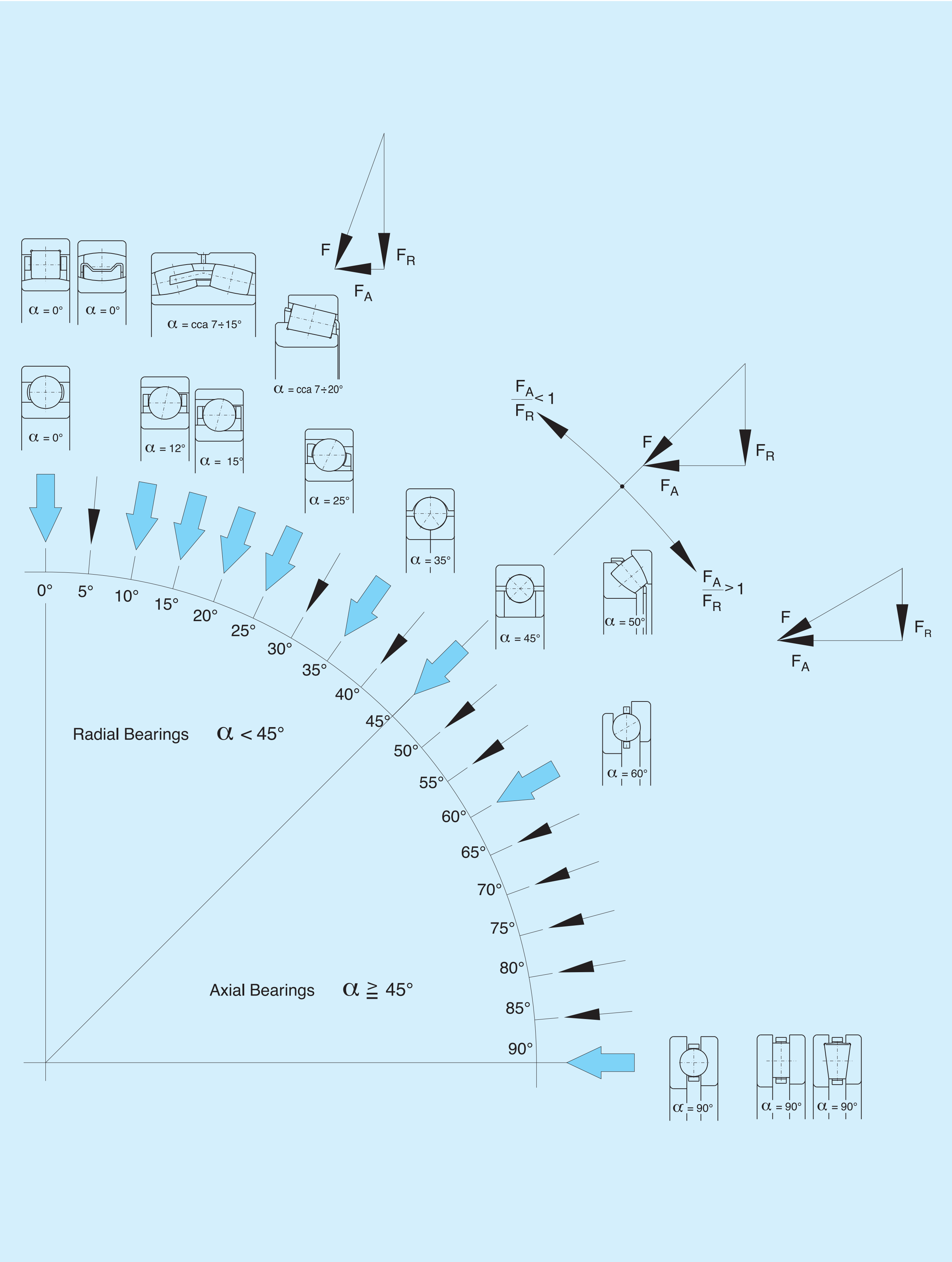
Axial Displaceability and Permissible Misalignment

Every arrangement must enable a shaft dilatation due to the operational temperature change.That is why one bearing is axially rigid, the other axially displacable. The axial displacability can be secured directly by the bearing (i.e. the bearing design enables the mutual displacement of the rings - e.g. cylindrical roller bearings of the N, NU design, etc.), or by a displacable arrangement of one ring (on the shaft or in the housing - according to the type of load).
In cases when it is not possible to secure sufficient alignment of the arrangement surfaces, or by great shaft deflection, it is necessary to use bearings which enable the required misalignment.
Values of the permissible misalignment of the individual bearing types are shown in chapter 2.5 of this publication.

Requirements on Mounting, Dismounting and Maintenace of Bearings in Operation

The arrangment must fulfil conditions for easy, simple mounting, dismounting and minimum bearing maintenance in operation.
The final decision, which type and size of bearing should be used, must be preceded by a complex economic analysis and optimisation of the arrangement.

Figure 1


F
F_R
F_A
α = 0°
α = 0°
α = cca 7÷15°
α = cca 7÷20°
α = 0°
α = 12°
α = 15°
α = 25°
α = 35°
α = 45°
α = 50°
α = 60°
$\frac{F_A}{F_R} < 1$
$\frac{F_A}{F_R} > 1$
0°
5°
10°
15°
20°
25°
30°
35°
40°
45°
50°
55°
60°
65°
70°
75°
80°
85°
90°
Radial Bearings $\alpha < 45°$
Axial Bearings $\alpha \geqq 45°$
α = 90°
α = 90°
α = 90°

1.2 Dynamic Load

1.2.1 Basic Dynamic Load Rating

The basic dynamic load rating is a constant, non-variable load under which the bearing attains the nominal life of one million revolutions. For radial bearings, the radial dynamic load rating C_r refers to a constant pure radial load. For thrust bearings, the axial dynamic load rating C_a refers to a constant pure axial load acting in the bearing axis.
The basic dynamic load ratings C_r and C_a which depend upon the bearing size, number of rolling elements, material and bearing design are given in the dimension tables for each bearing. Values of the basic dynamic load ratings have been determined according to the international standard ISO 281. These values are verified both in testing and in normal operation.
If the operational temperature is higher than 120°C, the hardness of the material due to changes of the material structure decreases and results in decrease of the load rating.

Following equation shows the decrease of the dynamic load rating due to the temperature:

$$C_T = f_t \,.\, C$$

where:	C_T	-	real dynamic load rating	[kN]
	C	-	basic dynamic load rating	[kN]
	f_t	-	factor of the operational temperature (Table 1)	[-]

Values of Factor f_t Table 1

Operational temperature to [°C]	150	200	250	300
Factor f_t	0.95	0.9	0.75	0.6

1.2.2 Equivalent Dynamic Load

The rolling bearing is generally subjected to forces of different directions and magnitudes sometimes at various rotational speeds and during different periods of time. It is necessary to convert all acting forces to the hypothetical constant load which, when acting, has the same effect on the bearing life as the actual load.
This derived hypothetical constant radial or axial load is called the equivalent dynamic load P or P_r (radial) and P_a (axial).
If the bearing is subjected to the radial and axial load of a constant magnitude and direction simultaneously, the following equation is valid for calculating of the equivalent dynamic load:

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

where:	P	-	equivalent dynamic load	[kN]
	F_r (F_{rs})	-	radial bearing load (medium radial bearing load)	[kN]
	F_a(F_{as})	-	axial bearing load (medium axial bearing load)	[kN]
	X	-	factor of the radial load	[-]
	Y	-	factor of the axial load	[-]

Factors X and Y for individual types and sizes of bearings - see publication “Rolling Bearings PSL - Production Programme 11/2001-VLO-A“.

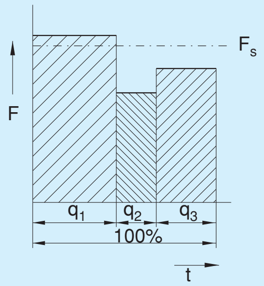
Fluctuating Load

In many applications, bearings are subjected to a fluctuating load under constant or fluctuating speed. The system of external forces acting on the bearing must be recalculated into forces acting in radial and axial directions. If the load is fluctuating, its course in relation to time must be known. The fluctuating load is converted to the hypothetical mean load having the same effect on the bearings as the actual acting fluctuating load.

Varying Load Magnitude

If the bearing is subjected to the load in a constant direction, whose magnitude is changed within a certain period of time at a constant speed (Figure 2), the mean constant load F_S can be calculated from the following equation:

Figure. 2



$$F_S = \left(\sum_{i=1}^{n} F_i^3 \frac{q_i}{100} \right)^{\frac{1}{3}}$$

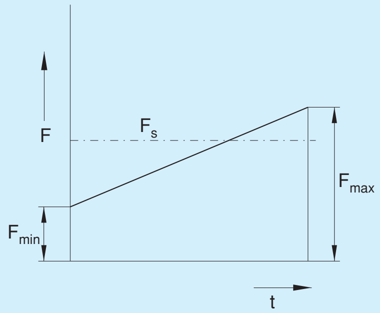
where: F_S - mean constant load [kN]

F_i = F_1 , F_2,...,F_n - constant fractional load [kN]

q_i = q_1, q_2,...,q_n - share of fractional load acting [%]

At a constant rotational speed with a linear change of the load with a constant direction (Figure 3), the mean constant load is calculated from the following equation:

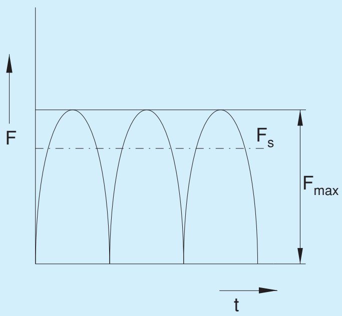
Figure. 3



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

At the load sine wave behaviour (Figure 4) the mean constant load is:

Figure. 4



$$F_S = 0.75\ F_{max}$$

Varying Load Magnitude and Varying Rotational Speed

If the bearing is subjected in time to a varying load and simultaneously with the change of load also the rotational speed is changed, the mean constant load is calculated from the following equation:

$$F_s = \left(\frac{\sum_{i=1}^{n} F_i^3 q_i . n_i}{\sum_{i=1}^{n} q_i . n_i} \right)^{\frac{1}{3}}$$

where: $n_i = n_1, n_2, ..., n_n$ - constant rotational speed while fractional loads $F_1, F_2 ....., F_n$ act [min^{-1}]

$q_i = q_1, q_2, ..., q_n$ - share of fractional load effects and rotational speed [%]

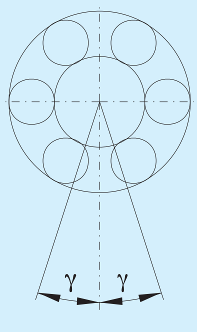
If only the rotational speed varies in time, the mean (constant) rotational speed is calculated from the following equation:

$$n_s = \frac{\sum_{i=1}^{n} q_i . n_i}{100}$$

Oscillating Motion

Under oscillating motion of amplitude γ (Figure 5) it is simplest to substitute the oscillating motion by a hypothetical rotation of the speed which equals the oscillation frequency. For radial bearings the mean constant load is calculated according to the following equation:

Figure 5



$$F_s = F_r \left(\frac{\gamma}{90} \right)^{1/p}$$

where: F_s – mean constant load [kN]

F_r - radial bearing load [kN]

γ – oscillating motion amplitude [°]

p – exponent $p = 3$ for ball bearings

$p = 10/3$ for cylindrical roller, spherical roller and tapered roller bearings

1.2.3 Life

The rolling bearing life is defined as the number of revolutions or operating hours at a constant rotational speed until the first signs of material fatigue occur on the ring raceways or on the rolling element.
Bearings of the same type can substantially differ in their lives, and therefore the life calculation according to the ISO 281 is based on the nominal life, i. e. the life which is attained or exceeded by 90% of a greater number of apparently identical bearings operating under the same conditions, i. e. life with 90% reliability.
The life is the time period of bearing operation until failure occurs due to the dynamic fatigue of rings or rolling elements, and it does not involve unforseen causes, as e. g. entered impurities, incorrect lubrication, unsuitable arrangement or non-professional mounting.

Life equation

The bearing nominal life is mathematically defined by the life equation valid for all bearing types

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

where: L_{10} - nominal life [10^6rev.]
C - basic dynamic load rating [kN]
P - equivalent dynamic bearing load [kN]
p - exponent p = 3 ball bearings
p = 10/3 for cylindrical roller, spherical roller and tapered roller bearings

The rotational speed is usually constant therefore the revised life equation, which expresses the nominal life in operating hours, is used:

$$L_{10h} = \left(\frac{C}{P}\right)^{p} \cdot \frac{10^6}{60.n}$$

where: L_{10h} = nominal life [h]
n = rotational speed [min^{-1}]

Adjusted Life

The increased design reliability and the effort to increase the bearing life require precise calculations. The revised equation serves for this purpose:

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

where: L_{na} - adjusted life for reliability of (100-n)% and operational conditions involved
a_1 - life factor for reliability different from 90%, see Table 2
a_{23} - life factor for material of unconventional properties according to the production technology level and operational conditions

Values of Factor a_1 Table 2

Reliability (%)	L_n	a_1
90	L_{10}	1.00
95	L_5	0.62
96	L_4	0.53
97	L_3	0.44
98	L_2	0.33
99	L_1	0.21

The factor a_{23} by common operational temperatures depends on the lubrication where two influences play a decisive role. The first is the physical one, i.e. viscosity and lubricant purity which substantially influences creation of the lubricating film on the functional surfaces of the bearing rings and rolling elements. The other is the chemical one, e.g. additives increasing the lubricant film efficiency. The factor a_{23} can be determined from the diagram in Figure 6 in dependence on the relation of the viscosities κ, where:

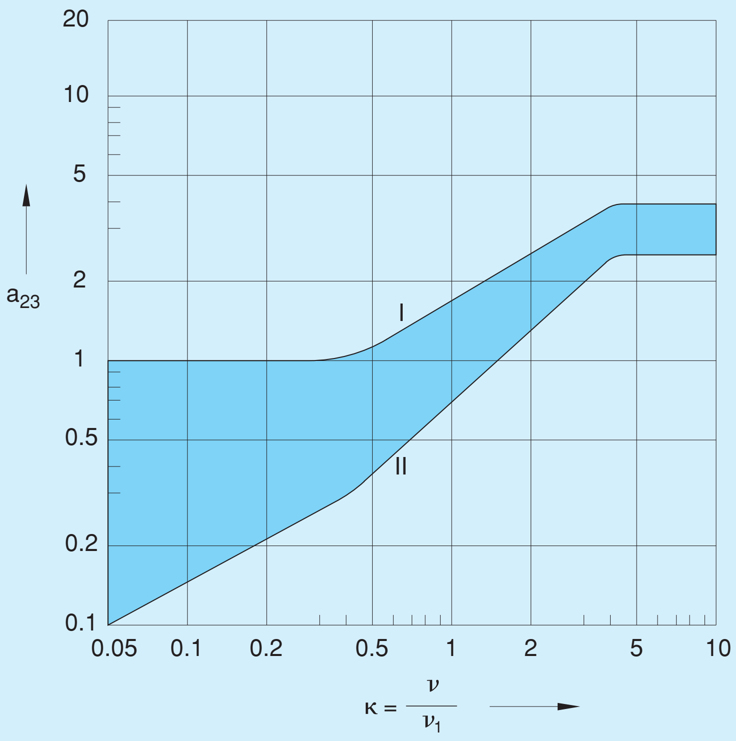
$$\kappa = \frac{\nu}{\nu_1}$$

where: ν - kinematic viscosity of the applied lubricant at the operating temperature. For common mineral oils it can be obtained from the diagram in the Figure 24, page 62 (chapter 4.2.1). If the bearings are lubricated by grease the calculation may be based on the basic oil viscosity of the grease.

ν_1 - required kinematc viscosity for securing of inevitable lubrication. According to the mean diameter of the bearing and the rotational speed it can be obtained from the diagram in Figure 23, page 62 (chapter 4.2.1).

The state, when the required viscosity ν_1 is the same as the operating one, i.e. $\kappa = 1$, corresponds to the lubrication level which is supposed when calculating the nominal life according to the ISO 281.
If $\kappa > 4$ a perfect separation of the contact surfaces by the lubricant film is achieved, factor a_{23} = approximately 3.
If $\kappa < 4$ a mixed lubrication with friction is achieved. The smaller the κ is, the greater is the share of the solid element contact and the lubricant film has a very thin thickness.

Figure 6



The dark part of the diagram is valid for the bearings with a small share of the sliding friction. When using suitable additives and the lubricant cleanliness is good, the factors a_{23} according to the line I can be used. For bearings with a high share of the sliding friction (especially when $\kappa < 1$) and when the cleanliness of the lubricant is poor, the factors a_{23} are according to the line II.

1.3 Static Load

1.3.1 Basic Static Load Rating

If the load acts on the rolling bearing at rest or at a very slow rotation ($n < 10\ min^{-1}$), at an oscillating motion or if the bearing is subjected to impacts or forces during a shorter period of time than one revolution, the bearing load must not be determined by the raceway dynamic fatigue but by the permissible permanent deformations of raceways and rolling elements.
The radial basic static load rating C_{or} or the axial basic static load rating C_{oa} are given in the dimension tables for each bearing. These values of the basic static load ratings have been stated according to the standard ISO 76.

1.3.2 Equivalent Static Load

The relation of the bearing equivalent static load to the actual load and its definition is similar to that of the dynamic equivalent load (section 1.2.2).
The general equation for the radial or axial equivalent static load calculation is

$$P_o = X_o\, F_r + Y_o\, F_a$$

where:	P_o	-	equivalent static load	[kN]
	F_r	-	radial bearing load	[kN]
	F_a	-	axial bearing load	[kN]
	X_o	-	radial load factor	[-]
	Y_o	-	axial load factor	[-]

The factors X_o a Y_o are given for individual bearing types and sizes in the publication.

1.3.3 Bearing Safety under Static Load

The ratio of the basic static load rating C_o and the equivalent static load P_o is compared with the safety factor verified in the practice.

$$s_o = \frac{C_o}{P_o}$$

where:	s_o	-	safety factor	[-]
	C_o	-	basic static load rating	[kN]
	P_o	-	equivalent static load or maximum impact force under distinct impact load	[kN]

Table 3 shows values of the smallest permissible safety factors under the static load s_o for various operating conditions. The precise factor s_o values cannot be stated because when determining them, they are based on experience and values verified in practice.

Factor s_0 Values

Table 3

Bearing Motion	Kind of Load, Requirements on Running	s_0	
		Ball Bearings	Cylindrical Roller, Spherical Roller and Tapered Roller Bearings
Rotary	Distinct impact load, high requirements on smooth running	2	4
	After static loading the bearing rotates under less heavy load	1.5	3
	Normal operating conditions and normal requirements on running	1	1.5
	Smooth impact free running	0.5	1
Oscillating	Small oscillation angle with high frequency with impact uneven loading	2	3.5
	Large swinging angle with low frequency and approximately constant periodical load	1.5	2.5
Bearing at rest - not rotating	Distinct impact load	1.5 to 1	3 to 2
	Normal and small load, no special requirements on bearing running	1 to 0.4	2 to 0.8

1.4 Limiting Speed

The limiting speed is determined by a summary of factors which influence the heat generation in the bearing. The limiting speed values shown in the publication "Rolling Bearings PSL- Production Programme 11/2001-VLO-A" were stated for standard bearings and normal tolerance class.They are valid under adequate load (if $C/P \geq 12$ and $F_a/F_r \leq 0.2$) and normal operational relations (correct running clearance, ring fixing, sealing, lubrication,...).

In some special arrangements (if $C/P < 12$ and $F_a/F_r > 0.2$, high rotational speed, sealing of bearings with a contact sealing) the table value of the limiting speed should be adapted according to the following equation:

$$n_k = f_{n1} \cdot f_{n2} \cdot f_{n3} \cdot f_{n4} \cdot n_g$$

where:
- n_k - revised rotational speed [min^{-1}]
- f_{n1} - factor of the load magnitude (Figure 7)
- f_{n2} - factor of load combination (Figure 8)
- f_{n3} - factor of limiting speed overspeeding (Table 4)
- f_{n4} - factor of sealing (Table 5)
- n_g - catalogue value of the limiting speed (see publication "Rolling Bearing PSL- Production Programme") [min^{-1}]

Overspeeding of the rotational speed ($f_{n3} > 1$) requires as a rule:
- adaptation of lubrication and cooling
- increased bearing and connecting components tolerance class
- greater radial clearance than normal
- solid cage of suitable design

In these cases we recommend contacting the specialists of the PSL Technical Consultancy Department (address see page 2).

Figure 7

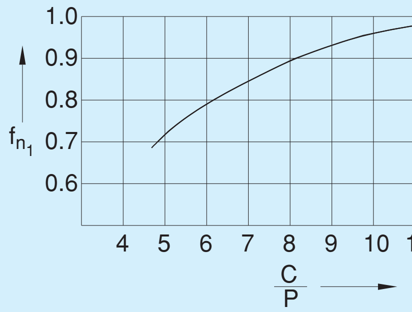
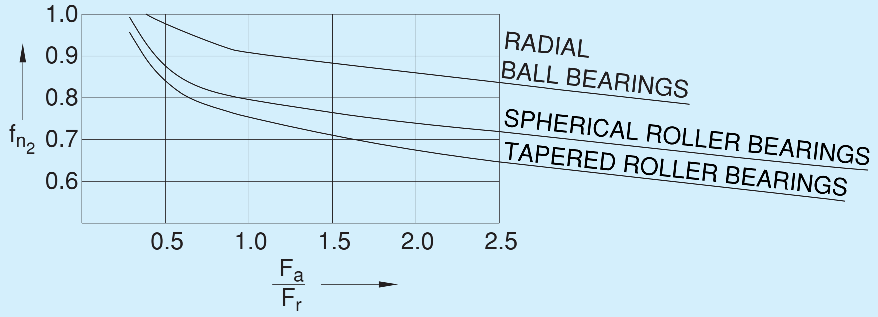


Figure 8



Factor f_{n3} Table 4

Bearing Type		max. f_{n3}
Radial	ball bearings	3
	cylindrical roller bearings	2.5
	tapered roller bearings	2
	spherical roller bearings	1.5
Thrust	ball bearings	1.4
	cylindrical roller and tapered roller bearings	2

Factor f_{n4} Table 5

Type of Sealing	f_{n4}
- unsealed bearing or bearing with non-contact sealing	1
- bearing with a simple contact sealing (RS, ZRS,...)	0.7
- lbearing with contact sealing combined with axial a high effective sealing	0.6

1.5 Friction

The friction level is influenced by following factors:
- design, size and accuracy of the bearing
- size of the operation clearance, or preload
- direction and size of the load
- method of lubrication, lubricant properties
- rotational speed

For normal operational conditions ($C/P > 10$; $n \leq 2/3\, n_g$, suitable lubrication) it is possible to calculate the friction moment with sufficient accuracy according to following equation:

$$M = \mu \cdot F \cdot \frac{d_m}{2}$$

where:
- M - friction moment [Nmm]
- F - bearing load [N]
- d_m - mean bearing diameter [mm]
- μ - friction factor (Table 6) [-]

Friction Factor Table 6

Bearing Type		μ
Radial	deep groove ball	0.0015
	cylindrical roller	0.0011
	tapered roller	0.0018
	spherical roller	0.0018
Thrust	deep groove ball	0.0013
	cylindrical roller	0.0040
	tapered roller	0.0030

2. DESIGN DATA OF ROLLING BEARINGS

2.1 Boundary Dimensions

The majority of the bearings in the publication “Rolling Bearings PSL - Production Programme 11/2001-VLO-A“ are manufactured with boundary dimensions complying with the international standards ISO 15, ISO 355 and ISO 104.
In the dimensional plan of individual bore diameters graded outer diameters are added (designated according to the ascending outer diameter 7, 8, 9, 0, 1, 2, 3 and 4) and widths (designated according to ascending width 8, 0, 1, 2, 3, 4, 5 and 6), or for the thrust bearings the heights (designated according to ascending height 7, 9, 1 and 2). In the pair of digits, the first digit designates the width (height) series and the second one the diameter series, creating together the so called dimension series (Figures 9 and 10).

Figure 9

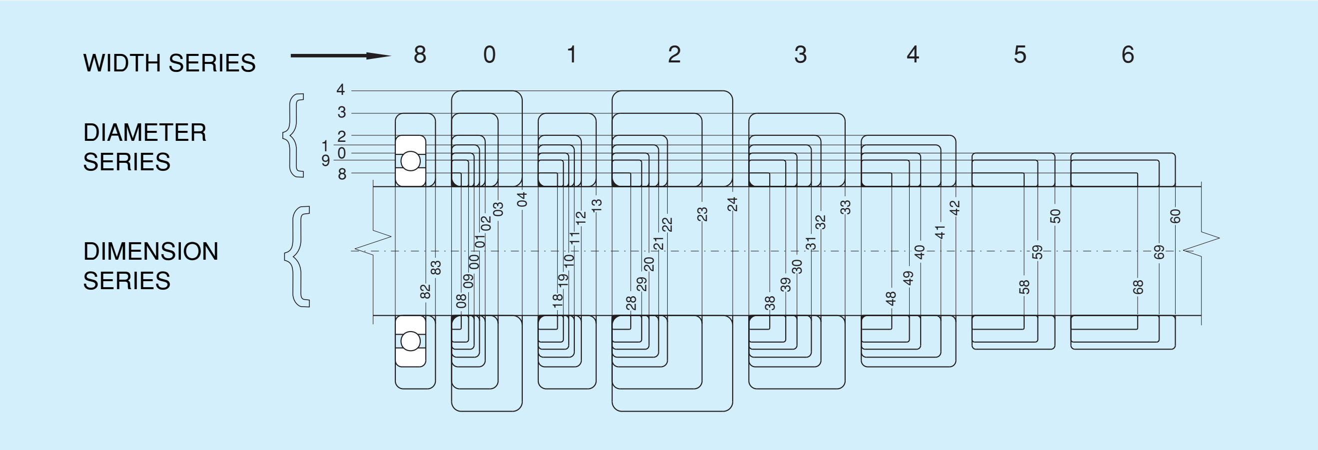


Figure 10

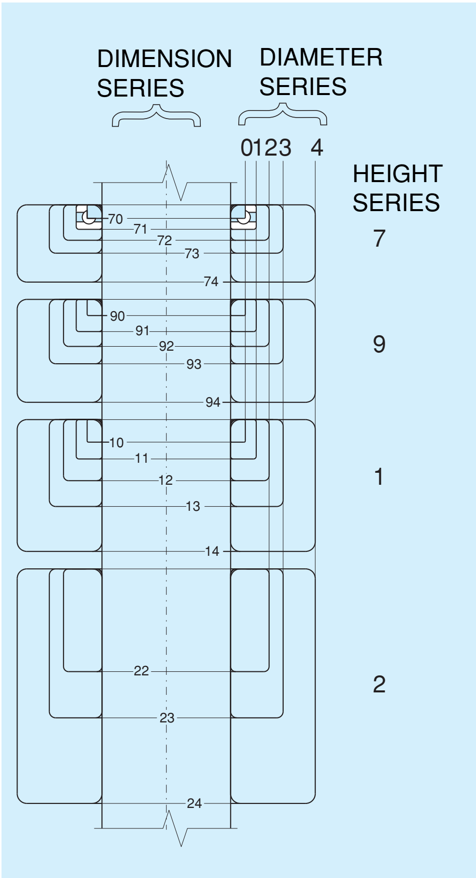
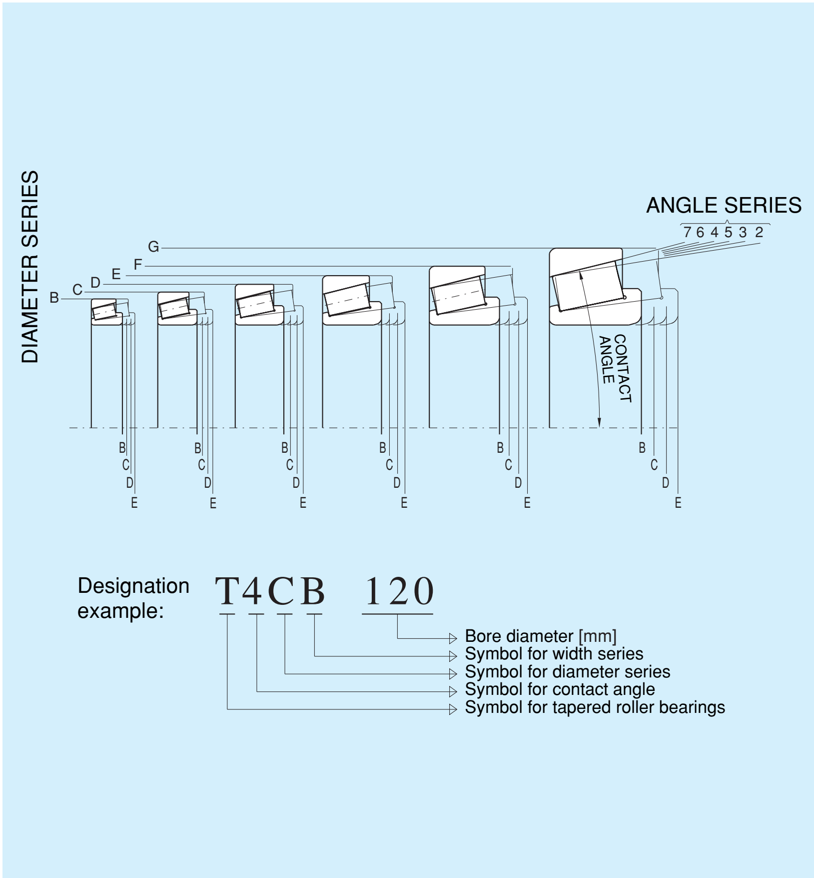


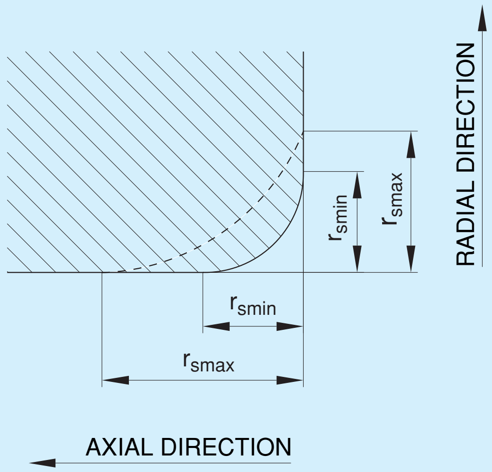
Figure 11



In the new ISO dimension plan for the tapered roller bearings the boundary dimensions are derived from the contact angle and are designated as angle series by digits 2, 3, 4, 5, 6 and 7 (according to the ascending angle $\alpha = 10^\circ$ to 30°). The diameter and width series are designated by letters. The designation of the dimension series is created by the angle series digit and the letters of the diameter and width series (Figure 11).

The dimensional plan also includes the bearing ring chamfer dimensions, i.e. mounting chamfer (Figure 12).
Chamfer limiting values according to ISO 582 - see Table 7.

Figure 12



Limitng Dimensions of Mounting Chamfer

Table 7

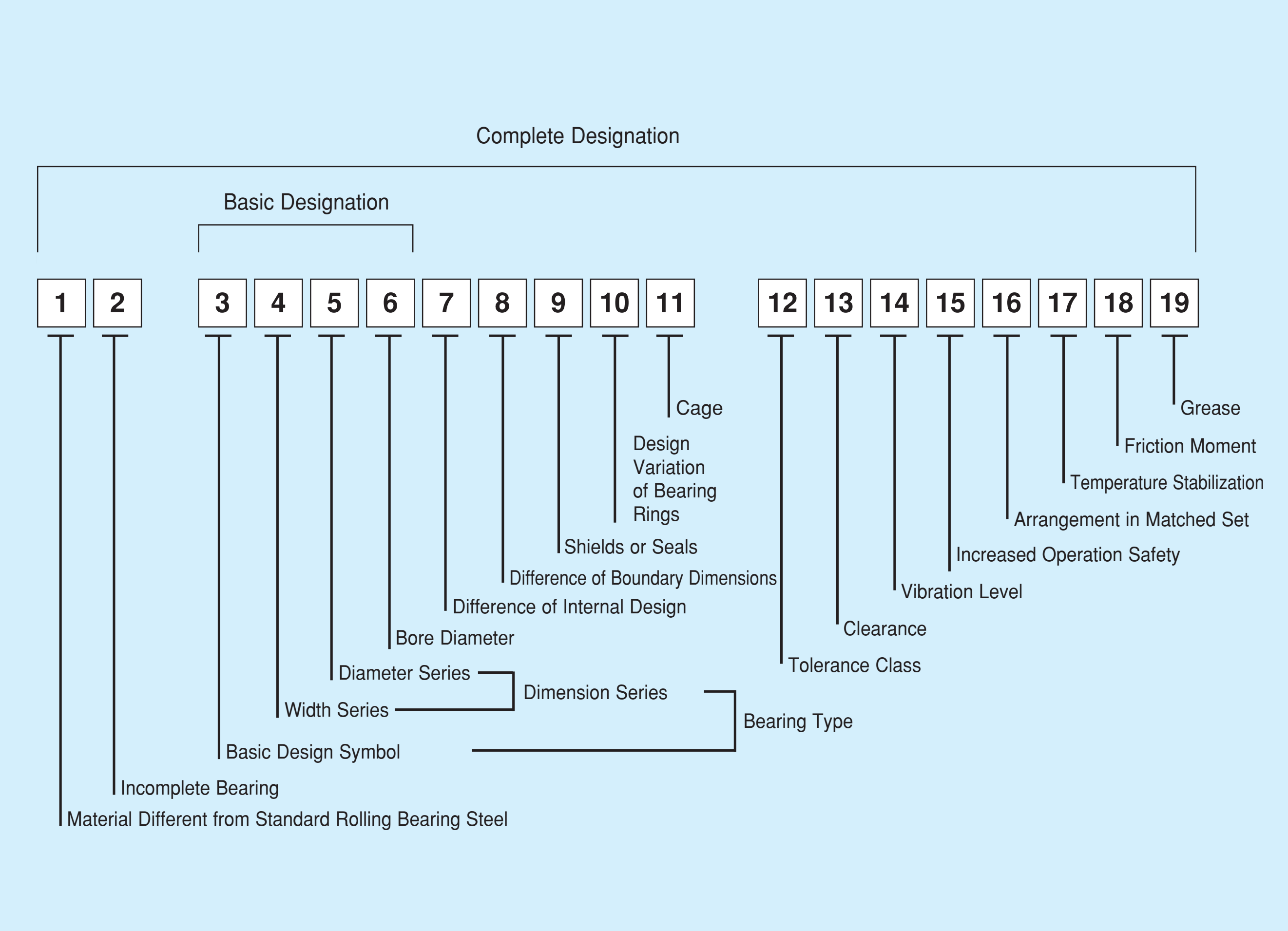
r_{smin}	Radial bearings except tapered roller bearings d or D		r_{smax}		Tapered roller bearings d or D		r_{smax}		Thrust bearings r_{smax}
	above	to	in radial direction	in axial direction	above	to	in radial direction	in axial direction	both in radial and axial direction
mm									
0.15	-	-	0.3	0.6	-	-		-	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

2.2 Designation

2.2.1 Designation of Standard Bearings - Basic Designation

The designation is created by numerical and letter symbols indicating the type, size and design of the bearing. Diagram 1 shows its sequence.
The designations of the type and size create so called "basic designation". The other data create so called "enlarged designations".

Diagram 1



Detailed survey of used symbols - see standard STN 02 4608.

Basic bearing designation consists of the type and size designation

Type designation is formed by the basic design symbol (see position 3, Diagram 1) and by the symbols for the dimension series (see chapter 2.1).
Survey of the design symbols used for the PSL bearings - see Table 8.

The bearing size is stated by the two digit number, whose 5 multiple gives the bearing bore diameter in mm. The exception are the bearings with a bore diameter greater than 500 mm and non-standardized bearings, where the bore size is given separately after a slash directly in mm, e.g. NNU 49/630.

Design Symbols of the PSL Bearings - Selection from the Standard STN 02 4608
(position 3 of the Diagram 1)

Table 8

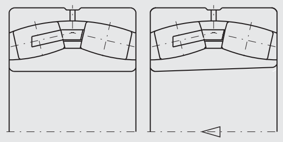
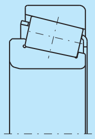
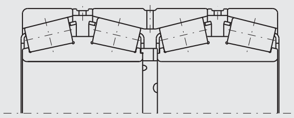
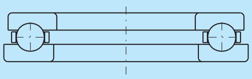
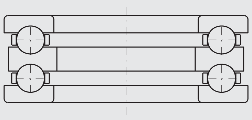
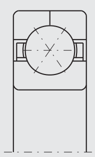
Symbol	Picture	Bearing Type	Designation Example
2	 2.... 2....K 2....K30	Double Row Spherical Roller Bearings	22338 23040K 23144 23238 239/750 24080K30
3		Single Row Tapered Roller Bearings	30218 32024 32222 32314 33020 33116
3		Four Row Tapered Roller Bearings	36032 36972 360/630
5		Thrust Ball Bearings - Single Direction	511/1000 51264
5		Thrust Ball Bearings - Double Direction	52264
8		Thrust Cylindrical Roller Bearings - Single Direction	811/500 81230
Q		Single Row Deep Groove Ball Bearings with Four Point Contact and Split Outer Ring	Q1944

Table 8 - continued

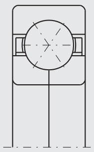
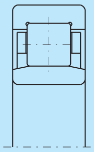
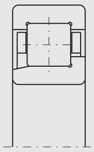
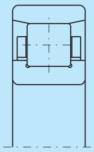
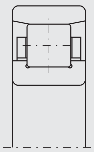
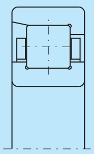
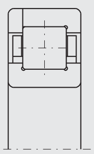
QJ		Single Row Deep Groove Ball Berаings with Four Point Contact and Split Inner Ring	QJ1928
NU		Single Row Cylindrical Roller Bearings with Two Guiding Ribs on Outer Ring and Smooth Inner Ring	NU1060 NU248 NU2280 NU29/118 NU3080
NJ		Single Row Cylindrical Roller Bearings with Two Guiding Ribs on Outer Ring and One on Inner Ring	NJ1060 NJ248
N		Single Row Cylindrical Roller Bearings with Two Guiding Ribs on Inner Ring and Smooth Outer Ring	N2252 N248
NG		One Purpose Cylindrical Roller Bearings of Design Symbol “N“, “Dimension Series 00“ (Digit is diameter of the bore in mm)	NG160 NG180 NG220
NF		Single Row Cylindrical Roller Bearings with Two Guiding Ribs on Inner Ring and One on Outer Ring	NF2240
NP		Single Row Cylindrical Roller Bearings with Two Guiding Ribs on Inner Ring and Two on Outer Ring, One of which Is Flat Loose Rib	NP2240

Table 8 - continued

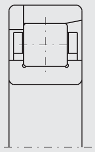
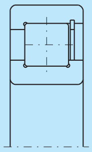
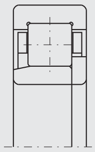
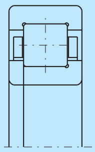
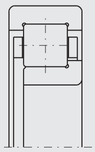
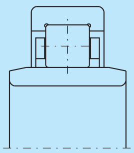
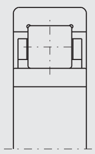
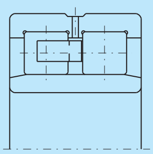
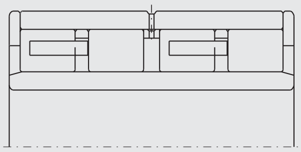
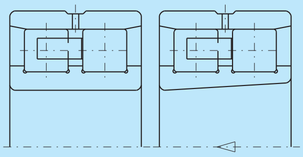
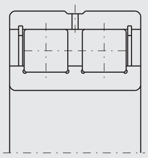
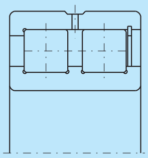
NFP		Single Row Cylindrical Roller Bearings with Two Guiding Ribs on Inner Ring, Rib on Outer Ring Is Created by Flat Loose Rib	NFP2240
NFD		Single Row Cylindrical Roller Bearings with Two Guiding Ribs on Inner Ring, with Rib on One Side of Outer Ring and with Retaining Ring on the Other Side of Outer Ring	NFD2240
NJP		Single Row Cylindrical Roller Bearings with Two Guiding Ribs on Outer Ring, Rib on Inner Ring Is Created by Flat Loose Rib	NJP2252
NUP		Single Row Cylindrical Roller Bearings with Two Guiding Ribs on Outer Ring and Two on Inner Ring, One of which is Flat Loose Rib	NUP1052
NUPJ		Single Row Cylindrical Roller Bearings of Design Symbol NUP without Flat Loose Rib	NUPJ1052
NUB		Single Row Cylindrical Roller Bearings with Two Guiding Ribs on Outer Ring and Wider Inner Ring	NUB1052
NUC		Single Row Cylindrical Roller Bearings with Two Guiding Ribs on Outer Ring and Raceway around Whole Width of Inner Ring	NUC1052
NNU		Double Row Cylindrical Roller Bearings with Ribs on Outer Ring and Smooth Inner Ring	NNU49/630

Table 8 - continued

NNU		Multi Row Cylindrical Roller Bearings with Two and More Guiding Ribs on Outer Ring, Created by Flat Loose Ribs and with Smooth Inner Ring	NNU6040
NN	 NN.... NN....K NN....K30	Double Row or Multi Row Cylindrical Roller Bearings with Guiding Ribs on Inner Ring and Smooth Outer Ring	NN3068K NN3944
NND		Double Row Cylindrical Roller Bearings with Guiding Ribs on Inner Ring and Two Retaining Rings on Outer Ring	NND3944
NNFD		DoubleRow Cylindrical Roller Bearings with Guiding Ribs on Inner Ring, One Guiding Rib on Outer Ring and Retaining Ring on the Other Side of Outer Ring	NNFD3948

2.2.2 Prefixes - Selected from the Standard STN 02 4608 (separated by a gap)

Material Different from Standard Rolling Bearing Steel
(position 1 according to the Diagram 1)

Symbol	Meaning
H	heat-resisting steel
X	corrosion-resisting steel
T	case-hardening steel
Z	steel with special additives (vanadium, molybdenum)

Symbols for Bearing Incompleteness
(position 2 according to the Diagram 1)

Symbol	Meaning
K	Axial cage with short cylindrical rollers - single row
L	Independent removable ring of separable bearing. By thrust ball bearings it is a bearing without shaft washer
R	Separable bearing without one removable ring. By thrust ball bearings it is a bearing without housing washer
E	Independent shaft washer of thrust ball bearing
WS	Independent shaft washer of thrust cylindrical roller bearing
W	Independent housing washer of thrust ball bearing
GS	Independent housing washer of thrust cylindrical roller bearing

2.2.3 Suffixes - Selected from the Standard STN 02 4608

Symbols at positions 7, 8, 9, 10 and 11 according to the Diagram are used together with the basic designation, the other symbols are placed after a gap.

Symbols for Difference of Internal Design
(position 7 according to the Diagram 1)

Symbol	Meaning
E	Single row cylindrical roller bearings with higher load rating
E	Double row spherical roller bearings without ribs with symmetric rolling elements and higher load rating
EE	Double row spherical roller bearings without ribs with symmetric rolling elements, modified rolling surface and higher load rating
C	Double row spherical roller bearings with a rib on inner ring, symmetric rolling elements and higher load rating
CB	Double row spherical roller bearings with drilled symmetric rolling elements, with higher load rating and riveted two-piece cage
CC	Double row spherical roller bearings with a rib on inner ring, symmetric rolling elements, modified rolling surface and higher load rating
A	Single row tapered roller bearings with higher load rating
B	Single row tapered roller bearings with contact angle v >17°

Symbols for Difference of Boundary Dimensions
(position 8 according to the Diagram 1)

Symbol	Meaning
X	Altered boundary dimensions with regard to the dimension plan ISO
X1, X2, X	Altered boundary dimensions, e.g. outer or inner diameter, width, chamfer, etc.

Symbols for Shields and Seals
(position 9 according to the Diagram 1)

Symbol	Meaning
RS *)	Bearings with seal on one side
-2RS *)	Bearings with seals on both sides
RSN*)	Bearings with seal on one side and groove on outer ring on the opposite side
RSNB*)	Bearings with seal on one side and groove on outer ring on the same side
KRS*)**)	Bearings with tapered bore and seal on the side of the smaller bore diameter
KRSB*)**)	Bearings with tapered bore and seal on the side of the larger bore diameter
-2RSN*)	Bearings with seal on both sides and groove on the outer ring
RSR*)	Bearings with seal on one side adhering to flat surface of the inner ring
-2RSR*)	Bearings with seal on both sides adhering to flat surface of the inner ring
-2FS	Bearings with capillary sealing on both sides
Z	Bearings with metal shield on one side
-2Z	Bearings with metal shields on both sides
ZN	Bearings with metal shield on one side and groove on the outer ring on the opposite side
ZNB	Bearings with metal shield on one side and groove on the outer ring on the same side
-2ZN	Bearings with metal shield on both sides and groove on the outer ring
ZR	Bearings with metal shield on one side adhering to flat rib of the inner ring
-2ZR	Bearings with metal shield on both sides adhering to flat rib of the inner ring
PZ	Bearings with shield made of plastic on one side
-2PZ	Bearings with shields made of plastic on both sides
PZR	Bearings with shield on one side made of plastic adhering to flat rib of the inner ring
-2PZR	Bearings with shields on both sides made of plastic adhering to flat rib of the inner ring
RSZ	Bearings with seal on one side and shield on the other side
KZ**)	Bearings with tapered bore and metal shield
KZB**)	Bearings with tapered bore and metal shield on the side of larger bore diameter

*) Behind the symbol RS, but in front of the symbols N or NB, the digital symbol of the seal operational temperature range is placed
- for operational temperatures -30°C to +110°C, not designated
- for operational temperatures -45°C to +120°C, symbol 1
- for operational temperatures -60°C to +150°C, symbol 2
- for operational temperatures -60°C to +200°C, symbol 3

**) For designation of bearings with tapered bore and seal there is an exception in the sequence of symbols in the Diagram 1

Symbols for Design Variation of Bearing Rings
(position 10 according to the Diagram 1)

Symbol	Meaning
K	Radial bearings with tapered bore - taper 1:12
K30	Radial bearings with tapered bore - taper 1: 30
N	Radial bearings with groove for snap ring on the outer ring
NS	Radial bearings with groove for snap ring in the middle of the outer ring
N1	Radial bearings with one slot on the chamfer and outer cylindrical surface
N2	Radial bearings with two slots placed within 180° on the chamfer and outer cylindrical surface
N4	Radial bearings with groove for snap ring on one side of the outer ring, with two slots placed within 180° on the chamfer and outer cylindrical surface on the opposite side
N6	Radial bearings with groove for snap ring on one side of the outer ring, with two slots placed within 180° on the chamfer and outer cylindrical surface on the same side
P	Double row radial bearings with splitted outer ring
D	Double row radial bearings with splitted inner ring
PR	Double row spherical roller bearings with split ring and inserted distance ring
R	Radial bearings with flange on the outer ring
W1	Radial bearings with cylindrical bore with tapered ending on both sides
W20	Radial bearings with lubricating holes on the circumference of the outer ring
W26	Radial bearings with lubricating holes on the circumference of the inner ring
W33	Radial bearings with groove and lubricating holes on the circumference of the outer ring
W513	Radial bearings with lubricating holes on the circumference of both rings and lubricating groove on the outer ring
W518	Radial bearings with lubricating holes on the circumference of both rings
W28	Radial bearings with thread lubricating groove on the surface of the inner ring bore
W528	Radial bearings with groove and lubricating holes on the circumference of the outer ring and thread lubricating groove on the surface of the inner ring bore

Symbols for Cages
(position 11 according to the Diagram 1)

Symbol		Meaning
J	Cage material[x)]	Pressed steel cage, rolling elements centered
Y		Pressed brass cage, rolling elements centered
F		Machined steel cage or cage made of special alloys, rolling elements centered
L		Machined light alloy cage, rolling elements centered
M		Machined brass or bronze cage, rolling elements centered
T		Solid cage made of hardened textite - fabric-reinforced resin, rolling elements centered
TN		Solid cage made of polyamide or similar plastic, rolling elements centered
TNG		Solid cage made of polyamid with glass fibre, rolling elements centered
TNGN		Solid cage made of polyamid with glass fibre for use to 100°C, rolling elements centered
.A	Design[xx)] Cage	Cage centered on outer ring
.B		Cage centered on inner ring
.P		Machined window type cage
.H		Cage One-piece open type cage
J1, J2		Pressed steel cage for tapered roller bearings
...S		Cage with lubricating grooves
...R		Silver coated cage
...F		Phosphated cage
...C		Copper coated cage
...K		Heat treated cage
D		Cage splitted in axial plane
V		Bearings without cage, full complement bearings
VH		Bearings without cage with full complement of cohesive cylindrical rollers
VT		Bearings without cage - rolling elements are separated by rolling elements of smaller diameter

x) If the bearing is in standard design with this cage, symbol is not used.

xx) Symbols of the cage design are used with symbols of the cage material. Behind the material and design symbol, another, as a rule a digital symbol, expressing the production variant, can be placed.

Symbols for Tolerance Class
(position 12 according to the Diagram 1)

Symbol	Meaning	Note
P0	Standard tolerance class	not indicated
P6	Higher tolerance class than P0	
P5	Higher tolerance class than P6	
P4	Higher tolerance class than P5	
P2	Higher tolerance class than P4	
P6E	Tolerance class for electric machines	
P5A	Tolerance class in some parameters higher than P5	
P4A	Tolerance class in some parameters higher than P4	
P6X	Tolerance class for single row tapered roller bearing	

Symbols for Clearances
(position 13 according to the Diagram 1)

Symbol	Meaning	Note
C1	Clearance less than C2	
C2	Clearance less than normal	
-	Normal clearance	not indicated
C3	Clearance greater than normal	
C4	Clearance greater than C3	
C5	Clearance greater than C4	
-	Radial clearance of cylindrical roller bearings with interchangable rings	not indicated
NA	Radial clearance of cylindrical roller bearings with non-interchangable rings [x)]	
R	Radial clearance of bearings in this range is not standardized [x)]	e.g.R10-20 radial clearance in the range of 10 to 20 µm
A	Axial clearance of bearings in this range is not standardized [x)]	e.g.A30-60 axial clearance in the range of 30 to 60 µm

[x)] Placed in the end of designation

Symbols for Vibration Level
(position 14 according to the Diagram 1)

Symbol	Meaning	Note
–	Normal vibration level of rolling bearings	not indicated
C6	Rolling bearing vibration level lower than normal	
C06	Rolling bearing vibration level lower than C6	
C66	Rolling bearing vibration level lower than C06	

Symbols for Increased Operation Safety Level
(position 15 according to the Diagram 1)

C7, C8, C9 - are symbols for bearings with increased operation safety designed primarily for aircraft industry.

Symbols for Arrangement in Matched Set
(position 16 according to the Diagram 1)

The designation of the arrangement in a matched set of two, three or four bearings consists of symbols indicating the bearing arrangement and of symbols determining the internal clearance or preload of the matched set of bearings.

Symbols showing the bearing arrangement

Symbol	Meaning
Matched pair of bearings	
0	“0“ arrangement - matched pair of bearings, the contact axis with regard to bearing axis are divergent
X	“X“ arrangement - matched pair of bearings, the contact axis with regard to bearing axis are convergent
T	Matched pair of bearings, the contact axis are parallel (arrangement in “tandem“)
Matched set of three bearings	
0T	- arrangement “0“ + “T“
XT	- arrangement “X“ + “T“
TT	- arrangement “T“ + “T“
Matched set of four bearings	
0TT	- arrangement “0“ + “TT“
XTT	- arrangement “X“ + “TT“
TTT	- arrangement “TT“ + “TT“
T0T	- arrangement “TT“ + “0“ + “TT“
U	Universally matched bearings

Symbols determining internal clearance or preload

Symbol	Meaning
A	- arrangement of bearings with clearance
0	- arrangement of bearings without clearance
L	- arrangement of bearings with small preload
M	- arrangement of bearings with medium preload
S	- arrangement of bearings with heavy preload
W	- arrangement of bearing pair with approximately the same radial clearance

Symbols for Temperature Stabilization
(position 17 according to the Diagram 1)

Symbol	Meaning	Note
	Bearings, both rings and rolling elements x)	
	of which have dimensions stabilized for operation at temperature to:	
S0	150°C	
S1	200°C	
S2	250°C	
S3	300°C	
S4	350°C	
S5	400°C	
A	Only outer ring has stabilized dimensions	always with stabilization symbol
B	Only inner ring has stabilized dimensions	always with stabilization symbol

x) Rolling elements are stabilized only in reasonable cases.

Symbols for Friction Moment
(position 18 according to the Diagram 1)

Symbol	Meaning
JU	Bearings with determined friction moment under operation
JUA	Bearings with determined friction moment for starting up
JUB	Bearings with determined friction moment for running out

Symbols for Grease
(position 19 according to the Diagram 1)

Symbol	Meaning	Note
TL	Grease for low operating temperatures from -60°C to +100°C	
TM	Grease for medium operating temperatures from -35°C to +140°C	Not necessary to show on bearings and packaging
TH	Grease for high operating temperatures from -30°C to +200°C	
TW	Grease for both low and high operating temperatures from -40°C to +150°C	

2.2.4 Symbol Combination

The symbols of the tolerance class, clearance, vibration level and increased operation safety (position 12 to 15 of the Diagram 1) are combined and the symbol C is omitted at the second and following bearing characteristics.

E.g.:

P4 + C8 to P48
P5 + C2 to P52
C3 + C6 to C36
P6 + C2NA + C6 to P626NA

2.2.5 Bearings according to Special Technical Terms (TP..., TPF..., TPX...)

In some cases the bearings are manufactured and delivered according to technical terms agreed with the customer. These bearings are designated by including the corresponding technical terms following the bearing designation, e.g.: 23236 TPF 1199.

2.2.6 Designation of Non-Standard Bearings

Non-standard - special bearings PSL are designated according to the following scheme:

P S L ☐ ☐ ☐ ☐ ☐ ☐

Letter symbol for special rollings bearings

Design group, or the type of the rolling bearings

- 0- Single row deep groove ball bearings
- 1- Double row ball bearings
- 2- Thrust ball bearings
- 3- Unoccupied group
- 4- Cylindrical roller, needle roller and single row spherical roller bearings
- 5- Cylindrical roller, needle roller, double- and multi row spherical roller bearings
- 6- Single row, double row and four-row tapered roller bearings
- 7- Spindels (special double row ball bearings)
- 8- Assemblies and separated parts
- 9- Cylindrical roller, needle roller, spherical roller and tapered roller thrust bearings

Dimension groups 1 - 12 according to the outside diameter D

Serial number in the respective dimension group

Difference of the internal design

2.3 Tolerance

The rolling bearing tolerance is given by the dimension and running accuracy. The P0 tolerance is the basic one - it is not indicated on either the bearing or packaging. Bearings of higher tolerance classes P6, P5, P4, etc. are indicated by suffixes behind the basic designation.
PSL manufactures standard rolling bearings in the basic tolerance class except for the bearings of the design NN30..K, which are produced in higher tolerance classes. Information about the running accuracy of the non-standard - special bearings PSL can be provided on request by the experts of the PSL Technical Consultancy Department (address - see page 2).
The tolerances of the running and dimension accuracy are shown in tables 9 to 14 and comply with the standards ISO492 and ISO199.

Meaning of the symbols used in the tables:

Symbol	Meaning
d	- Nominal bore diameter
d_1	- Nominal diameter of larger theoretical tapered bore diameter
d_2	- Nominal shaft washer bore diameter of double direction thrust bearings
Δ_{dmp}	- Mean cylindrical bore diameter deviation in one radial plane
Δ_{d1mp}	- Deviation of mean larger theoretical diameter of tapered bore
Δ_{d2mp}	- Mean shaft washer bore diameter deviation of double direction thrust bearings in one radial plane
Δ_{ds}	- Deviation of a single bore diameter
Δ_{Ds}	- Deviation of a single outside diameter
V_{dp}	- Single outside cylindrical surface diameter variation in one radial plane
V_{dmp}	- Mean outside cylindrical surface diameter variation
V_{d2p}	- Shaft washer bore diameter variation of double direction thrust bearings in one radial palne
D	- Nominal outside diameter
Δ_{Dmp}	- Mean outside cylindrical surface diameter deviation in one plane
V_{Dp}	- Single outside cylindrical surface diameter variation in one radial plane
V_{Dmp}	- Mean outside cylindrical surface diameter variation
B	- Inner ring nominal width
T	- Total nominal width of tapered roller bearings
T_1	- Internal sub-unit real width
T_2	- Outside sub-unit real width
Δ_{Bs}	- Inner ring single width variation
Δ_{Cs}	- Outer ring single width variation
Δ_{Ts}	- Bearing single width deviation (total)
Δ_{T1s}	- Internal sub-unit nominal effective width deviation
Δ_{T2s}	- Outside sub-unit nominal effective width deviation
C	- Outer ring nominal widht
V_{Bs}	- Inner ring single width variation
V_{Cs}	- Inner ring single width variation
K_{ia}	- Radial runout of assembled bearing inner ring
K_{ea}	- Radial runout of assembled bearing outer ring
S_i	- Shaft washer raceway axial runout
S_e	- Housing washer raceway axial runout
S_{ia}	- Inner ring flat seat face axial runout of assembled bearing
S_{ea}	- Outer ring flat seat face axial runout of assembled bearing
S_d	- Flat seat face axial runout
S_D	- Runout of outside cylindrical surface towards outer ring face

Dimension and Running Accuracy of Radial Bearings (except Tapered Roller Bearings)
Tolerance Class P0 — Table 9

Inner Ring

d		Cylindrical Bore										Tapered Bore				
		Δ_{dmp}		V_{dp} Diameter Series 7,8 9	V_{dp} Diameter Series 0,1	V_{dp} Diameter Series 2,3,4	V_{dmp}	K_{ia}	Δ_{Bs}		V_{Bs}	Δ_{dmp}		Δ_{d1mp} -Δ_{dmp}		V_{dp} 1)
over	to	max	min	max	max	max	max	max	max	min	max	max	min	max	min	max
mm		µm														
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	-	-	-	-	-
1250	1600	0	-160	-	-	-	-	120	0	-1600	120	-	-	-	-	-
1600	2000	0	-200	-	-	-	-	140	0	-2000	140	-	-	-	-	-

Outer Ring

D		ΔD_{dmp}		V_{Dp} Diameter Series 7,8,9	V_{Dp} Diameter Series 0.1	V_{Dp} Diameter Series 2,3,4	Bearings 2) with Shields	V_{Dmp}	K_{ea}	Δ_{Cs}, V_{Cs}
over	to	max	min	max	max	max	max	max	max	
mm		µm								
50	80	0	- 13	16	13	10	20	10	25	Corresponding to Δ_{Bs} and V_{Bs} of inner ring of the same bearing
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	
1600	2000	0	-200	-	-	-	-	-	220	
2000	2500	0	-250	-	-	-	-	-	250	

1) Valid in any bore radial plane.
2) Valid only for bearings in diameter series 2, 3 and 4.

Dimension and Running Accuracy of Radial Bearings (except Tapered Roller Bearings)
Tolerance Class P6

Table 10

Inner Ring

d		Δ_{dmp}		V_{dp} Diameter Series 7,8,9	0,1	2,3,4	V_{dmp}	K_{ia}		Δ_{Bs}	V_{Bs}
over	to	max	min	max	max	max	max	max	max	min	max
mm		µm									
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
630	800	0	- 50	-	-	-	-	-	0	- 750	55
800	1000	0	- 65	-	-	-	-	-	0	-1000	60
1000	1250	0	- 80	-	-	-	-	-	0	-1250	70

Outer Ring

D		Δ_{Ddmp}		V_{Dp} Diameter Series 7,8,9	0,1	2,3,4	Bearings [1] with Shields	V_{Dmp}	K_{ea}	Δ_{Cs} , V_{Cs}
over	to	max	min	max	max	max	max	max	max	
mm		µm								
50	80	0	- 11	14	11	8	16	8	13	Corresponding to Δ_{Bs}, V_{Bs} of inner ring of the same bearing
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	- 60	75	75	45	-	45	75	
1000	1250	0	- 80	-	-	-	-	-	85	
1250	1600	0	-100	-	-	-	-	-	100	

[1] Valid only for bearings in diameter series 0, 1, 2, 3 and 4.

Dimension and Running Accuracy of Radial Bearings (except Tapered Roller Bearings)
Tolerance Class P5

Table 11

Inner Ring

d		Δ_{dmp}		V_{dp} Diameter Series 7,8,9	V_{dp} Diameter Series 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										
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
400	500	0	-27	28	21	14	17	18	23	0	- 450	18
500	630	0	-33	35	26	18	19	20	25	0	- 500	20
630	800	0	-40	-	-	-	22	26	30	0	- 750	26
800	1000	0	-50	-	-	-	26	32	30	0	-1000	32
1000	1250	0	-65	-	-	-	30	38	30	0	-1250	38

Outer Ring

D		Δ_{Dmp}		V_{Dp} Diameter Series [2)] 7,8,9	V_{Dp} Diameter Series [2)] 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		
mm		µm									
50	80	0	- 9	9	7	5	8	8	10	Corresponding to Δ_{Bs} of inner ring of the same bearing	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		1
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
800	1000	0	-40	50	29	25	35	25	35		25
1000	1250	0	-50	-	-	-	40	30	45		30
1250	1600	0	-65	-	-	-	45	35	55		35

1) Valid only for deep groove ball bearings.
2) Not valid for bearings with shields or seals.

Dimension and Running Accuracy of Tapered Roller Bearings

Tolerance Class P0

Table 12

Inner Ring and Total Width of the Bearing

d over	d to	Δ_{dmp} max	Δ_{dmp} min	V_{dp} max	V_{dmp} max	K_{ia} max	Δ_{Bs} max	Δ_{Bs} min	Δ_{Ts} max	Δ_{Ts} min	Δ_{T1s} max	Δ_{T1s} min	Δ_{T2s} max	Δ_{T2s} min
mm		µm												
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
250	315	0	- 35	35	26	60	0	- 350	+ 350	- 250	+ 150	- 150	+ 200	- 100
315	400	0	- 40	40	30	70	0	- 400	+ 400	- 400	+ 200	- 200	+ 200	- 200
400	500	0	- 45	45	34	70	0	- 450	+ 400	- 400	-	-	-	-
500	630	0	- 50	50	38	85	0	- 500	+ 500	- 500	-	-	-	-
630	800	0	- 75	75	56	100	0	- 750	+ 600	- 600	-	-	-	-
800	1000	0	-100	100	75	120	0	-1000	+ 750	- 750	-	-	-	-
1000	1250	0	-125	-	-	120	0	-1250	+1000	-1000	-	-	-	-
1250	1600	0	-160	-	-	120	0	-1600	+1500	-1500	-	-	-	-
1600	2000	0	-200	-	-	120	0	-2000	+1500	-1500	-	-	-	-

Outer Ring

D over	D to	Δ_{Dmp} max	Δ_{Dmp} min	V_{Dp} max	V_{Dmp} max	K_{ea} max	Δ_{Cs}
mm		µm					
80	120	0	- 18	18	14	35	Corresponding to Δ_{Bs} of inner ring of the same bearing
120	150	0	- 20	20	15	40	
150	180	0	- 25	25	19	45	
180	250	0	- 30	30	23	50	
250	315	0	- 35	35	26	60	
315	400	0	- 40	40	30	70	
400	500	0	- 45	45	34	80	
500	630	0	- 50	50	38	100	
630	800	0	- 75	75	55	120	
800	1000	0	-100	100	75	120	
1000	1250	0	-125	125	94	120	
1250	1600	0	-160	160	120	120	
1600	2000	0	-200	-	-	120	
2000	2500	0	-250	-	-	120	

Dimension and Running Accuracy of Tapered Roller Bearings
Tolerance Class P6X

Table 13

Inner Ring and Total Width of the Bearing

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												
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
180	250	0	-30	30	23	50	0	-50	+150	0	+ 50	0	+100	0
250	315	0	-35	35	26	60	0	-50	+200	0	+100	0	+100	0
315	400	0	-40	40	30	70	0	-50	+200	0	+100	0	+100	0

Outer Ring

D		Δ_{Dmp}		V_{Dp}	V_{Dmp}	K_{ea}	Δ_{Cs}	
over	to	max	min	max	max	max	max	min
mm		µm						
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
315	400	0	-40	40	30	70	0	-100
400	500	0	-45	45	34	80	-	-
500	630	0	-50	50	38	100	-	-

Dimension and Running Accuracy of Tapered Roller Bearings
Tolerance Class P6

Table 14

Inner Ring and Total Width of the Bearing

d		Δ_{dmp}		V_{dp}	K_{ia}	Δ_{Bs}		Δ_{Ts}	
over	to	max	min	max	max	max	min	max	min
mm		µm							
50	80	0	-12		10	0	- 300	+ 200	0
80	120	0	-15	11	13	0	- 400	+ 200	- 200
120	180	0	-18	14	18	0	- 500	+ 350	- 250
180	250	0	-22	17	20	0	- 600	+ 350	- 250
250	315	0	-25	19	25	0	- 700	+ 350	- 250
315	400	0	-30	23	30	0	- 800	+ 400	- 400
400	500	0	-35	26	35	0	- 900	+ 400	- 400
500	630	0	-40	30	40	0	-1000	+ 500	- 500
630	800	0	-50	50	45	0	-1500	+ 600	- 600
800	1000	0	-60	60	50	0	-2000	+ 750	- 750
1000	1250	0	-75	75	60	0	-2500	+1000	-1000

Outer Ring

D		Δ_{Dmp}		K_{ia}	Δ_{Cs}
over	to	max	min	max	
mm		µm			
80	120	0	- 13	18	Corresponding to Δ_{Bs} of inner ring of the same bearing
120	150	0	- 15	20	
150	180	0	- 18	23	
180	250	0	- 20	25	
250	315	0	- 25	30	
315	400	0	- 28	35	
400	500	0	- 33	40	
500	630	0	- 38	50	
630	800	0	- 45	60	
800	1000	0	- 60	75	
1000	1250	0	- 75	85	
1250	1600	0	- 90	95	
1600	2000	0	-120	115	

Dimension and Running Accuracy of Tapered Roller Bearings
Tolerance Class P5

Table 15

Inner Ring and Total Width of the Bearing

d		Δ_{dmp}		V_{dp}	V_{dmp}	K_{ia}	S_d	Δ_{Bs}		Δ_{Ts}	
over	to	max	min	max	max	max	max	max	min	max	min
mm		µm									
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
180	250	0	-22	17	11	13	11	0	-600	+350	-250
250	315	0	-25	19	13	16	13	0	-700	+350	-250
315	400	0	-30	23	15	19	15	0	-800	+400	-400
400	500	0	-35	26	18	22	18	0	-900	+400	-400
500	630	0	-40	30	20	26	20	0	-1000	+500	-500
630	800	0	-50	50	25	30	26	0	-1500	+600	-600
800	1000	0	-60	60	30	35	32	0	-2000	+750	-750

Outer Ring

D		Δ_{Dmp}		V_{Dp}	V_{Dmp}	K_{ea}	S_D	Δ_{Cs}
over	to	max	min	max	max	max	max	
mm		µm						
80	120	0	- 13	10	7	10	9	Corresponding to Δ_{Bs} of inner ring
120	150	0	- 15	11	8	11	10	
150	180	0	- 18	14	9	13	10	
180	250	0	- 20	15	10	15	11	
250	315	0	- 25	19	13	18	13	
315	400	0	- 28	22	14	20	13	
400	500	0	- 33	25	17	23	15	
500	630	0	- 38	29	19	25	18	
630	800	0	- 45	34	23	30	20	
800	1000	0	- 60	45	30	35	25	
1000	1250	0	- 80	75	38	40	30	
1250	1600	0	-100	90	45	45	35	

Dimension and Running Accuracy of Tapered Roller Bearings in Inch Dimensions Table 16

Inner Ring D_{ds}

d		Tolerance Class									
		4		2		3		0		00	
over	to	max	min	max	min	max	min	max	min	max	min
mm		µm									
-	76.2	+13	0	+13	0	+13	0	+13	0	+8	0
76.2	304.8	+25	0	+25	0	+13	0	+13	0	+8	0
304.8	609.6	+51	0	+51	0	+25	0	-	-	-	-
609.6	914.4	+76	0	-	-	+38	0	-	-	-	-
914.4	1219.2	+102	0	-	-	+51	0	-	-	-	-
1219.2	-	+127	0	-	-	+76	0	-	-	-	-

Outer Ring D_{Ds}

D		Tolerance Class									
		4		2		3		0		00	
over	to	max	min	max	min	max	min	max	min	max	min
mm		µm									
-	304.8	+25	0	+25	0	+13	0	+13	0	+8	0
304.8	609.6	+51	0	+51	0	+25	0	-	-	-	-
609.6	912.4	+76	0	+76	0	+38	0	-	-	-	-
914.4	1219.2	+102	0	-	-	+51	0	-	-	-	-
1219.2	-	+127	0	-	-	+76	0	-	-	-	-

Runouts K_{ia}, K_{ea}, S_{ia}, S_{ea}

D		Tolerance Class				
		4	2	3	0	00
over	to	max	max	max	max	max
mm		µm				
-	304.8	51	38	8	4	2
304.8	609.6	51	38	18	-	-
609.6	914.4	76	51	51	-	-
914.4	-	76	-	76	-	-

Note: Tolerance class 4 is a normal tolerance class

Table 16 - continued

Tolerance of the total width of single row bearings ΔT_S

d		D		Tolerance Class									
				4		2		3		0		00	
over	to	over	to	max	min	max	min	max	max	min	max	min	max
mm				µm									
-	101.6			+203	0	+203	0	+203	-203	+203	-203	+203	-203
101.6	304.8			+356	-254	+203	0	+203	-203	+203	-203	+203	-203
304.8	609.6	-	508.0	+381	-381	+381	-381	+203	-203	-	-	-	-
304.8	609.6	508.0	-	+381	-381	+381	-381	+381	-381	-	-	-	-
609.6	-			+381	-381	-	-	+381	-381	-	-	-	-

Tolerance of internal sub-unit ΔT_{1S} width

d		D		Tolerance Class									
				4		2		3		0		00	
over	to	over	to	max	min	max	min	max	max	min	max	min	max
mm				µm									
-	101.6			+102	0	+102	0	+102	-102	+102	-102	+102	-102
101.6	304.8			+152	-152	+102	0	+102	-102	+102	-102	+102	-102
304.8	609.6		508.0	+178	-178	+178	-178	+102	-102	-	-	-	-
304.8	609.6	508.0	-	+178	-178	+178	-178	+178	-178	-	-	-	-
609.6	-			+178	-178	-	-	+178	-178	-	-	-	-

Tolerance of outside sub-unit ΔT_{2S} width

d		D		Tolerance Class									
				4		2		3		0		00	
over	to	over	to	max	min	max	min	max	max	min	max	min	max
mm				µm									
-	101.6			+102	0	+102	0	+102	-102	+102	-102	+102	-102
101.6	304.8			+203	-102	+102	0	+102	-102	+102	-102	+102	-102
304.8	609.6		508.0	+203	-203	+203	-203	+102	-102	-	-	-	-
304.8	609.6	508.0	-	+203	-203	+203	-203	+203	-203	-	-	-	-
609.6	-			+203	-203	-	-	+203	-203	-	-	-	-

Note: Tolerance class 4 is a normal tolerance class

Dimension and Running Accuracy of Thrust Bearings
Tolerance classes P0, P6 and P5

Table 17

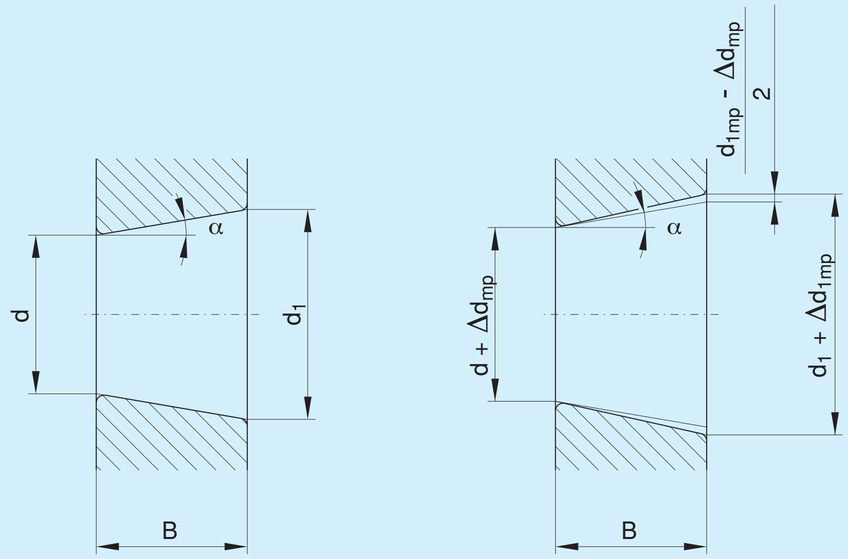
Shaft Washer

d d_2 over	 to	Δ_{dmp} Δ_{d2mp} max	 min	V_{dp} V_{d2p} max	S_i P0 max	 P6 max	 P5 max
mm		µm					
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
800	1000	0	-100	-	45	30	15
1000	1250	0	-125	-	50	35	18
1250	1600	0	-160	-	60	40	21
1600	2000	0	-200	-	75	50	25

Housing Washer

D over	 to	Δ_{Dmp} max	 min	V_{Dp} max	S_e
mm		µm			
80	120	0	- 22	17	Corresponding to S_i of the shaft washer of the same bearing
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	-	
1600	2000	0	-200	-	
2000	2500	0	-250	-	

Tolerances for Tapered Bores



Tolerances of Tapered Bores for Normal Tolerance Class (P0)

Table 18

d		Taper 1:12					Taper 1:30				
		Δd_{mp}		$\Delta d_{1mp} - \Delta d_{mp}$		V_{dp} 1)2)	Δd_{mp}		$\Delta d_{1mp} - \Delta d_{mp}$		V_{dp} 1)2)
over	to	max	min	max	min	max	max	min	max	min	max
mm		µm									
50	80	+ 46	0	+ 30	0	19	+15	0	+30	0	19
80	120	+ 54	0	+ 35	0	22	+20	0	+35	0	22
120	180	+ 63	0	+ 40	0	40	+25	0	+40	0	40
180	250	+ 72	0	+ 46	0	46	+30	0	+46	0	46
250	315	+ 81	0	+ 52	0	52	+35	0	+52	0	52
315	400	+ 89	0	+ 57	0	57	+40	0	+57	0	57
400	500	+ 97	0	+ 63	0	63	+45	0	+63	0	63
500	630	+110	0	+ 70	0	70	+50	0	+70	0	70
630	800	+125	0	+ 80	0	-					
800	1000	+140	0	+ 90	0	-					
1000	1250	+165	0	+105	0	-					
1250	1600	+195	0	+125	0	-					

1) Valid in single real bore cut.
2) Not valid for diameter series 7 and 8.

2.4 Bearing Clearance

The bearing clearance is the displacement value of one ring in reference to the other from one end position to the other in the radial direction (radial clearance) or in axial direction (axial clearance).
Normal bearing clearances are determined so that after bearing mounting the bearing clearance remains suitable for common operational conditions. For special arrangements (great temperature difference of the inner and outer rings, high rotational speed, high required rigidity, etc.) bearings with greater (C3, C4, C5) or smaller (C1, C2) clearances than normal are selected.
Clearance values for bearing types produced in PSL are shown in Tables 19 to 23. Values shown in these tables are valid for bearings before mounting without load by measurment and are in compliance with the standard ISO 5753.

Radial Clearance of Single Row Cylindrical Roller Bearings with Cylindrical Bore Table 19

d		Radial Clearance									
		C2		normal		C3		C4		C5	
over	to	min	max	min	max	min	max	min	max	min	max
mm		μm									
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
1250	1400	270	530	530	790	790	1050	1050	1310	1680	1940
1400	1600	330	610	610	890	890	1170	1170	1450	1920	2200
1600	1800	380	700	700	1020	1020	1340	1340	1660	2160	2480
1800	2000	400	760	760	1120	1120	1480	1480	1840	2400	2750

Radial Clearance of Double Row Cylindrical Roller Bearings with Tapered Bore
Bearings with Non-Interchangable Rings Determined for Spindels of Machine Tools

Table 20

d		Radial Clearance			
		C1NA		C2NA	
over	to	min	max	min	max
mm		μm			
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

Radial Clearance of Double Row Spherical Roller Bearings

Table 21

d		Radial Clearance									
		C2		normal		C3		C4		C5	
over	to	min	max	min	max	min	max	min	max	min	max
mm		μm									
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
1120	1250	320	580	580	860	860	1120	1120	1460	1460	1870
1250	1400	350	640	640	950	950	1240	1240	1620	1620	2060
1400	1600	400	720	720	1060	1060	1380	1380	1800	1800	2300
1600	1800	450	810	810	1180	1180	1550	1550	2000	2000	2550

Radial Clearance of Double Row Spherical Roller Bearings with Tapered Bore Table 22

d		Radial Clearance									
		C2		normal		C3		C4		C5	
over	to	min	max	min	max	min	max	min	max	min	max
mm		µm									
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
1120	1250	570	830	830	1120	1120	1420	1420	1830	1830	2250
1250	1400	620	910	910	1230	1230	1560	1560	2000	2000	2450
1400	1600	680	1000	1000	1350	1350	1720	1720	2200	2200	2700
1600	1800	750	1110	1110	1500	1500	1920	1920	2400	2400	2950

Radial Clearance of Double Row and Four Row Tapered Roller Bearings Table 23

d		Radial Clearance									
		C1		C2		normal		C3		C4	
over	to	min	max	min	max	min	max	min	max	min	max
mm		µm									
80	100	0	20	20	45	45	70	70	100	100	130
100	120	0	25	25	50	50	80	80	110	110	150
120	140	0	30	30	60	60	90	90	120	120	170
140	160	0	30	30	65	65	100	100	140	140	190
160	180	0	35	35	70	70	110	110	150	150	210
180	200	0	40	40	80	80	120	120	170	170	230
200	225	0	40	40	90	90	140	140	190	190	260
225	250	0	50	50	100	100	150	150	210	210	290
250	280	0	50	50	110	110	170	170	230	230	320
280	315	0	60	60	120	120	180	180	250	250	350
315	355	0	70	70	140	140	210	210	280	280	390
355	400	0	70	70	150	150	230	230	310	310	440
400	450	0	80	80	170	170	260	260	350	350	490
450	500	0	90	90	190	190	290	290	390	390	540
500	560	0	100	100	210	210	320	320	430	430	590
560	630	0	110	110	230	230	350	350	480	480	660
630	710	0	130	130	260	260	400	400	540	540	740
710	800	0	140	140	290	290	450	450	610	610	830
800	900	0	160	160	330	330	500	500	670	670	920
900	1000	0	180	180	370	370	550	550	730	730	990
1000	1250	0	200	200	420	420	610	610	790	790	1050
1250	1600	0	220	220	460	460	650	650	850	850	1100
1600	2000	0	240	240	480	480	680	680	900	900	1150

2.5 Permissible Misalignment

Misalignment, i.e. permissible angle of the mutual misalignment of the inner and outer bearing rings depends on the internal design of the bearing, on the bearing clearance and on acting forces and moments.
The reference values of the permissible radial bearing misalignment are in Table 24.

Permissible Misalignment Table 24

Bearing Type	Load	
	small ($F_r < 0.15\ C_{or}$)	heavy ($F_r \geq 0.15\ C_{or}$)
Single Row Cylindrical Roller Bearings		
NU 10, NU2	2' ÷ 3'	5' ÷ 7'
NU 22, NU29, NU30	1' ÷ 2'	3' ÷ 4'
NJ , NUJ, NUP, NH, N	1' ÷ 2'	3' ÷ 4'
Double Row Spherical Roller Bearings		
239, 230, 231	1°30'	
223, 240	2°	
232	2°30'	
Single Row Tapered Roller Bearings	1' ÷ 1.5'	2' ÷ 4'

Thrust bearings - deep groove ball bearings, cylindrical roller bearings and tapered roller bearings require as precise alignment of the arrangement surfaces as possible, any misalignment causes increased stress of the raceways and rolling elements.

2.6 Cages

The rolling bearing cages serve mainly for even separation of the rolling elements around the periphery, they prevent them from mutual contact and sliding. Small and medium bearings have cages pressed from the steel or brass sheet. For larger bearings due to manufacturing reasons machined cages made of steel, brass, light metals, textite, etc. are used.
In some special cases (exceeding the limiting speed, great acceleration, or vibrations, etc.) bearings with machined cage centred on one ring should be used. If the guiding surface of the cage has no lubricating grooves, the bearing should be lubricated with oil. Special arrangements should be consulted with experts of the PSL Technical Consultancy Department - address see page 2.

Information about material and basic cage design for individual bearing types can be found in the publication - “Rolling Bearings PSL - Production Programme“.

3. ARRANGEMENT DESIGN

The main principles which must be taken into account when designing the arrangement design are as follows:
- rotating component should be guided both in the radial as well as axial direction so that it can be statically determined, i.e. supported in two points radially and in one point axially,
- the arrangement must secure reliable transmission of the operational load and required rigidity and operational accuracy
- the arrangement design must enable easy mounting and dismounting so that no additional loads (axial as well as radial) can rise due to pressing or dilatation at the operational temperature changes, excessive mutual ring misalignments, etc.
- the arrangement design must secure reliable sealing, cooling and lubrication, or relubrication during the life of the equipment.

3.1 Bearing Arrangement in Assembly

The bearing arrangement can be „asymmetrical" where the axial forces are accommodated by the axially locating bearing, or „symmetrical" where the shaft is guided by each of the bearings only in one axial direction.

Several typical examples of the bearing assembly are shown in the Table 25.

Examples of Bearing Arrangement in Assembly Table 25

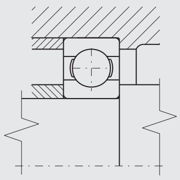
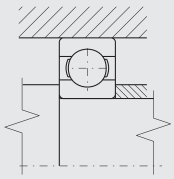
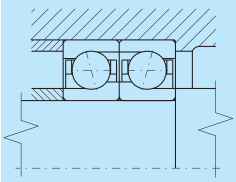
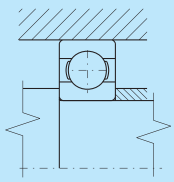
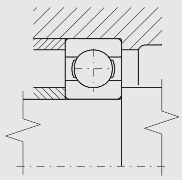
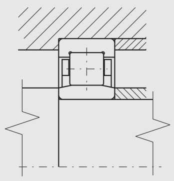
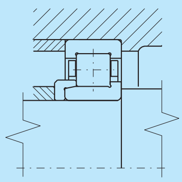
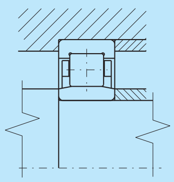
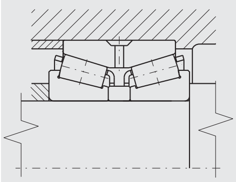
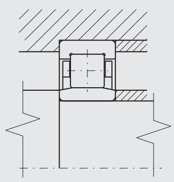
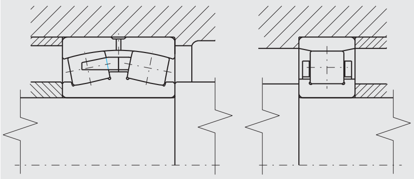
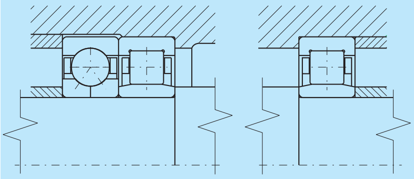
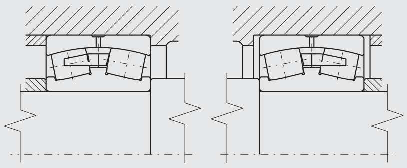
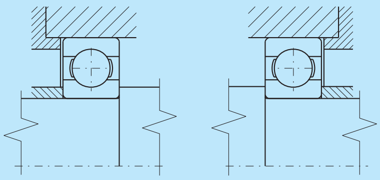
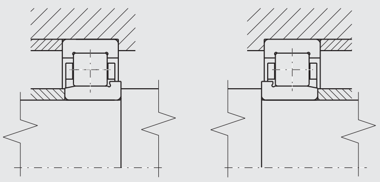
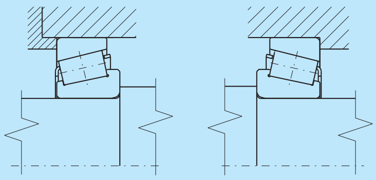
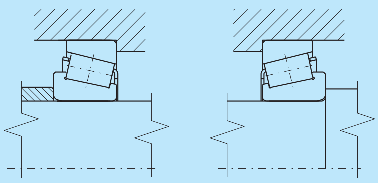
Asymmetrical Bearing Arrangement side axially locating	side axially free	Notes	Application
		- common arrangement for accommodation of radial forces and mild axial forces	- small electric motors - gearboxes - woodworking machines - pumps
		- for accommodation of radial forces and relatively great axial forces	- screwcutting gearboxes - spindels of the machine tools
		- usual arrangement for transmisson of medium and mild axial forces - enables great dilatation - requires a good alignment of the arrangement surfaces	- medium sized electric motors, fans - pulleys
		- this arrangement can transmit great radial and impact forces and small axial forces - bearings are separable (advantageous for mounting and dismounting) - require a good alignment of the arrangement surfaces	- traction motors of railway vehicles - vibration rolls
		- for transmisson of great loads, by high arrangement rigidity - requires high accuracy and alignment of the arrangement surfaces	- reduction rolling mill stands - roller tables - lathe spindels

Table 25 - continued

	- for high radial and small axial loads - suitable mainly there where the shaft is mounted from one side	- large gearboxes - electric motors - rolls of the paper machines - locomotive axles
	- for larger radial and axial loads at high rotational speed - axially guiding bearing must be arranged with radial clearance	- gearboxes
	- arrangement suitable for large radial load at large shaft bend or large misalignment of the arrangement surfaces	- sorters, vibrating screens - rolls of rolling mills - shafts of crank presses - gearboxes

Symmetrical Bearing Arrangement	Notes	Utilization
	- usual arrangement for smaller loads	- smaller electric motors - pumps, gearboxes
 NJ + NJ	- suitable for large impact load - in the arrangement the axial clearance must be secured - similar arrangement with bearings NF + NF	- output shafts of the gearboxes
 Arrangement “X“  Arrangement “O“	- usual arrangement for large and impact loads, accommodates also tilting moments - "O" arrangement has higher rigidity - bearings can be mounted with axial clearance or preload	- gearboxes, wheels and diffrentials of cars

3.2 Location of Rolling Bearings

When selecting the method of the radial and axial ring location, the character and magnitude of acting forces, the operation temperature in the arrangement and the material of the mating components must be considered. The dimensions must be determined with regard to the type and size of the bearing and also the mounting and dismounting method.

3.2.1 Radial Location of Bearing Rings

The rolling bearing rings are located in the radial direction on the mating cylindrical surfaces of shafts and housing bores. In some cases, adapter or withdrawal sleeves are used for mounting on journals, or the bearings are directly mounted on the tapered journal. Correct radial location of the bearing rings on the journal and into the housing considerably influences the utilization of the bearing load rating and its function in the arrangement. Following principles are important:
a) safe location and uniform support of the rings
b) simple mounting and dismounting
c) non-locating bearing displacement in axial direction

Both bearing rings must be mounted in tight fits, because only in this way reliable support around the whole periphery and radial fixing against turning can be achieved. To make the mounting and dismounting of the non-separable bearing easier or to enable displacement the ring of the non-locating bearing, a push fit of one of the rings is permissible.
For a correct selection of the radial location of the ring, following must be taken into account:

Type of Load
- **Circumferential load:** when the respective ring of the bearing rotates and the load direction does not change or if the ring does not rotate and the load rotates. The bearing ring periphery is gradually loaded during one revolution. The bearing ring loaded in this way must be always fitted with the interference fit (fixed).
- **Stationary load:** when the bearing ring does not rotate and the external force is constantly directed towards the same raceway point or if the ring and the load rotate at the same speed. The ring subjected to a stationary load can be mounted with loose fits, if required.
- **Indeterminate load:** when the ring is subjected to varying external forces for which directions and load variations cannot be determined, e.g. forces caused by unbalance rotating mass, shock loads, etc. The indeterminate load requires the interference fits for both bearing rings (fixed). Under these conditions in most application cases it is necessary to select bearings with a greater radial clearance.

Load Magnitude
The heavier the load is, the larger interference fit is selected, especially in the case of impact loads. The interference fit on the shaft or in the housing causes ring deformation, and thereby reduces the radial clearance in the bearing. The resulting clearance after mounting differs according to the bearing types and sizes. In some cases, due to interference fitting, bearings with a greater radial clearance must be used.

Bearing Size and Type
The extent of the mounted ring interference depends on the bearing size and type. Smaller interferences are selected for bearings of smaller sizes and conversely. Relatively smaller interferences are used, e.g. for ball bearings of the same sizes in comparison to cylindrical roller, tapered roller or spherical roller bearings.

Material and Design of Connecting Components
Recommended tolerances for mounting the rings shown in the following tables are valid for solid steel shafts and housings made of steel, alloy or cast steel. If the bearings are mounted in a light alloy housing or journals with a hollow, arrangements with higher interferences are selected.
Split housings are not suitable for arrangements with great interference fits because bearing pinching in the dividing plane can occur.

Temperature Effect
The heat generated in the bearing can make the interference on the shaft loose causing the ring to turn. In the housing a higher interference can occur, limiting the displacement ability of the non-locating bearing outer ring.

Fitting Accuracy
Surface deviations under the bearing seating are transmitted to the bearing raceways and decrease the arrangement accuracy.
When using bearings with standard tolerance class, the tolerance class IT6 for the seating surface on the shaft and the tolerance class IT7 for the housing seating surface are selected.
When higher bearing tolerance class is used, there are also higher requirements on the shape accuracy of the seating surfaces. Recommended values of the standard tolerances IT for selecting the seating surface shape for bearings are in the Table 27 and the values of respective standard tolerances IT are in the Table 28.
The quality of the bearing ring location is influenced not only by the dimension and shape accuracy, but also by the roughness of the seating surfaces. Recommended values of roughness are shown in the Table 26.

Recommended Roughness of Seating Surfaces

Table 26

Diameter of Seating Surface		Bearing Tolerance Class	
		P0	P6. P5
over	to	Roughness R_a	
mm		µm	
-	250	0.8	0.4
250	500	1.6	0.8
500	1250	3.2	-

Recommended Shape Accuracies of Seating Surfaces for Bearings

Table 27

Bearing Tolerance Class	Fitting Location	Permissible Ovality Deviation	Permissible Lateral Runout of Carrying Surfaces in reference to Axis
P0, P6	shaft	$\frac{IT5}{2}$	IT3
	housing	$\frac{IT6}{2}$	IT4
P5, P4	shaft	$\frac{IT3}{2}$	IT2
	housing	$\frac{IT4}{2}$	IT3

Standard Tolerances IT2 to IT7

Table 28

Nominal Diameter		Tolerance Class					
over	to	IT2	IT3	IT4	IT5	IT6	IT7
mm		µm					
30	50	2.5	4	7	11	16	25
50	80	3	5	8	13	19	30
80	120	4	6	10	15	22	35
120	180	5	8	12	18	25	40
180	250	7	10	14	20	29	46
250	315	8	12	16	23	32	52
315	400	9	13	18	25	36	57
400	500	10	15	20	27	40	63
500	630	-	-	-	29	44	70
630	800	-	-	-	32	50	80
800	1000	-	-	-	36	56	90
1000	1250	-	-	-	42	66	105
1250	1600	-	-	-	50	78	125
1600	2000	-	-	-	60	92	150
2000	2500	-	-	-	70	110	175

Mounting and Dismounting Methods

When one of the bearing rings is fitted with a loose fit, mounting and dismounting are simple. If it is necessary to fit both rings with interference, it is necessary to select a suitable bearing type, e.g. a separable bearing (cylindrical roller, needle roller, tapered roller) or a bearing with a tapered bore.

Bearings with a tapered bore are mounted directly on the tapered shaft, or are fastened on the shaft by means of adapter or withdrawal sleeves. The journals for fitting the sleeves can be in the tolerance h9, or h10, the geometrical shape must be, however, in the tolerance class IT5, or IT7, according to the requirements.

Axial Displaceability of Non-Locating Bearing Rings

One of the non-locating bearing rings should be displaceable in the axial direction under all operating conditions. In non-separable bearings, the displaceability of the stationary loaded ring is reached by its fitting with a clearance (moveable).

In the light metal alloy housings it is necessary, in case of fitting the outer ring with a clearance, to put a steel bush in the bore. Reliable displaceability in the axial direction is reached by using the cylindrical roller bearings - types N and NU.

The required type of fitting is reached by selecting the tolerance of the shaft and housing bore. Tables 29 to 32 show the recommended diameter tolerances of the shafts and housing bores for radial and thrust bearings indicated in the dimension tables of this publication. Values of the recommended tolerance limiting deviations are shown in the Tables 33 and 34.

Shaft Diameter Tolerance for Radial Bearings (valid for solid steel shafts) Table 29

Operating Conditions	Arrangement Examples	Shaft Diameter [mm]			Tolerance
		Ball Bearings	Cylindrical Roller and Tapered Roller Bearings	Spherical Roller Bearings	
Inner Ring Stationary Load					
Small and normal load $P_r \leq 0.15\ C_r$	Free wheels, sheaves, pulleys	All Diameters			g6[1]
Heavy impact load $P_r > 0.15\ C_r$	Truck wheels, tension pulleys				h6
Inner Ring Circumferential Load or Indeterminite Load					
Small and variable load $P_r \leq 0.07\ C_r$	Conveyors, fans	(18) to 100 (100) to 200	≤40 (40) to 140	- - - -	j6 k6
Normal and heavy load $P_r > 0.07\ C_r$	General engineering, electric motors, turbines, pumps, combustion motors, gearboxes, woodworking machines	(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	k5(k6)[2] m5(m6)[2] m6 n6 p6
Especially heavy load, impacts, difficult operating conditions $P_r > 0.15\ C_r$		- - -	(50) to 140 (140) to 500 > 500	(50) to 100 100) to 500 > 500	n6[3] p6[3] r6(p6)[3][4]
High arrangement accuracy at small load $P_r \leq 0.07\ C_r$	Machine tools	(18) to 100 (100) to 200	≤40 (40) to 100 (140) to 200	-	j5 k5 m5
Only axial load		All Diameters			j6
Bearings with Tapered Bore and Adapter or Withdrawal Sleeve					
All kinds of load	All arrangaments, axle bearing of railway vehicles	All Diameters			h9I/T5
	Simple arrangements				h10/IT7

[1] For large sized bearings it is possible to select the f6 tolerance so that axial dispalceability can be secured.
[2] Tolerances in brackets are selected as a rule for single row tapered roller bearings or at low rotational speeds where the tolerance dispersion is not significant.
[3] It is necessary to use bearings with higher radial clearance than normal.
[4] Tolerance e7 is selected for a loose arrangement on the shaft rolls of the rolling mills.

Housing Bore Diameter Tolerances for Radial Bearings
(valid for housings made of steel, cast iron and cast steel)

Table 30

Operating Conditions	Outer Ring Displaceability	Housing	Arrangement Examples	Tolerance
Outer Ring Circumferential Load				
Heavy impact load $P_r > 0.15C_r$ thin walled housings	Not displaceable	One-part	Wheel hubs with cylindrical roller bearings, big end bearings	P7
Normal and heavy load $P_r > 0.07C_r$			Wheel hubs with ball bearings, crane travel wheels, crankshaft bearings	N7
Small and variable load $P_r \leq 0.07C_r$			Conveyor rollers, tension pulleys	M7
Indeterminite Load				
Heavy impact load $Pr > 0.15C_r$	Not displaceable	One-part	Traction motors	M7
Normal and heavy load $P_r > 0.07C_r$	Not displaceable as a rule		Electric motors, pumps, crankshafts	K7
Small and variable load $P_r \leq 0.7C_r$	Displaceable as a rule		Electric motors, fans, crankshafts, pumps	J7
Accurate Arrangement				
Small load $P_r \leq 0.7C_r$	Not displaceable as a rule	One-part	Cylindrical roller bearings for machine tools	K6 [1)]
	Displaceable		Ball bearings for machine tools	J6
	Easily displaceable		Small electric motors	H6
Outer Ring Stationary Load				
Any load	Easily displaceable	One-part or two-part	General engineering, axle bearings of railway vehicles	H7 [2)]
Small and Normal Load $P_r \leq 0.15C_r$			General engineering, simpler arrangements	H8
			Drying rollers of paperworking machines, big electric motors	G7 [3)]

[1)] For heavy loads tighter tolerances M6 or N6 are selected. Tolerances K5 or M5 are selected for cylindrical roller bearings with a tapered bore.
[2)] For bearings with outer diameter D < 250mm with the temperature gradient between the outer ring and the housing over 10°C the tolerance G7 is selected.
[3)] For bearings with outer diameter D > 250mm with the temperature gradient between the outer ring and the housing over 10°C the tolerance F7 is selected.

Shaft diameter tolerances for tapered roller bearings in inch dimensions - tolerance class - 4 [mm] Table 29a

Bearing Bore			Shaft Diameter Deviations					
Range		Tolerance	Rotating Shaft	Rotating Shaft	Stationary Shaft			
			Ground, Constant Loads with Moderate Shock	Ground, or Unground Heavy Loads High Speed or Shock	Unground, Moderate Loads No Shock	Ground, Moderate Loads No Shock	Unground, Sheaves, Wheels, Idlers	Hardened and Ground, Wheel Spindles
Over	Inclusive							
0	76.2	0 +0.013	+0.038 +0.025	+0.064 +0.038	+0.013 0	0 -0.013	0 -0.013	-0.005 -0.018
76.2	304.8	0 +0.025	+0.064 +0.038	See note	+0.025 0	0 -0.025	0 -0.025	-0.005 -0.030
304.8	609.6	0 +0.051	+0.127 +0.076		+0.051 0	0 -0.051	0 -0.051	- -
609.6	723.9	0 +0.076	+0.190 +0.114		+0.076 0	0 -0.076	0 -0.076	- -

Note: Use an average interference fit of 0.0005 mm / mm of bearing bore.
Minimum interference fit 0.025 mm for bearings with a bore between 76.2 mm and 101.6 mm.

Housing bore tolerances for tapered roller bearings in inch dimensions - tolerance class - 4 [mm] Table 30a

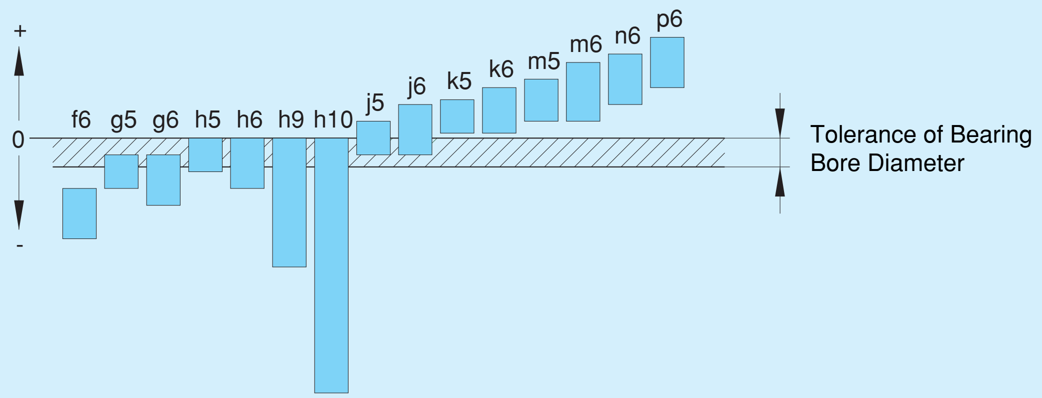
Bearing Outer Diameter			Housing Bore Deviations			
Range		Tolerance	Stationary Housing	Stationary Housing	Stationary or Rotating Housing	Rotating Housing
			Floating or Axially Clamped Race	Adjustable (Axially Displaceable Race)	Stationary Race (Nonadjustable)	Sheave, Axially Unclamped Race
Over	Inclusive					
0	76.2	+0.025 0	+0.051 +0.076	0 +0.025	-0.038 -0.013	-0.076 -0.051
76.2	127.0	+0.025 0	+0.051 +0.076	0 +0.025	-0.051 -0.025	-0.076 -0.051
127.0	304.8	+0.025 0	+0.051 +0.076	0 +0.051	-0.051 -0.025	-0.076 -0.051
304.8	609.6	+0.051 0	+0.102 +0.152	+0.026 +0.076	-0.076 -0.025	-0.102 -0.051
609.6	914.4	+0.076 0	+0.152 +0.229	+0.051 +0.127	-0.102 -0.025	- -

Shaft Diameter Tolerances for Thrust Bearings Table 31

Bearing Type	Type of Load	Shaft Diameter [mm]	Tolerance
Thrust Ball Bearing - single direction - double direction	Axial Load Only	All Diameters	 j6 k6 (j6)
Thrust Cylindrical Roller and Tapered Roller Bearings			h6

Housing Bore Diameter Tolerances for Thrust Bearings Table 32

Bearing Type	Type of Load	Note	Tolerance
Thrust Ball, Cylindrical Roller and Tapered Roller Bearings	Axial Load Only	For common arrangements the housing washer can have clearance	H8
		The housing washer is mounted with radial clearance	-



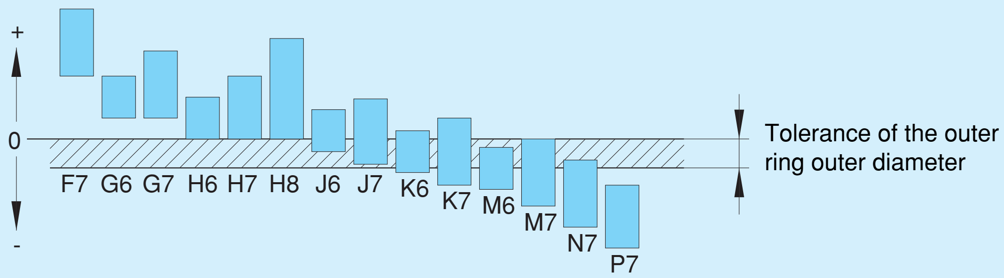
Tolerance Limiting Deviations of Journal Diameters

Table 33

Journal Nominal Diameter over	to	f6 upper	lower	g5 upper	lower	g6 upper	lower	h5 upper	lower	h6 upper	lower	h9[1)] upper	lower	h10[1)] upper	lower
mm		µm													
50	80	- 30	- 49	-10	-23	-10	- 29	0	-13	0	-19	0	- 74	0	-120
80	120	- 36	- 58	-12	-27	-12	- 34	0	-15	0	-22	0	- 87	0	-140
120	180	- 43	- 68	-14	-32	-14	- 39	0	-18	0	-25	0	-100	0	-160
180	250	- 50	- 79	-15	-35	-15	- 44	0	-20	0	-29	0	-115	0	-185
250	315	- 56	- 88	-17	-40	-17	- 49	0	-23	0	-32	0	-130	0	-210
315	400	- 62	- 98	-18	-43	-18	- 54	0	-25	0	-36	0	-140	0	-230
400	500	- 68	-108	-20	-47	-20	- 60	0	-27	0	-40	0	-155	0	-250
500	630	- 76	-120	-22	-50	-22	- 66	0	-28	0	-44	0	-175	0	-280
630	800	- 80	-130	-24	-56	-24	- 74	0	-32	0	-50	0	-200	0	-320
800	1 000	- 86	-142	-26	-62	-26	- 82	0	-36	0	-56	0	-230	0	-360
1 000	1 250	- 98	-164	-28	-70	-28	- 94	0	-42	0	-66	0	-260	0	-420
1 250	1 600	-110	-188	-30	-80	-30	-108	0	-50	0	-78	0	-310	0	-500
1 600	2 000	-120	-212	-32	-92	-32	-124	0	-60	0	-92	0	-370	0	-600

Journal Nominal Diameter over	to	j5 upper	lower	j6 (js6) upper	lower	k5 upper	lower	k6 upper	lower	m5 upper	lower	m6 upper	lower	n6 upper	lower	p6 upper	lower
mm		µm															
50	80	+6	-7	+12	-7	+15	+2	+21	+2	+24	+11	+30	+11	+39	+20	+51	+32
80	120	+6	-9	+13	-9	+18	+3	+25	+3	+28	+13	+35	+13	+45	+23	+59	+37
120	180	+7	-11	+14	-11	+21	+3	+28	+3	+33	+15	+40	+15	+52	+27	+68	+43
180	250	+7	-13	+16	-13	+24	+4	+33	+4	+37	+17	+46	+17	+60	+31	+79	+50
250	315	+7	-16	+16	-16	+27	+4	+36	+4	+43	+20	+52	+20	+66	+34	+88	+56
315	400	+7	-18	+18	-18	+29	+4	+40	+4	+46	+21	+57	+21	+73	+37	+98	+62
400	500	+7	-20	+20	-20	+32	+5	+45	+5	+50	+23	+63	+23	+80	+40	+108	+68
500	630	-	-	+22	-22	-	-	+44	0	-	-	+70	+26	+88	+44	+122	+78
630	800	-	-	+25	-25	-	-	+50	0	-	-	+80	+30	+100	+50	+138	+88
800	1 000	-	-	+28	-28	-	-	+56	0	-	-	+90	+34	+112	+56	+156	+100
1 000	1 250	-	-	+33	-33	-	-	+66	0	-	-	+106	+40	+132	+6	+186	+120
1 250	1 600	-	-	-39	-39	-	-	+78	0	-	-	+126	+48	+156	+78	+218	+140
1 600	2 000	-	-	+46	-46	-	-	+92	0	-	-	+150	+58	+184	+92	+262	+170

1) For journals manufactured in the tolerances h9 and h10 for bearings with adapter or withdrawal sleeve the deviations of roudness and cylindricity must not exceed the basic tolerance IT5 and IT7.



Tolerance Limiting Deviations of Bore Diameters

Table 34

Nominal Bore Diameter over	to	F7 upper	lower	G6 upper	lower	G7 upper	lower	H6 upper	lower	H7 upper	lower	H8 upper	lower	J6 (J_S6) upper	lower
mm		µm													
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	1 000	+176	+ 86	+ 82	+26	+116	+26	+56	0	+ 90	0	+140	0	+28	-28
1 000	1 250	+203	+ 98	+ 94	+28	+133	+28	+66	0	+105	0	+165	0	+33	-33
1 250	1 600	+235	+110	+108	+30	+155	+30	+78	0	+125	0	+195	0	+39	-39
1 600	2 000	+270	+120	+124	+32	+182	+32	+92	0	+150	0	+230	0	+46	-46
2 000	2 500	+305	+130	+144	+34	+209	+34	+110	0	+175	0	+280	0	+55	-55

Nominal Bore Diameter over	to	J7 (J_S7) upper	lower	K6 upper	lower	K7 upper	lower	M6 upper	lower	M7 upper	lower	N7 upper	lower	P7 upper	lower
mm		µm													
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	1 000	+45	-45	0	-56	0	- 90	-34	- 90	-34	-124	- 56	-146	-100	-190
1 000	1 250	+52	-52	0	-66	0	-105	-40	-106	-40	-145	- 66	-171	-120	-225
1 250	1 600	+62	-62	0	-78	0	-125	-48	-126	-48	-173	- 78	-203	-140	-265
1 600	2 000	+75	-75	0	-92	0	-150	-58	-150	-58	-208	- 92	-242	-170	-320
2 000	2 500	+87	-87	0	-110	0	-175	-68	-178	-68	-243	-110	-285	-195	-370

3.2.2 Axial Location of Bearing Rings

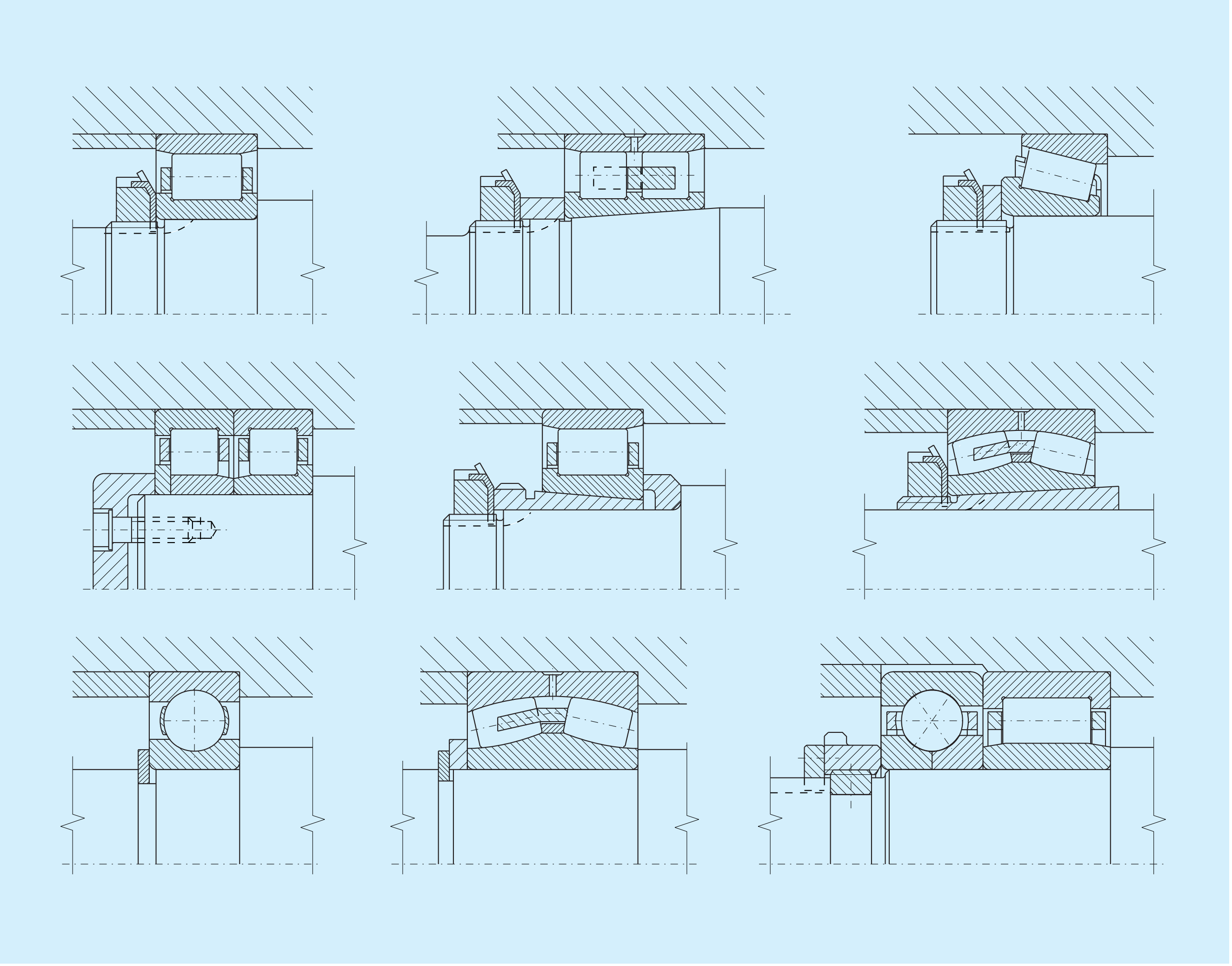
Inner rings of the bearings with a cylindrical bore, located on shafts with an interference fit (fixed) are usually located in the axial direction by a locknut with a lockwasher, a plate or a snap ring and the other face is usually supported by the shaft shoulder. Surrounding parts are used as abutment faces for the inner rings and if it is necessary, spacing rings are inserted between this component and the bearing inner ring. Figure 13 shows some most common cases of the location.

The permissible axial load of bearings fixed by an adapter sleeve on smooth shafts without bearing abutment on the shaft shoulder is calculated according to the following equation:

$$F_a = 3 \, . \, B \, . \, d$$

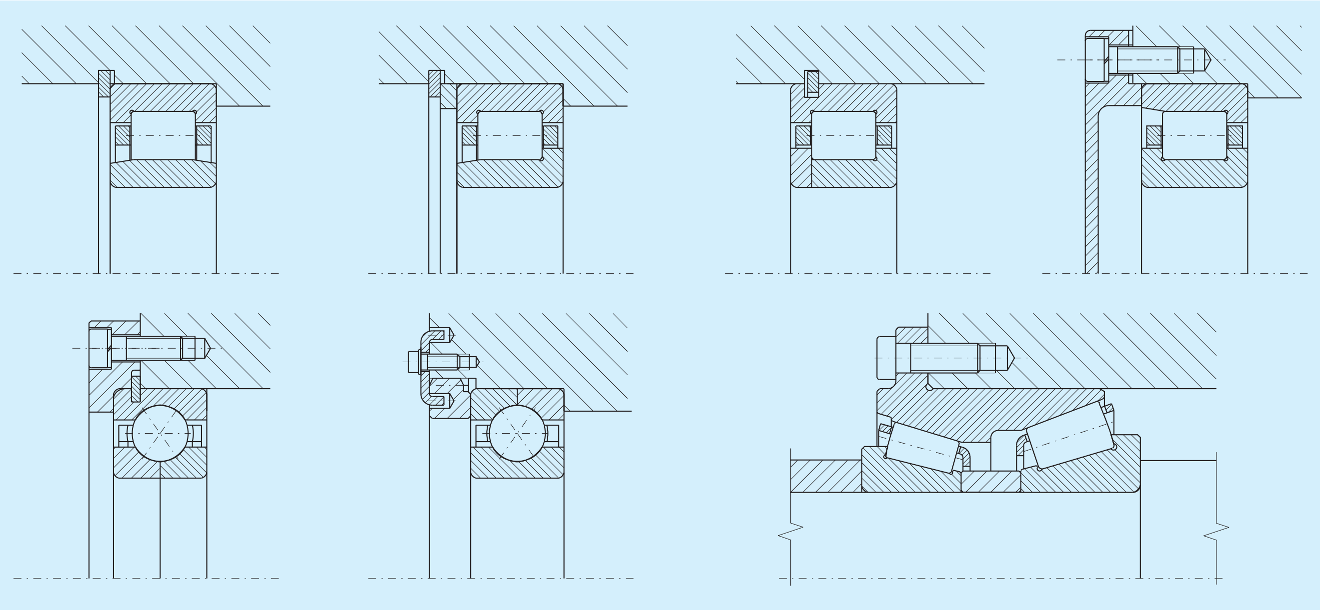
where: F_a	- permissible axial load of the bearing	[N]
B	- bearing width	[mm]
d	- bearing bore diameter	[mm]

Figure 13



The axial displacement of the outer bearing rings in the housing, if not required, is limited by the supporting parts of covers, nuts or snap rings. Bearings with a groove and a snap ring (NR) do not require much space and their locating is simple. Common locating methods - see Figure 14.

Figure 14



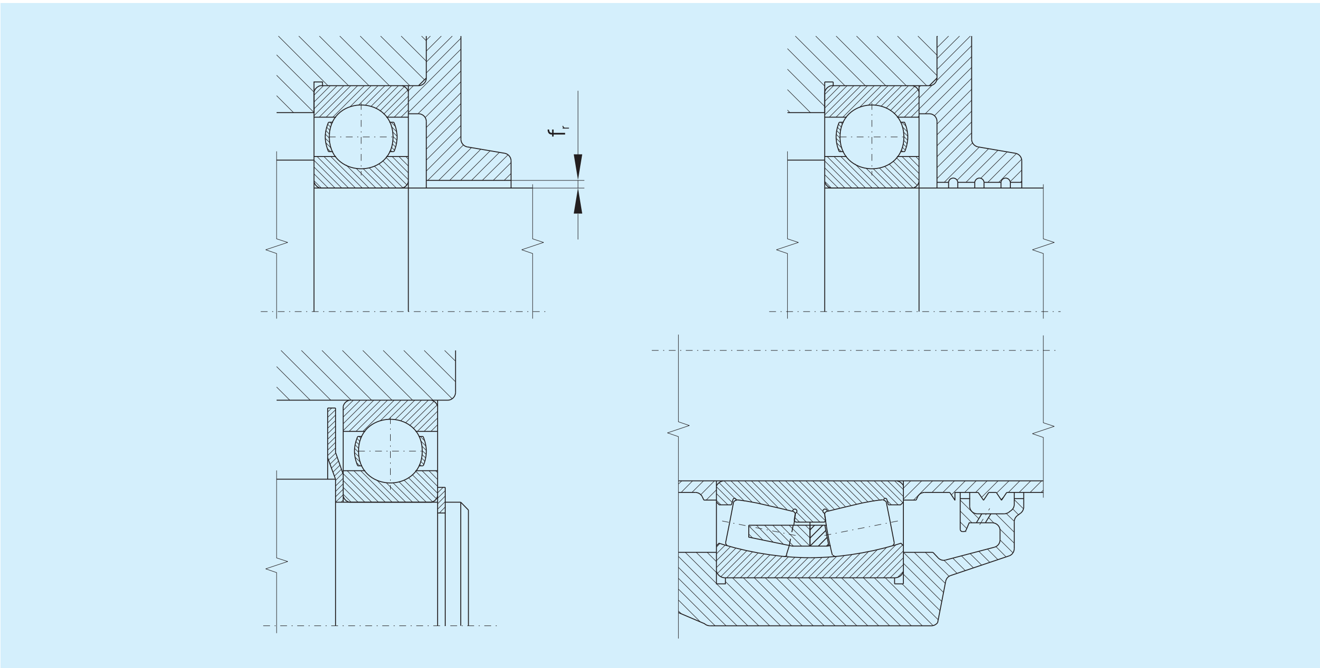
3.3 Sealing

To reach reliable operation of the arrangement during the whole life, the sealing effectivness of the arrangement plays a decisive role. The main task of the sealing is to prevent penetration of impurities into the arrangement and the bearing lubricant.
The sealing of the rolling bearings can be divided into three main groups:
- Non-contact sealing
- Contact sealing
- Combined sealing

3.3.1 Non-Contact Sealing

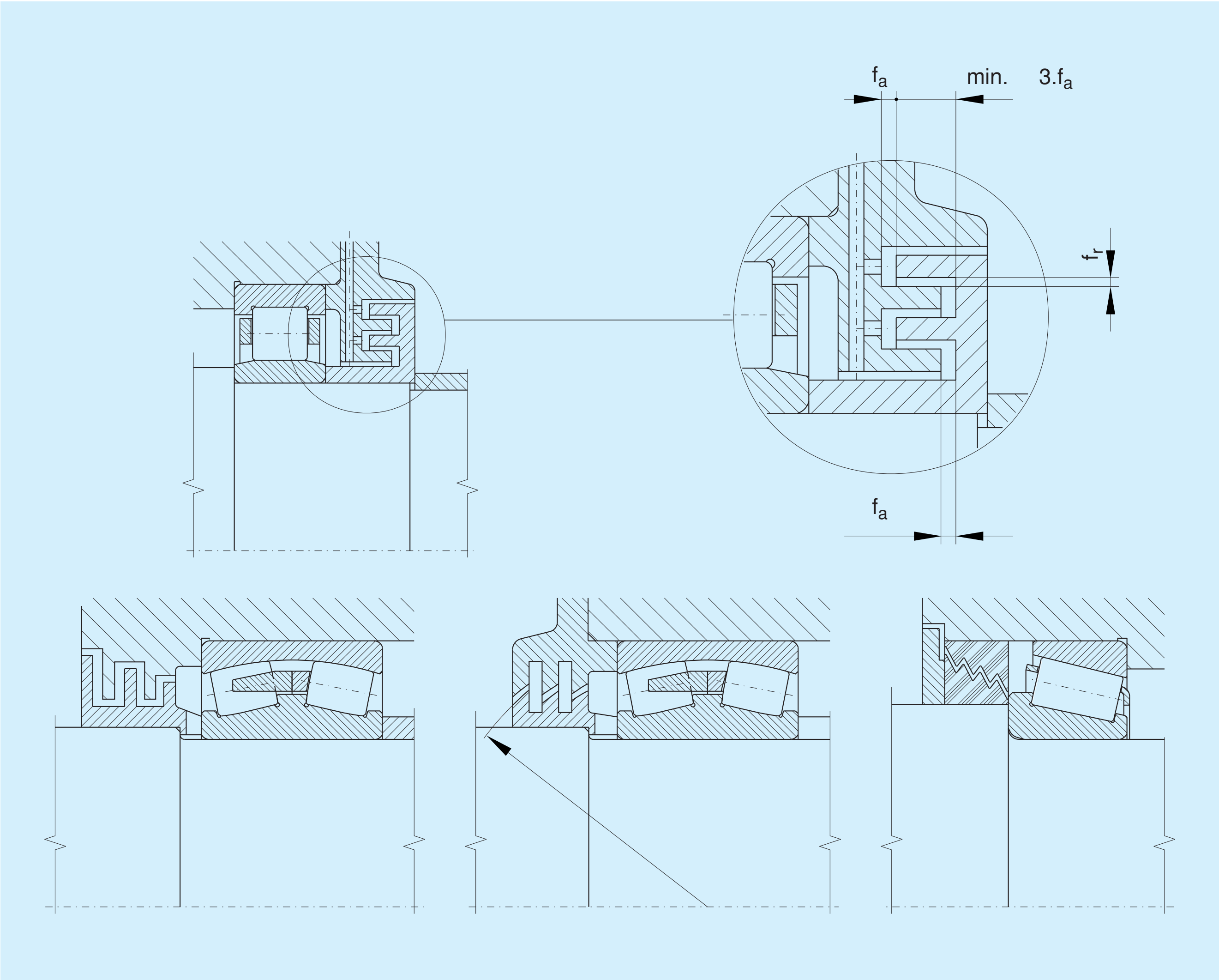
When using this sealing method, only a small gap is left between the rotating and non-rotating parts which is sometimes filled with grease. No wear due to friction can rise with this sealing and that is why this sealing can be used for the highest rotational speeds and it is also suitable for high operating temperatures. Various types of the gap sealing are shown in Figure 15.

Figure 15



Another very efficient sealing is the labyrinth sealing. It can increase the sealing effect by a higher number of labyrinths or by making the sealing gap longer.
Figure 16 shows some most common designs of this sealing.

Figure 16



Recommended sizes of the sealing gaps in the radial direction (f_r) are shown in the Table 35.
The size of the gap in the axial direction (f_a) is created to enable the displaceability of the non-locating bearing.

Size of Sealing Gap

Table 35

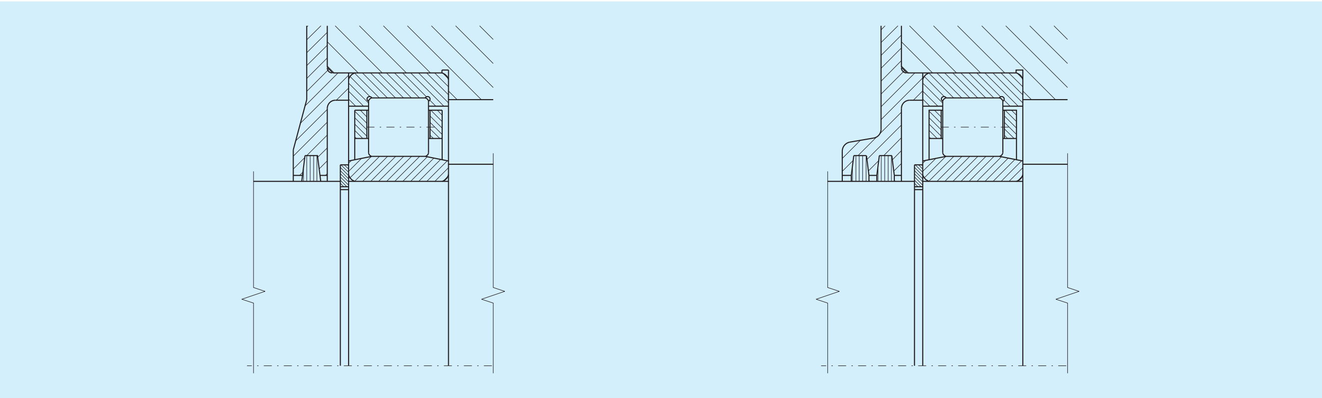
Nominal Diameter over	to	Size of Sealing Gap in Radial Direction fr ±0.2
mm		
–	120	0.7
120	180	0.8
180	250	0.9
250	315	1.0
315	400	1.1
400	500	1.2
500	630	1.3
630	800	1.5
800	1000	1.7
1000	1250	1.9
1250	1600	2.3

3.3.2 Contact Sealing

Contact sealing is created by an elastic or soft but sufficiently solid and impremiable material inserted between the rotating and non-rotating part. The contact sealing is simple and not expensive and is suitable for various designs. Its disadvantage is the sliding friction of contacting surfaces and thereby limiting application for high peripheral speeds.

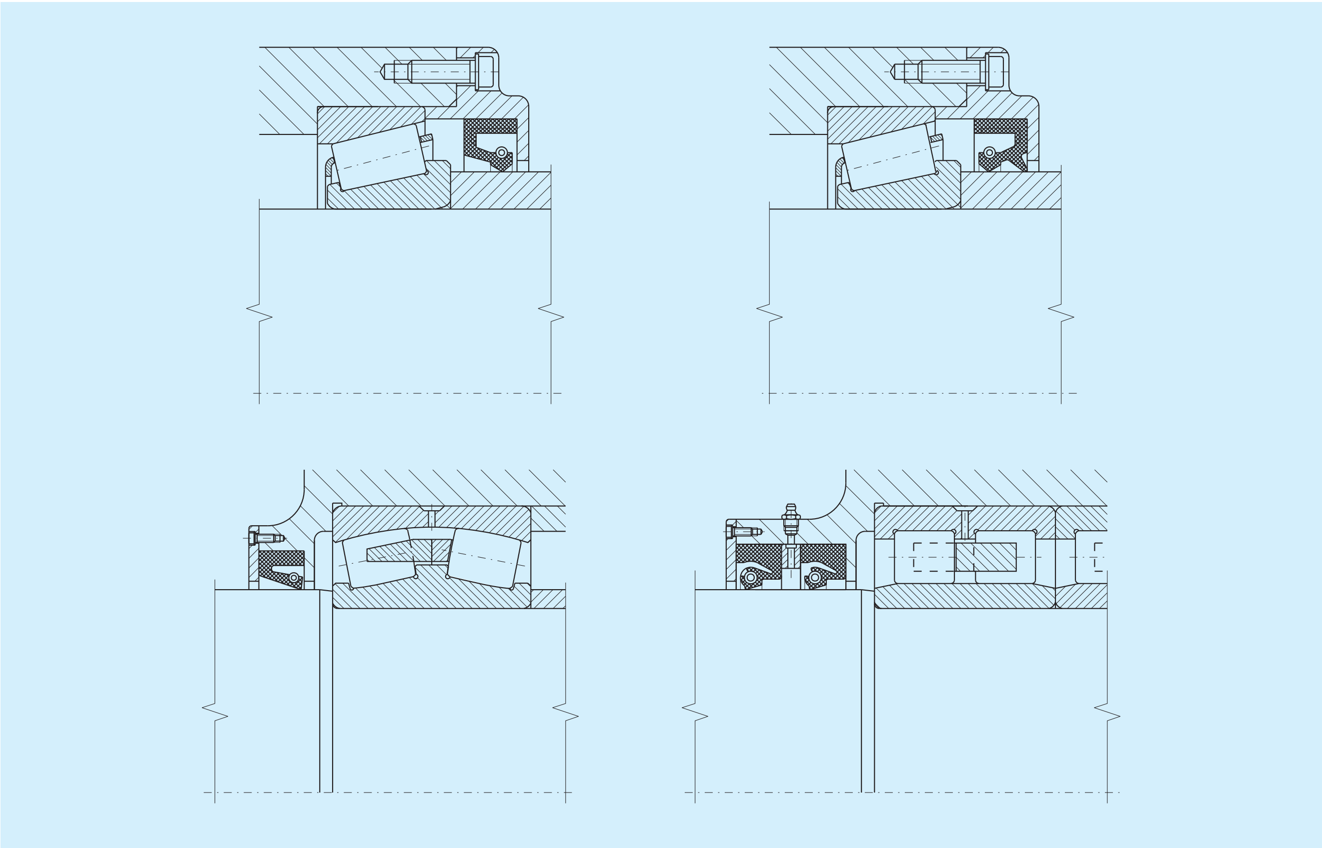
The sealing with felt rings (Figure 17) is the simplest one. It is suitable for operating temperatures-40 to +80°C and peripheral speeds to 7 $m.s^{-1}$ (the sliding surface roughness max. R_a = 0.16), hardness min. 45 HRc or hard chromium-plated surface.

Figure 17



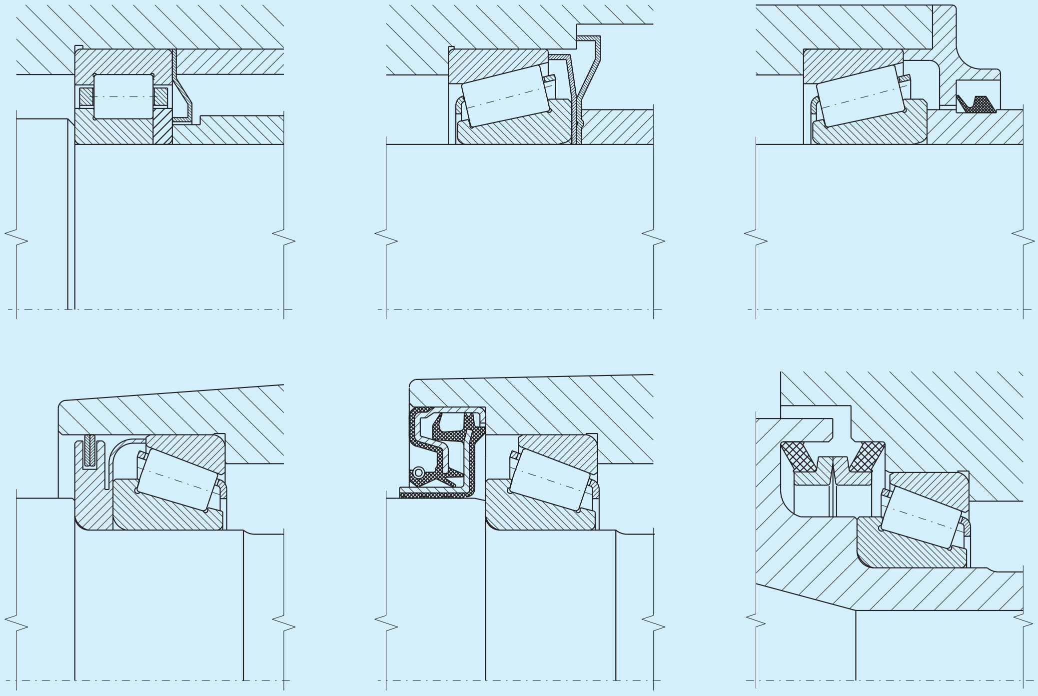
Another widely utilized sealing method is the sealing with a radial lip-type seal. These are made of synthetic rubber and are reinforced by metal reinforcements (Figure 18). Shaft rings of this design can be used for operating temperatures from -3 to + 100°C and peripheral speeds up to 2 $m.s^{-1}$ (the sliding surface roughness max. R_a = 0.8), ... up to 4 $m.s^{-1}$ (max R_a = 0.4) and up to 12 $m.s^{-1}$ (max R_a = 0.2)
The shaft rings made of special rubber can be used in the environment with temperatures -65 to 180°C up to the maximum peripheral speed 35 $m.s^{-1}$.

Figure 18



Except for the above mentioned most common sealing rings there are other contact seal designs which use specially formed sealing rings made of rubber, plastic or special elastic rings. This sealing is selected either for arrangement with high requirements on the protection of the bearing space (polluted environment, high temperature, influence of chemical substances), or for economic reasons by the mass or batch production.
Figure 19 shows some examples of this sealing.

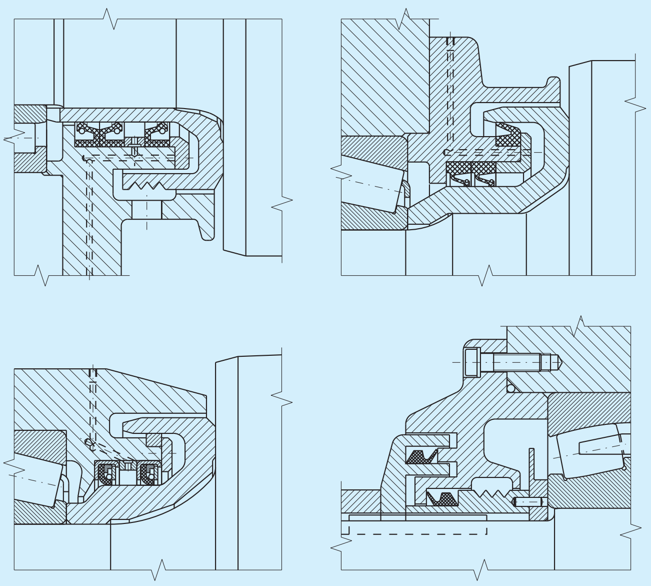
Figure 19



3.3.3 Combined Sealing

An increased sealing effect is achieved by the combination of both the contact and non-contact sealing. This sealing is used in environments with high content of impurities and moisture, e.g. rollers in rolling mills.
Figure 20 shows some examples of the combined sealing.

Figure 20



4. LUBRICATION OF ROLLING BEARINGS

The correct rolling bearing lubrication is as important as its correct selection because it can decisively influence the life of bearings. The lubricant forms a carrying lubrication film on the functional bearing surfaces, which hinders the metal contact of the rolling elements with the ring raceways. It lubricates the surfaces with sliding friction, has a cooling effect, protects the bearing from corrosion and in many cases seals the bearing space.

Rolling bearings are mostly lubricated with grease or oil, exceptionally with different lubricants. When deciding which kind of lubricant and method of lubrication is to be used, we must take into account the operating conditions, characteristic properties of the lubricant and the design of the whole machine or equipment, as well as economy and reliability of its operation.

4.1 Grease Lubrication

Grease lubrication compared with the oil lubrication has many advantages, that is why it is preferred where it can be used. The arrangement design of the bearings lubricated with grease is usually simple, also the sealing properties of the lubricant can be used and the maintenance is simpler.

An analysis of all requirements on the arrangement should be carried out before selecting a suitable grease and according to the priority of individual criteria the suitable grease is selected.

The main criteria include :

- ratio of the dynamic load rating to the equivalent load (C/P)
- arrangement high-speed (product $n.d_s$)
- requirements of the operation properties (friction moment, noise...)
- influence of the bearing arrangement (arrangement position, influence of centrifugal forces,etc.)
- influence of the operation environment (operating temperature, dustiness, vibrations, radioactivity, etc.)
- requirements of maintenance (re-lubrication interval,...)

4.1.1 Selection of Grease with Regard to Load and Rotational Speed

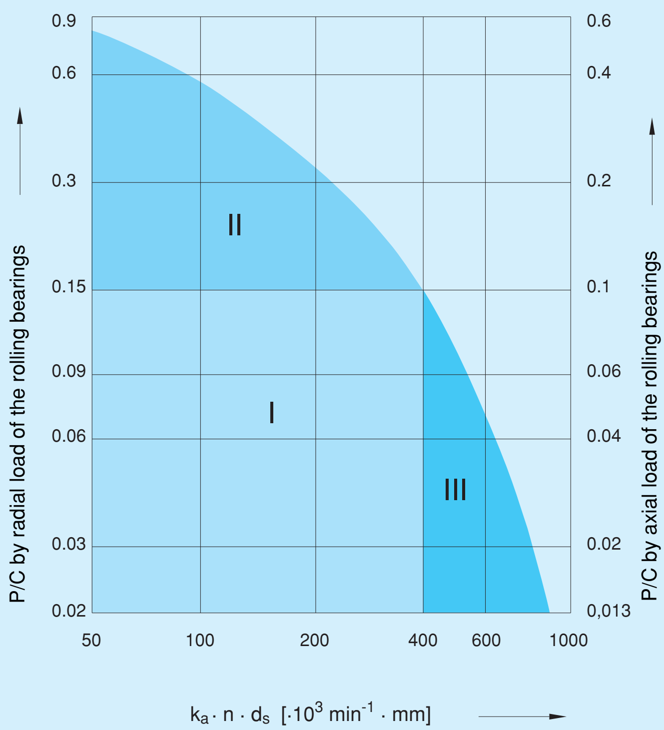
The specific lubricant load is: $P \geq 0.15.C$ - for radial bearings
$P \geq 0.1.C$ - for thrust bearings

where: P - dynamic equivalent bearing load [kN]
C - basic dynamic load rating [kN]

The high-speed ratio for common greases is $n.d_s < 0.5.10^6$ mm.min^{-1}. When using special lubricants, the high-speed ratio can be up to $n.d_s < 1.3.10^6$ mm.min^{-1}.

Selection of a suitable lubricant with regard to the load and rotational speed can be done according to the diagram in Figure 21.

Figure 21



Range I.

- normal operational range
- greases for general use with basic oil viscosity ISO VG 46-150

Range II.

- range of higher load
- greases with EP additives with basic oil viscosity CG 150 and more

Range III.

- range of higher rotational speeds
- greases for high-speed bearings with basic oil viscosity ISO VG 10-46 and high adhesivity

k_a = 1 - ball bearings with a four-point contact, radially loaded cylindrical roller bearings, thrust ball bearings

k_a = 2 - self-aligning spherical roller bearings, tapered roller bearings

k_a = 3 - axially loaded cylindrical roller bearings, cylindrical roller bearings with a full complement of rolling elements

4.1.2 Greases for Rolling Bearings

Greases for rolling bearing lubrication are made of high quality mineral or synthetic oils (or with additives) thickened with fatty acid metallic soaps. The greases must have a good lubricating ability and a high chemical, temperature and mechanical stability. The Table 36 shows the survey of the rolling bearing lubricants.

Properties of Greases for Rolling Bearings Table 36

Kind of Grease		Properties		
Thickening Agent	Basic Oil	Operating Temperature Range [° C]	Resistance against Water	Usage
lithium soap	mineral	-20 ÷ 130	resistant	multi-purpose lubricant
lithium soap	ester	-60 ÷ 130	resistant	for low temperatures and high rotational speed
lithium soap	silicon	-40 ÷ 170	very resistant	suitable for wide temperature range at medium rotational speed
lime soap	mineral	-20 ÷ 50	very resistant	a good sealing effect against water
soda soap	mineral	-20 ÷ 100	not resistant	emulsifies with water
aluminium soap	mineral	-20 ÷ 70	resistant	a good sealing effect against water
complex lithium soap	mineral	-20 ÷ 150	resistant	multi-purpose lubricant
complex lime soap	mineral	-30 ÷ 130	very resistant	multi-purpose lubricant suitable for higher temperatures and load
complex soda soap	mineral	-20 ÷ 130	resistant	suitable for higher temperatures and load
complex aluminium soap	mineral	-20 ÷ 150	resistant	suitable for higher temperatures and load
complex barium soap	mineral	-30 ÷ 140	resistant	suitable for higher temperatures and load
complex barium soap	ester	-60 ÷ 140	resistant	suitable for higher temperatures and higher rotational speeds
bentonite	mineral	-20 ÷ 150	resistant	suitable for higher temperatures at low rotational speed
polyurea	mineral	-20 ÷ 160	resistant	suitable for hig temperatures at medium rotational speed

Selection of a suitable lubricant with regard to the consistency (degree of the lubricant formability) can be carried out according to the Table 37.

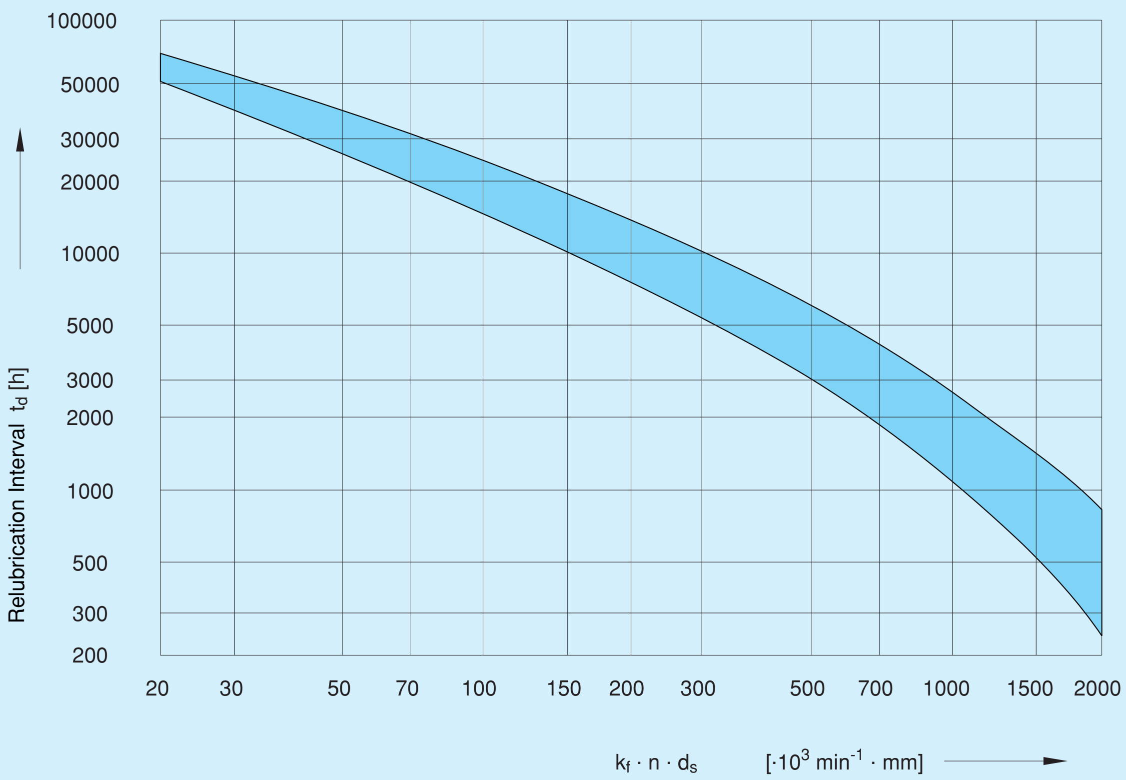
Table of Lubricant Selection with regard to Consistency Table 37

Consistency			Usage
NLGI Degree	Designation	Penetration	
0	very soft	355 ÷ 385	For central distribution systems, in cases of friction corrosion
1	soft	310 ÷ 340	For low temperatures, central distribution systems in case of friction corrosion
2	medium soft	265 ÷ 295	For general usage, as a permanent filling
3	medium	220 ÷ 250	For general usage, as a permanent filling, for higher temperatures, dusty environment
4	medium stiff	175 ÷ 205	For high temperatures, as sealing of labyrinths

4.1.3 Relubrication Interval and Lubrication Quantity for One Relubrication

The relubrication interval or exchange of the grease is necessary if because of unfavourable conditions (high temperature, impurities, agressive substances...) the lubricant loses its functional properties before the bearing life ends.
The relubrication period can be approximately calculated from the diagram in Figure 22. The diagram is valid under normal operating conditions ($P < 0.1.C$, $t < 70°C$, lubrication by normal lithium lubricant).

Figure 22



Bearing Type	k_f
Single row and double row cylindrical roller bearings	1.8...2.3
Cylindrical roller bearings with full complement of rolling elements	25
Tapered roller bearings	4
Double row spherical roller bearings	9,...12
Ball bearings with four-point contact	1.6
Thrust ball bearings	5,...6
Thrust cylindrical roller and tapered roller bearings	90

In unfavourable operating conditions the relubricating interval calculated from the diagram 22 should be corrected according to the following equation:

$$t_{dk} = t_d \,.\, k_1 \,.\, k_2 \,.\, k_3 \,.\, k_4 \,.\, k_5$$

where: k_1 - factor of humidity and dust influence
k_2 - factor of impact load and vibration influence
k_3 - factor of higher temperature influence
k_4 - factor of high load influence
k_5 - factor of the air flow through the bearing (lubricant aging)

Table of Operating Influence Factors Table 38

Influence	Factor				
	k_1	k_2	k_3	k_4	k_5
Mild	0.7 ÷ 0.9	0.7 ÷ 0.9	0.7 ÷ 0.9	0.7 ÷ 1.0 P=(0.1 ÷ 0.15).C	0.5 ÷ 0.7
Strong	0.4 ÷ 0.7	0.4 ÷ 0.7	0.4 ÷ 0.7	0.4 ÷ 0.7 P=(0.15 ÷ 0.25).C	0.1 ÷ 0.5
Very Strong	0.1 ÷ 0.4	0.1 ÷ 0.4	0.1 ÷ 0.4	0.1 ÷ 0.4 P=(0.25 ÷ 0.35).C	-

At the first mounting of the bearing approximately 30 to 50% of the bearing space is filled with grease. This prevents excessive over-filling that could cause temperature increase and bearing depreciation.
The arrangement design should enable the removal of the excessive lubricant from the bearing, i.e. the surroundig space around the bearing should be large enough, or the arrangement can be equipped with a so called grease escape valve.
The necessary quantity of grease for one relubricating is calculated from the following equation:

$$Q = 0.005\, DB$$

where: Q - grease quantity [g]
D - bearing outer diameter [mm]
B - bearing width [mm]

4.2 Oil Lubrication

Oil lubrication is used in cases when the rotational speed is so high that the relubricating interval for grease lubrication is too short. There are other reasons for oil lubrication, e.g. the necessity to transfer the heat generated by the friction or heat from the surrounding sources, or high operating temperature which does not allow lubrication by usual grease, or if the surrounding parts are already lubricated by oil (e.g. geared wheels in the gearboxes).
For oil lubricating the quantity of oil must be sufficient so that the lubrication can be secured at the start, as well as during operation. Oil excess increases its temperature and also the bearing temperature.
The oil feed into all bearings of the machine, device or equipment is secured by many methods of which, lubrication with the oil bath where the oil level reaches the height of the lowest rolling element center, oil-circualtion lubrication, jet lubrication, oil-mist lubrication, lubrication by the system oil - air, etc., are the most commonly used methods.

4.2.1 Selection of Suitable Oil

Mineral, or for extreme operation conditions, the synthetical oils are usually used for bearing lubrication.
The basic physical property of the lubricating oils is their kinematic viscosity (it decreases with increasing temperature). A suitable mineral oil viscosity ν_1 can be stated according to the diagram in Figure 23. If the operating temperature is known, the viscosity of a suitable mineral oil ν is determined from the diagram 24. It is necessary for calculating the ratio $\kappa = \nu / \nu_1$ (see chapter 1.2.3).

If $\kappa < 1$, the oil with the EP additives should be used, if $\kappa > 0.4$, the usage of the oil with EP additives is inevitable. By normal life requirements the value should be $\kappa = 1$.

We recommend:
κ = 2 for bearings with a higher share of the sliding friction (axially loaded cylindrical roller and tapered roller bearings, double row spherical roller bearings, thrust cylindrical roller and tapered roller bearings, etc.),
κ = 2.5 for low-speed bearings ($n.d_s < 10000$ mm. min^{-1}) and for large sized bearings.

In unusual operating conditions (low or high rotational speed, temperature or large load, etc.) help can be provided on request by the experts of the PSL Technical Consultancy Department (address see page 2).

Figure 23

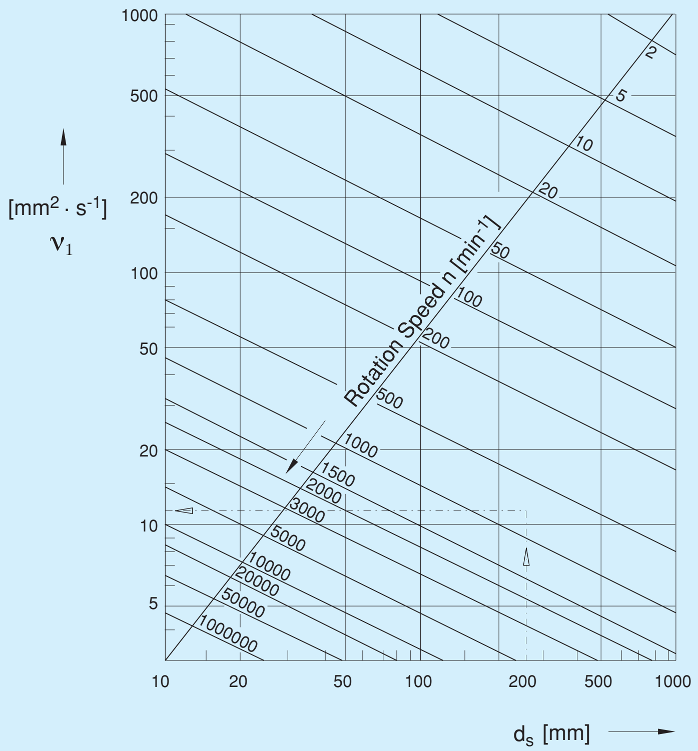
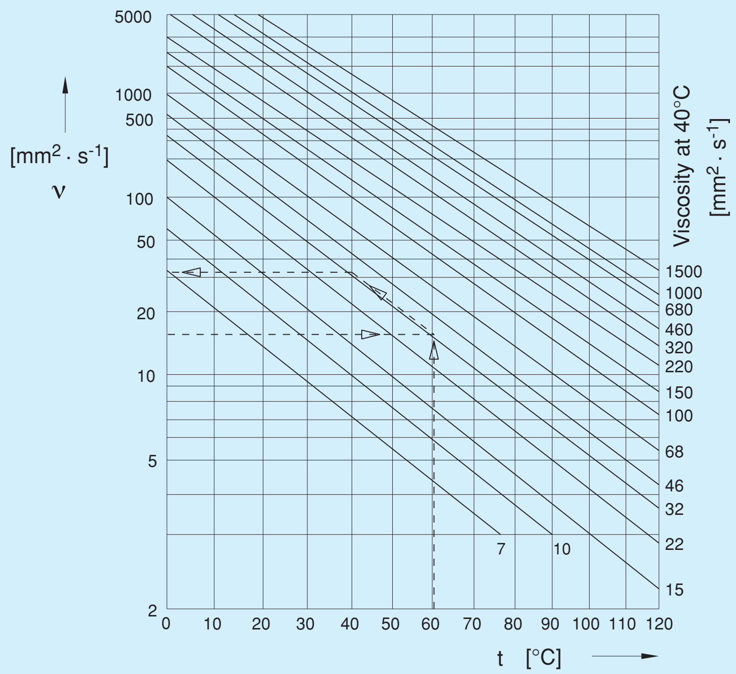


Figure 24

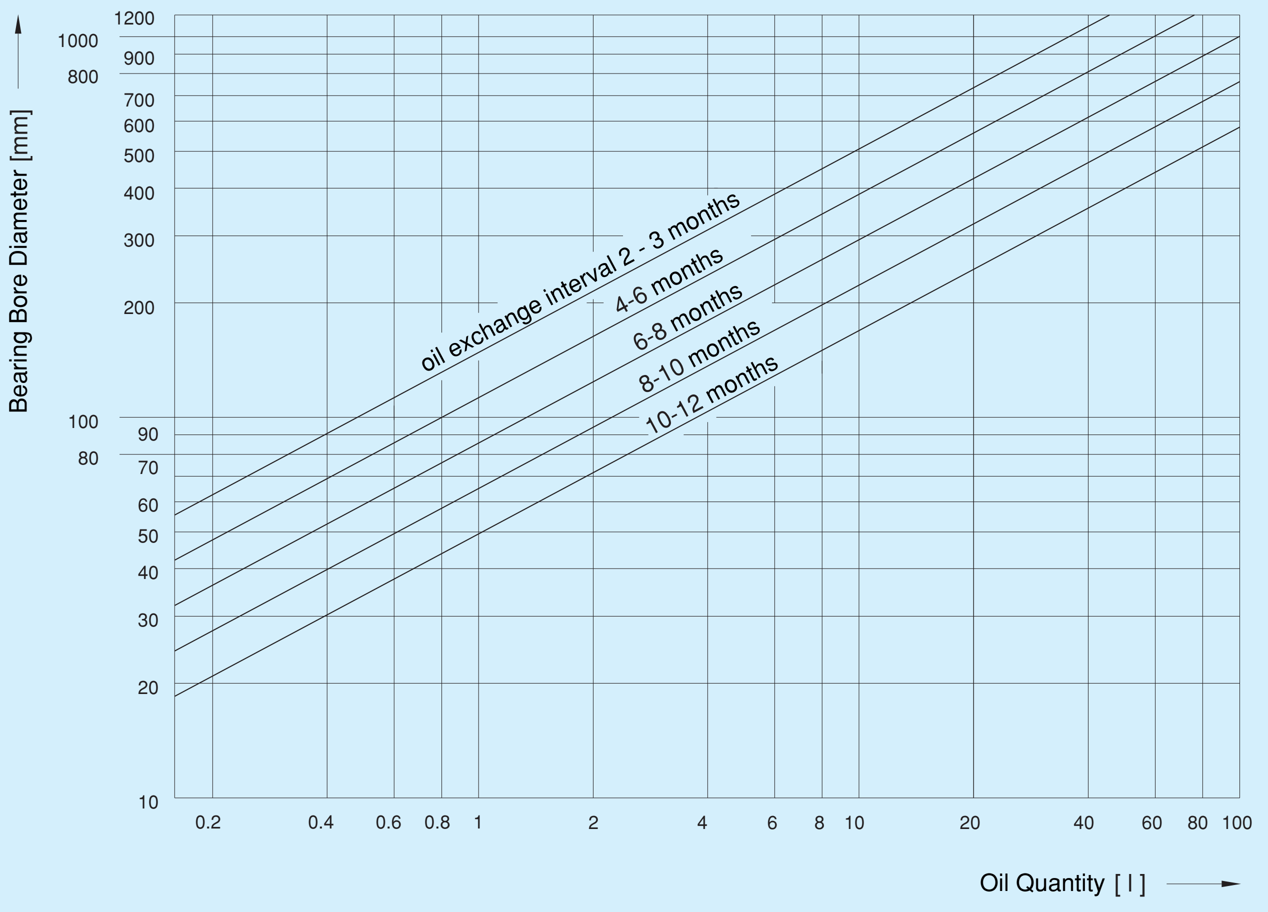


An example for stating the oil viscosity at 40°C.
A bearing with the bore diameter d = 180 mm and the outside diameter D = 320 mm has the operating rotational speed $n = 500.min^{-1}$.
According to the diagram in Figure 23 for the mean bearing diameter $d_S = (D+d)/2 = 250$ mm and the rotational speed $n = 500.min^{-1}$ the minimal kinematic oil viscosity is at the operating temperature $\nu_1 = 17 mm^2.s^{-1}$.
At the supposed operating temperature 60°C the oil at 40°C using the diagram in Figure 24 must have the kinematic viscosity minimally 35 $mm^2.s^{-1}$.

4.2.2 Quantity and Period of Oil Exchange

The oil exchange interval depends on the operating conditions, the oil quantity in the arrangement and the lubrication method.
When lubricating by oil bath the oil level should reach the height of the lowest rolling element center.
If the high-speed $n.d_S < 150\ 000\ min^{-1}.mm$, the oil level can also be higher. The high-speed treshold for oil bath lubrication is $n.d_S \leq 300\ 000\ min^{-1}.mm$, by often oil exchange up to $500\ 000\ min^{-1}.mm$.
The oil exchange interval at normal operating conditions should be according to the diagram in Figure 25.

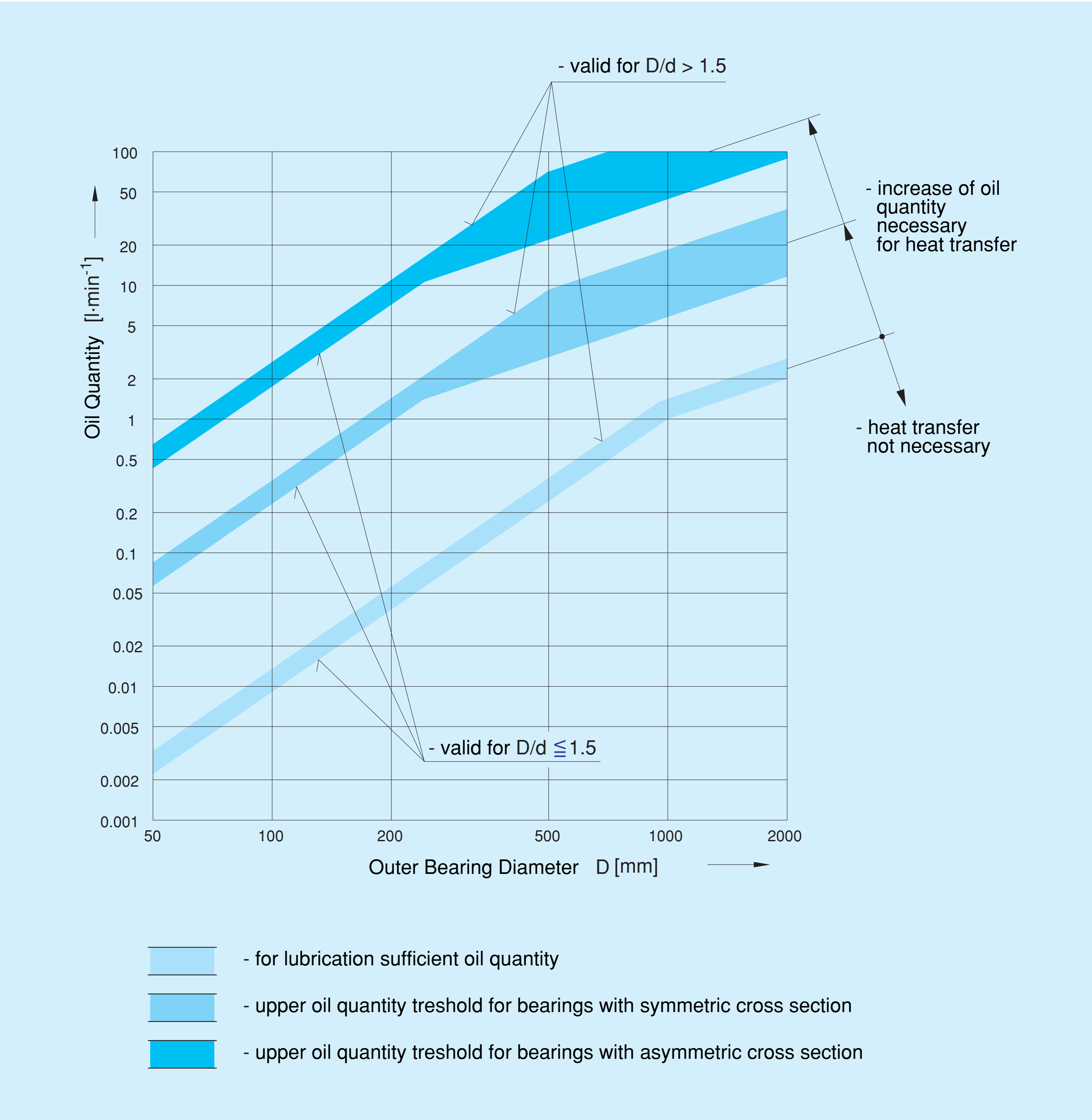
Figure 25



When lubricating by oil circulation the exchange interval depends on the oil quantity which passes through the lubricating system for a period of time, and also on the fact if it is cooled or not. The oil exchange interval can be determined during the testing period of the arrangement. This is valid also for lubrication by direct oil injecting into the bearing.

The oil quantity in the tank should by large enough to circulate 3 to 8 times an hour.
For orientation it can be read in dependence on the outer bearing diameter - see diagram in Figure 26.

Oil Quantity for Circulating Lubrication Figure 26



For oil mist lubrication (for high-speed $n.d_S \leq 1\ 500\ 000\ min^{-1}.mm$) the oil with the viscosity ≥ ISO VG 460 in the quantity 0.007 ÷ 0.5cm^3/1 l air is used. This method of lubrication is disadvantageous, the oil supplied into the bearing does not return, it disappears as mist in the environment.

Oils with viscosity up to ISO VG 1500 are used for lubrication of the oil-air system (for high-speed $n.d_S < 1\ 500\ 000\ min^{-1}.mm$). The oil is transported to the lubricated place in macrodrops by air. It escapes into the environment only negligibly and this is the main advantage of this system compared with oil mist lubrication. The ratio oil-air is in the range of 0.004 ÷ 0.5 cm^3/l l air. The necessary quantity of oil for rolling bearing lubrication can be calculated from the following equation:

$$M = (0.7 \div 1.4) \,.\, 10^{-4} \,.\, d \,.\, i \,.\, f_p$$

where: M	-	oil quantity	[$cm^3.min^{-1}$]
d	-	bearing bore diameter	[mm]
i	-	number of rolling element rows	[-]
f_p	-	factor of the operating conditions	[-]
f_p	=	1 for normal operating conditions	[-]

4.3 Lubrication with Solid Lubricants

Solid lubricants are used for the rolling bearing lubrication if grease or liquid lubricants cannot fulfil the exceptional requirements on lubrication in conditions of limiting friction or from the point of view of the resistance against high temperatures, chemical effect, etc. In such arrangements we recommend to consult the kind of lubricant and suitable type of lubrication with the experts of the PSL Technical Consultancy Department (address see page 2).

4.4 Rolling Bearing Inspection in Operation

The bearing service in operation includes regular inspection of the operation, relubrication, or bearing cleaning and relubrication.
In important equipment where high reliability of operation is required, it is suitable to check the bearings during operation by special devices, to record the measurement results and to evaluate the wear, as well as lubrication.

4.5 Storage of Rolling Bearings

PSL bearings are lubricated before packing by a liquid lubricant which need not be removed before mounting. This anticorrosive preservation is effective only if the storaging rooms comply with following conditions:
- relative air humidity must not be greater than 60%,
- the temperature range of the store must be 5 ÷ 25°C
- agressive chemicals e.g. acids, ammonia, chlorinated lime, etc. must not be stored in the same room as the bearings

5. MOUNTING AND DISMOUNTING OF ROLLING BEARINGS

An important condition of reliable rolling bearing operation is the correct selection of the bearings, the arrangement design and their professional mounting. The expected life can be decreased also by incorrect transport handling, unsuitable storaging, pollution or damaging the rolling surfaces, etc.

5.1 Preparation for Mounting or Dismounting of Rolling Bearings

The preparation of mounting or dismounting includes following activities:
- preparation of the mounting workplace (cleaning and preparing all tools necessary for mounting or dismounting so that work can be smooth),
- preparing the working procedure
- assembly parts preparation (cleaning, check of appearance, dimensions, deviations of the form and position of the journal arrangement surfaces before mounting or after dismounting. For bearing washing technical petrol, petroleum, etc. are used. The cleaned components must be protected against corrosion).

5.2 Mounting and Dismounting Methods

Various rolling bearing types and sizes require also different procedures and types of mounting and dismounting. The survey of the used methods and suitable fixtures, or tools can be seen in Table 39.

Table 39

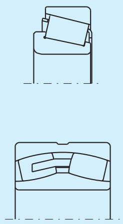
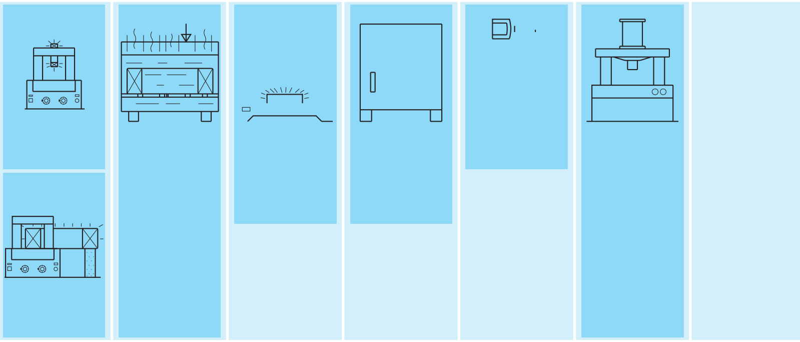
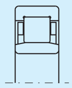
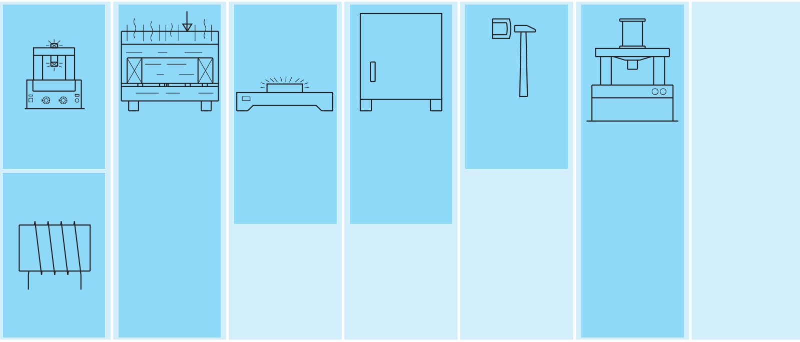
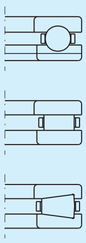
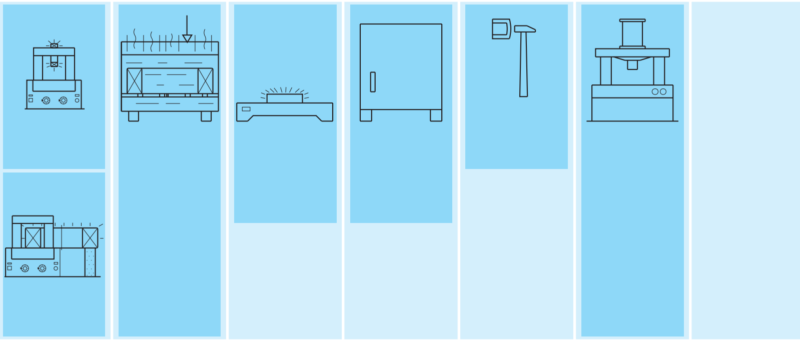
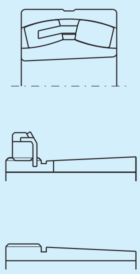
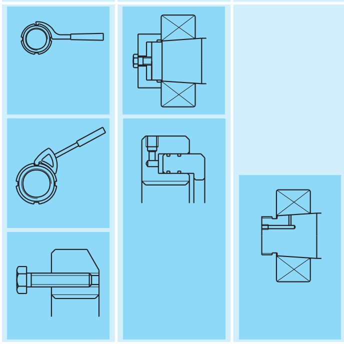
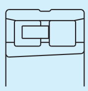
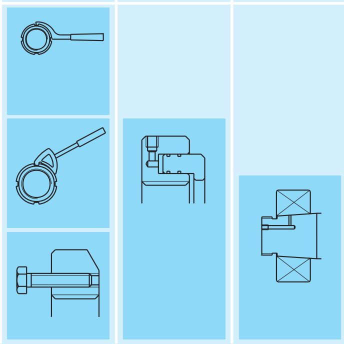
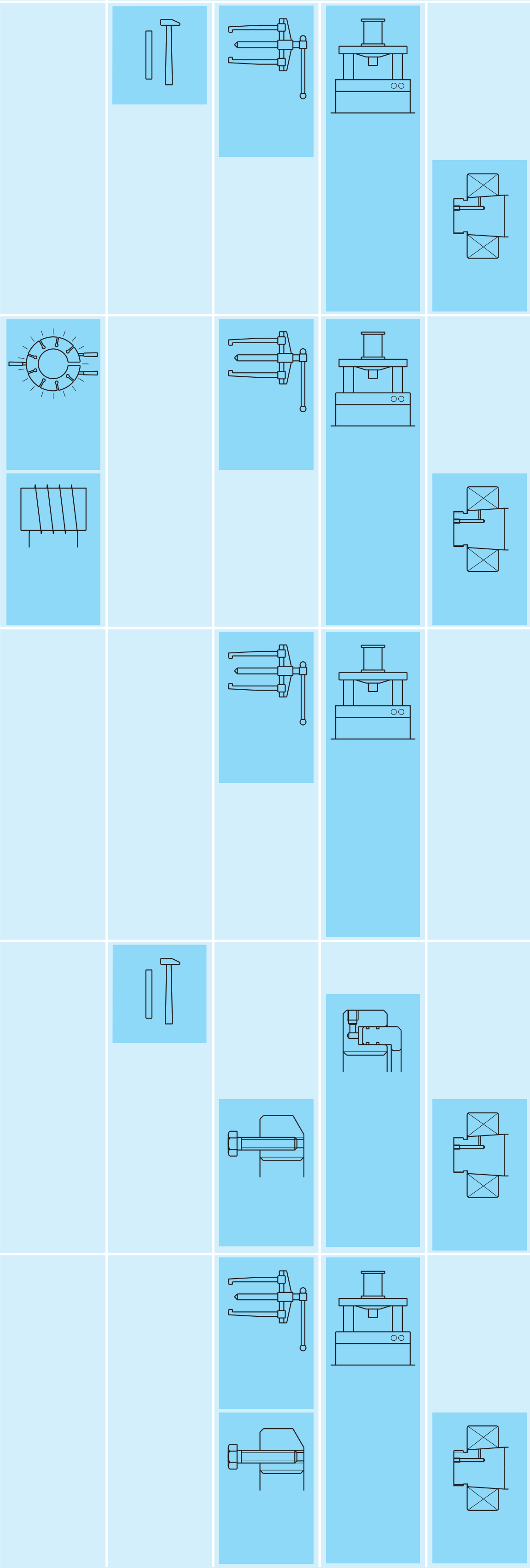
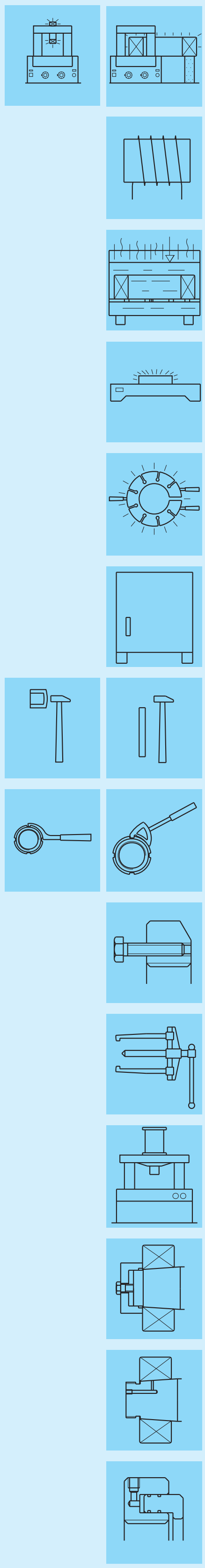
Bearing Type		Bearing Bore	Bearing Size	Thermal Mounting / Cold Mounting
	Tapered Roller Bearings Spherical Roller Bearings	Cylindrical	Small Sized	
			Medium Sized	
			Large Sized	
	Tapered Roller Bearings	Cylindrical	Small Sized	
			Medium Sized	
			Large Sized	
	Thrust Ball Bearings Thrust Cylindrical Roller Bearings Thrust Tapered Roller Bearings	Cylindrical	Small Sized	
			Medium Sized	
			Large Sized	
	Spherical Roller Bearings Adapter Sleeve Wthdrawal Sleeve	Tapered	Small Sized	
			Medium Sized	
			Large Sized	
		Tapered	Small Sized	
			Medium Sized	
			Large Sized	

Table 39 - continued

Thermal Dismounting	Cold Dismounting	Symbol Meaning





Universal Induction Heater

Induction Heating Device

Heating in Oil Bath

Electric Heating Plate

Induction Coil

Hot-Air Heater
Heating Box

Hammer and Mounting Sleeve
Hammer and Mandrel

Nut and Hook Spanner,
Simple or Double
Nut and Hook Spanner
for Clamping by Hammer

Nut and Mounting Screws

Puller

Mechanical or Hydraulic Press

End Packing

Hydraulic Mounting or Dismounting

Mounting or Dismounting
by Hydraulic Nut

5.3 Mounting of Rolling Bearings

5.3.1 Some Principal Recommendations for Rolling Bearing Mounting

- Force necessary for pressing the bearing into the arrangement must not be transmitted through the rolling elements. The assembly jig (a sleeve made of soft steel) should be always placed on the ring which is being pressed or on both rings simultaneously - uniformly on the circumference. The shape and dimensions of the sleeve must not cause the damage of the bearing cage or rings.

- By thermal mounting the temperature required is 80 ± 10°C. The temperature must not exceed 120°C. The sealed bearing filled with grease can be heated to max. 80°C, but not in the oil bath.

- In housings made of aluminium alloys the seat can be easily damaged when pressing the outer ring with firm fitting. In these cases the housing should be heated, or the bearing can be frozen. Large sized bearings are cooled in a mixture of dry ice and alcohol. The bearing temperature must not drop below -50°C, so that the bearing steel should not embrittle.

- Bearings with greater weight can be mounted with a crane and a flexible suspension. The spring inserted between the crane hook and a suitable suspension enables to adjusting the bearing position when pressing it on the journal or into the housing more precisely and easier.

- When mounting by the hydraulic method the seating surfaces must not be damaged (e.g. scratches, places with the contact corrosion, etc.) so that creation of the oil film can be secured and the oil should not escape from contact gap through the damaged places.

- If the bearing is to be arranged on the journal loosely (e.g. in the arrangements of the rolling mill rollers because of often dismounting) the inner rings must not be axially clamped.

5.3.2 Clearance in Arrangement - Selection and Its Adjustment by Mounting

One of the decisive factors influencing the functionality and reliability of the arrangement operation is the internal operating clearance (or preload) in the arrangement. The necessary clearance size, or preload, is selected according to the operating requirements and also according to the bearing types and the arrangement design.
The values of the radial or axial bearing clearances before mounting are shown in the Tables 19 - 23, chapter 2.4.
The operating clearance of the cylindrical roller and spherical roller bearings with cylindrical bore will be changed due to pressing the rings on the journal, or into the housing (smaller clearance) and due to the temperature difference between the inner and outer ring (the clearance is greater or smaller).

In practice the change of the radial clearance is checked by a calculation beforehand and its size is verified when mounting. The size of the radial clearance after mounting the bearing into the arrangement can be calculated according to the following equation:

$$v_{rm} = v_{ro} - k_i \cdot p_i - k_e \cdot p_e$$

where: v_{rm}	-	radial clearance in the bearing after mounting	[mm]
v_{ro}	-	radial clearance in the bearing before mounting	[mm]
p_i	-	size of the inner ring interference on the journal	[mm]
p_e	-	size of the outer ring interference in the housing	[mm]
k_i k_e	-	factors	[-]

$k_i \approx$ 0.8 - for solid journal
$k_i \approx$ 0.6 - for hollow journal
$k_e \approx$ 0.7 - for housing made of steel or cast iron
$k_e \approx$ 0.5 - for housing made of light metals

The change of the radial clearance due to the temperature difference between the inner and outer ring is as follows:

$$\Delta v_r = \alpha_1 \,.\, De_{(i)} \,.\, \Delta T$$

where: α_1 - coefficient of the linear expansion [°C^{-1}]
Δv_r - change of the radial clearance [mm]
$De_{(i)}$ - diameter of the bearing inner (outer) ring raceway [mm]
ΔT - teplotný rozdiel [°C]

The theoretical operating radial clearance then will be as follows:

$$v_{rp} = v_{rm} \pm \Delta v_r$$

v_{rp} - theoretical operating radial clearance
$+ \Delta v_r$ - if the outer ring is warmer, the radial clearance increases
$- \Delta v_r$ - if the inner ring is warmer, the radial clearance decreases

When mounting the tapered roller bearings which are arranged in pairs in "O" or "X" arrangement, the arrangement clearance (or preload) is adjusted on the required value by axial displacement of one ring against the other by tightening of the clamping nut on the shaft, or inserting calibrating washers into the housing.
The relationship between the axial and radial clearance of the bearing pair is as follows:

1. if both bearings have the same contact angle α, then:

$$v_a = \frac{v_r}{tg\alpha}$$

2. if one bearing has the contact angle α_1 and the other α_2, then:

$$v_a = \frac{v_r}{2}\left(\frac{1}{tg\alpha_1} + \frac{1}{tg\alpha_2}\right)$$

where: v_a - axial clearance of a pair of tapered roller bearings [mm]
v_r - radial clearance of a pair of tapered roller bearings [mm]

When selecting a suitable arrangement clearance with a pair of tapered roller bearings, it is necessary to take into account the thermal expansion of the shaft. By the "O" arrangement of the bearings through the operating temperature increase the axial clearance decreases. That is why the clearance adjusted at mounting must be greater by the expected decrease due to the temperature.
Recommended sizes of the axial clearance of a pair of tapered roller bearings after mounting into the "X" arrangement are shown in the Table 40.

Axial Clearance of Tapered Roller Bearings in "X" Arrangement Table 40

Bore Diameter d		Axial Clearance	
over	to	min	max
mm		mm	
–	80	0.05	0.15
80	120	0.08	0.25
120		0.10	0.30

Double row spherical roller bearings with a tapered bore are mounted on the shaft by means of the adapter or withdrawal sleeves, or they are mounted directly on the journal. By pressing the bearing on the taper the decrease of the radial clearance, or the axial displacement of the inner ring on the taper is checked. The measured values characterize the connection stiffness.

Recommended values of the decreased radial clearance, the axial displacement on the taper, as well as the values of the radial clearance after mounting of the spherical roller bearings are shown in theTable 41.

Radial Clearance of Double Row Spherical Roller Bearings after Mounting Table 41

Bore Diameter d		Decrease of Radial Clearance		Axial Displacement[1]				Smallest Permissible Clearance after[2] Mounting of Bearing with Clearance:		
				Taper 1:12		Taper 1:30				
over	to	min	max	min	max	min	max	normal	C3	C4
mm										
65	80	0.040	0.050	0.6	0.75	–	–	0.025	0.040	0.070
80	100	0.045	0.060	0.07	0.09	1.75	2.25	0.035	0.050	0.080
100	120	0.050	0.070	0.75	1.1	1.9	2.75	0.050	0.065	0.100
120	140	0.065	0.090	1.1	1.4	2.75	3.5	0.075	0.080	0.110
140	160	0.075	0.100	1.2	1.6	3.0	4.0	0.055	0.090	0.130
160	180	0.080	0.110	1.3	1.7	3.25	4.25	0.060	0.100	0.150
180	200	0.090	0.130	1.4	2.0	3.5	5.0	0.070	0.100	0.160
200	225	0.100	0.140	1.6	2.2	4.0	5.5	0.080	0.120	0.180
225	250	0.110	0.150	1.7	2.4	4.25	6.0	0.090	0.130	0.200
250	280	0.120	0.170	1.9	2.7	4.75	6.75	0.100	0.140	0.220
280	315	0.130	0.190	2.0	3.0	5.0	7.5	0.110	0.150	0.240
315	355	0.150	0.210	2.4	3.3	6.0	8.25	0.120	0.170	0.260
355	400	0.170	0.230	2.6	3.6	6.5	9.0	0.130	0.190	0.290
400	450	0.200	0.260	3.1	4.0	7.75	10	0.130	0.200	0.310
450	500	0.210	0.280	3.3	4.4	8.25	11	0.160	0.230	0.350
500	560	0.240	0.320	3.7	5.0	9.25	12.5	0.170	0.250	0.360
560	630	0.260	0.350	4.0	5.4	10	13.5	0.200	0.290	0.410
630	710	0.300	0.400	4.6	6.2	11.5	15.5	0.210	0.310	0.450
710	800	0.340	0.450	5.3	7.0	13.3	17.5	0.230	0.350	0.510
800	900	0.370	0.500	5.7	7.8	14.3	19.5	0.270	0.390	0.570
900	1000	0.410	0.550	6.3	8.5	15.8	21	0.300	0.430	0.640
1000	1120	0.450	0.600	6.8	9.0	17	23	0.320	0.480	0.700
1120	1250	0.490	0.650	7.4	9.8	18.5	25	0.340	0.540	0.770
1250	1400	0.550	0.720	8.3	10.8	21	27	0.360	0.590	0.840

[1] Valid only for solid steel shafts.
[2] The clearance after mounting must be checked, if the radial clearance of the bearing is in the lower half of the value range and if in operation significant temperature differences of the inner and outer ring can rise. The clearance after mounting must not be smaller than values shown in the Table.

Axial bearings working under higher rotational speeds must be permanently preloaded so that no sliding of the balls due to the centrifugal forces can rise. The size of the preload, or the minimum axial load can be calculated according to the following equation:

$$F_{amin} = M \left(\frac{n_{max}}{1000}\right)^2$$

where: F_{amin} - minimum axial load [kN]
M - coefficient of the minimum axial load (values-see publication: Rolling Bearings PSL - Production Programme) [-]
n_{max} - maximum rotational speed [min^{-1}]

5.3.3 Special Mounting Procedures

In some cases, which are different from the common practice, e.g. mounting of the four-row tapered roller bearings or double row cylindrical roller bearings - type NN30..K, it is necessary to have at disposal a detailed mounting instruction containing the working procedure, list of necesssary tools, gauges, etc.
In these cases the necessary help can be provided on request by the experts of the PSL Technical Consultancy Department (address - see page 2).

5.4 Dismounting of Rolling Bearings

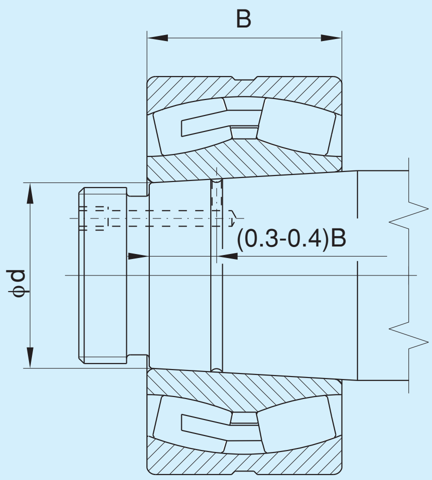
The survey of the used methods and tools necessary for dismounting of the rolling bearings is shown in the Table 39.
The dismounting method should be solved already in the arrangement design, i.e.it should include there the grooves for the puller, or bores for extraction bolts, etc.
If the dismounted bearings and mating parts are to be used again, they must be dismounted in a suitable way, so that they cannot be damaged. The force necessary for dismounting must not be transmitted through the rolling elements, so that the functional bearing surfaces cannot be damaged.
When dismounting bearings with a tapered bore, the axial rebound of the bearing should be limited by a nut, an end plate, or by a stop. The pressed connection is released by an impact and here the danger of injury rises.
In difficult operating conditions the contact corrosion, indentations of the surfaces can arise in some cases. This leads to a more complicated dismounting. In these cases it is suitable to use an oil of higher viscosity with additives for dissolving the rust.
The basic information for the design of grooves for supply the pressurized oil into the surfaces by the hydraulic mounting and dismounting is shown in the Table 42.

Hydraulic Method of Mounting and Dismounting
- Groove Design for Oil Supply

Table 42

LOCATION OF GROOVE FOR OIL SUPPLY

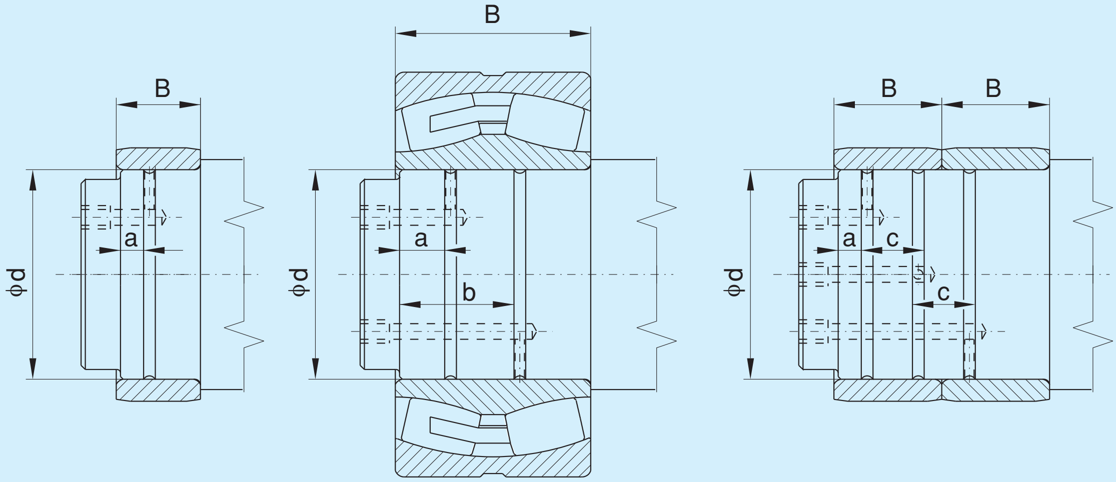
BEARINGS WITH TAPERED BORE



BEARINGS WITH CYLINDRICAL BORE

B ≦ 80

B > 80

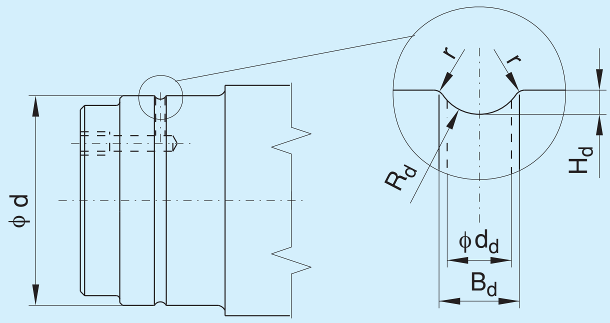


$a = \sqrt{d}$

$b = (0.5\text{-}0.6)B$

$c = (1.5\text{-}2)a$

GROOVE PROFILE



d	–	shaft diameter	[mm]
B_d	=	0.01 d+2	[mm]
H_d	=	$0.3B_d$	[mm]
d_d	=	$0.8\,B_d$	[mm]
R_d	=	$1.1\,B_d$	[mm]
r	=	$(0.10\text{-}0.15)\ B_d$	[mm]

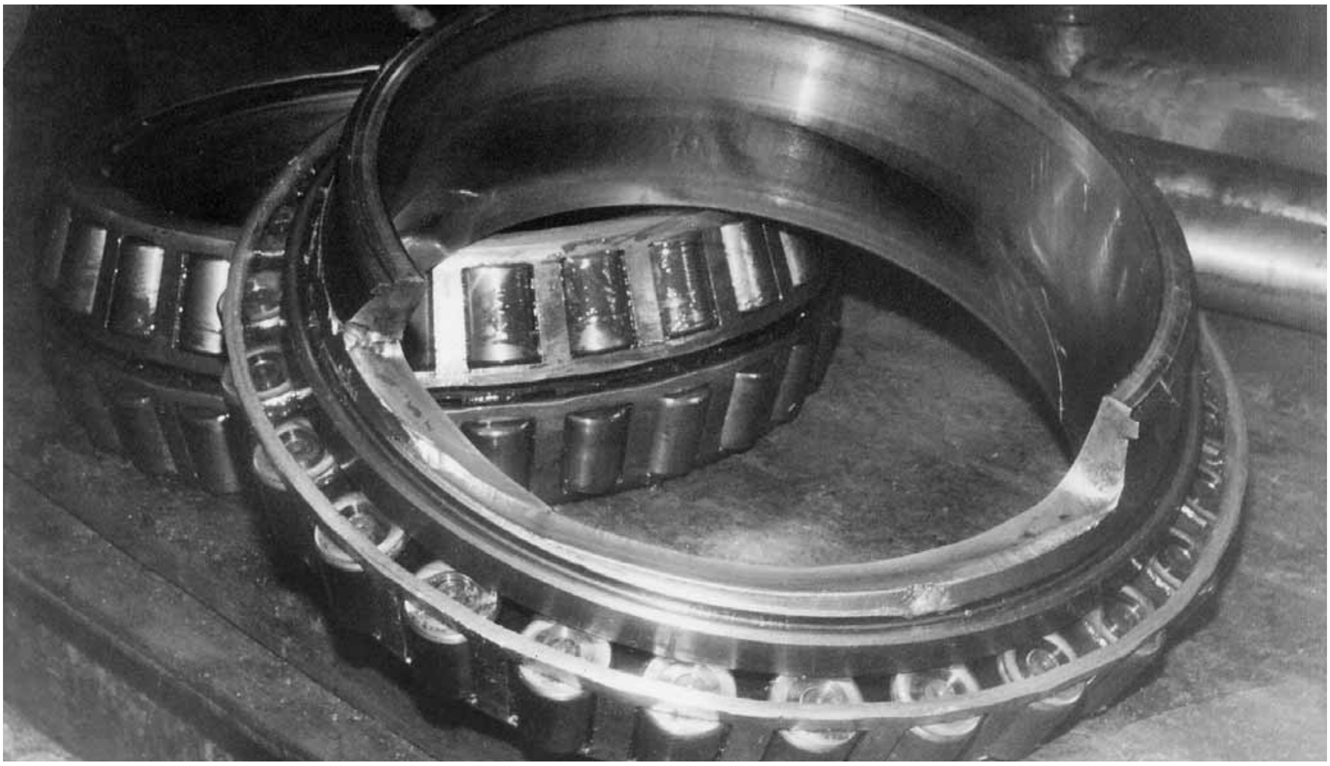
5.5 Typical Causes of Rolling Bearing Damage

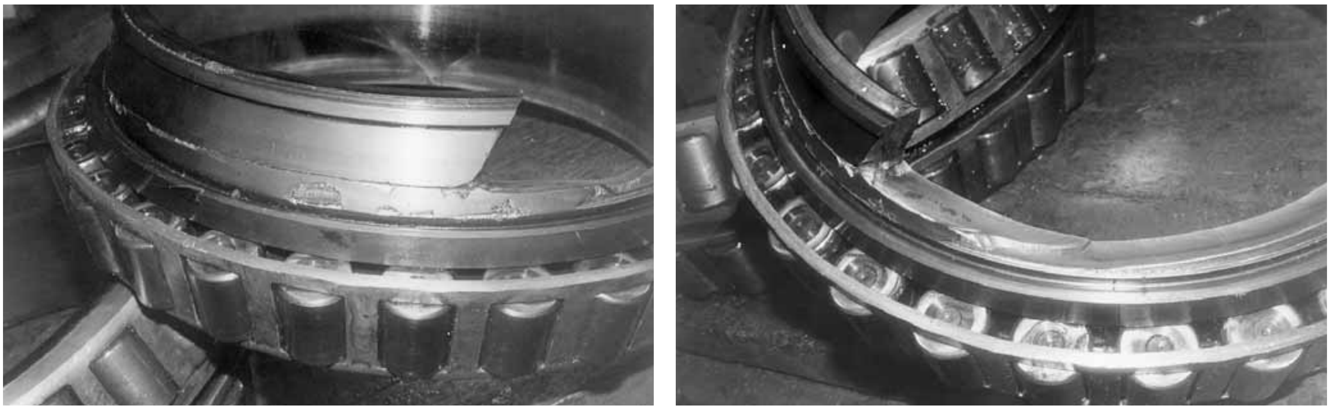
Survey of Some Bearing Failure Types Table 43

CAUSE / DAMAGE	VISIBLE AFTER BEARING DISMOUNTING																				APPEARANCE DURING OPERATION		
	CRACKS AND DEFORMATIONS						FATIGUE		OVERHEATING				CORROS.		OTHER WEAR								
	-cracks and deformations of rings	-cracks and deformations of cage	-chipping, spalling of material	-deformation of raceways	-indentations from hard impurities	-indentations from rolling elements	-functional surface flaking or spalling	-fatigue cracks	-colouring	-annealing, overheating	-crakcks due to overheating	-material transfer and rolldown	-chemical corrosion	-friction corrosion (contact, vibration)	-strong circumference marks due to operation	-abrasive wear of raceways	-grooves, scratches, craters	-slipping, seizing	-signs after vibrations	-irregular or difficult operation	-excessive, irregular noise	-non-typical temperature course
TRANSPORT AND STORAGE																						
-unsuitable storage				■	■								■							■	■	■
-unsuitable handling and transport				■									■	■					■	■	■	■
MOUNTING																						
-insufficient protection - pollution					■							■					■			■	■	
-non-professional mounting without suitable tools	■	■	■	■	■	■														■	■	
-non-professional heating at mounting				■							■									■		
-misaligned fixing		■					■	■								■				■		
-preload in radial or axial direction	■	■	■	■			■	■	■	■	■	■			■					■	■	■
-insufficient fixing	■		■	■		■	■	■						■	■			■		■	■	■
-irregular rigidity of supporting surfaces	■						■	■						■		■				■	■	■
-shape and tolerance errors of the surfaces	■	■	■	■		■	■	■	■	■				■						■	■	■
OPERATING CONDITIONS																						
-increased rotational speed		■							■	■	■					■		■				■
-overloading	■			■			■	■				■				■		■			■	■
-increased number of loading cycles		■					■	■														
-excessive vibrations		■		■		■	■	■									■		■		■	
LUBRICATION AND SEALING																						
-insufficient lubricationn		■					■		■	■	■	■			■	■		■		■	■	■
-excessive lubrication									■	■	■	■										■
-errorous viscosity, insufficient quality							■		■	■	■	■								■	■	■
-lubrication pollution, firm or liquid			■	■		■	■		■				■		■	■	■			■	■	
ENVIRONMENTAL INFLUENCE																						
-strange heat source							■	■	■	■												■
-dust, impurities							■	■								■				■	■	
-passage of electric current																	■			■	■	
-humidity and aggressive media													■							■	■	

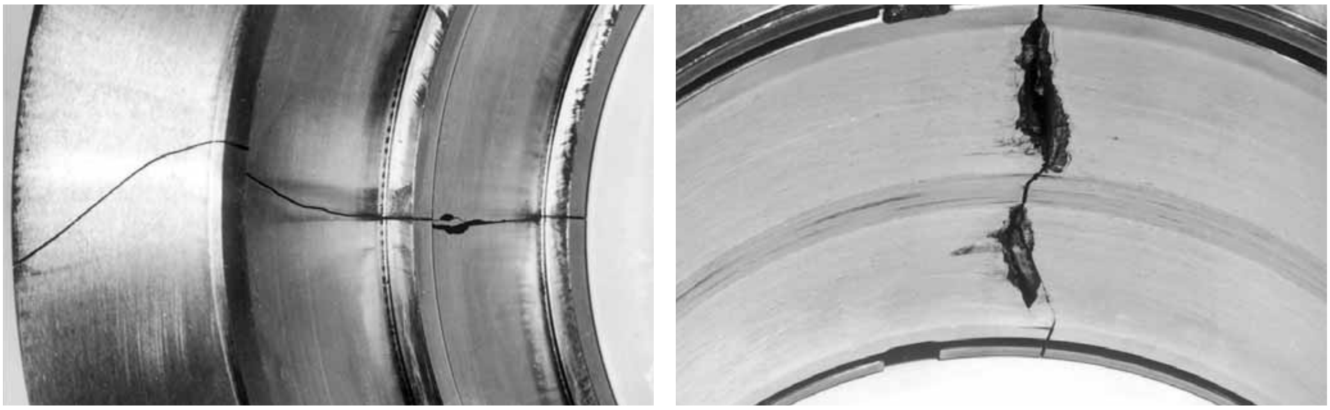
5.5.1 Visual Characteristics of Most Common Damages

Cracks and Deformations

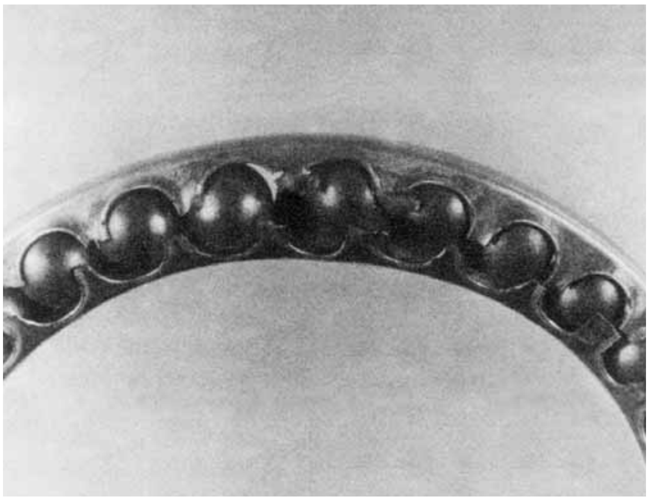




The inner ring of a four-row tapered roller bearing damaged by a circumference crack due to excessive wear of roll journal, i.e. an irregular leaning of the ring on the whole width.



A cross crack in place of an insufficient leaning of the cylindrical roller bearing outer ring.

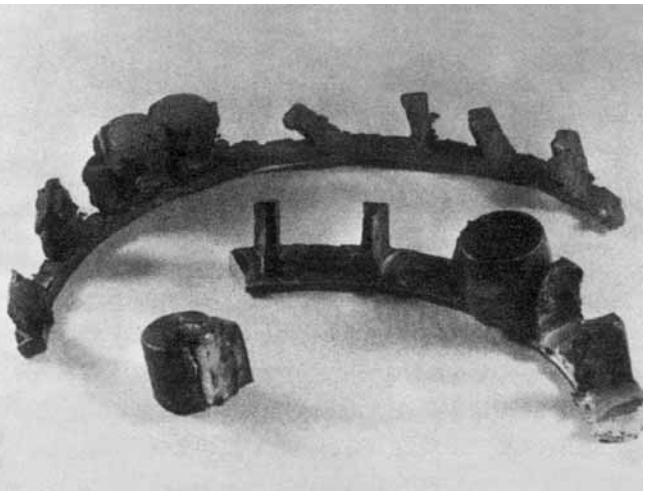


Crack of a cage of an thrust bearing due to insufficient lubrication and excessive wear of the rolling surfaces.

Chipping of a rib of a tapered roller bearing due to non-professional mounting.

Deformation, or re-forming of the raceway and the supporting face of the inner ring of a tapered roller bearing due to the axial overload or insufficient lubrication.

Deformation a spherical roller bearing rolling elements due to the axial overload or insufficient lubrication.

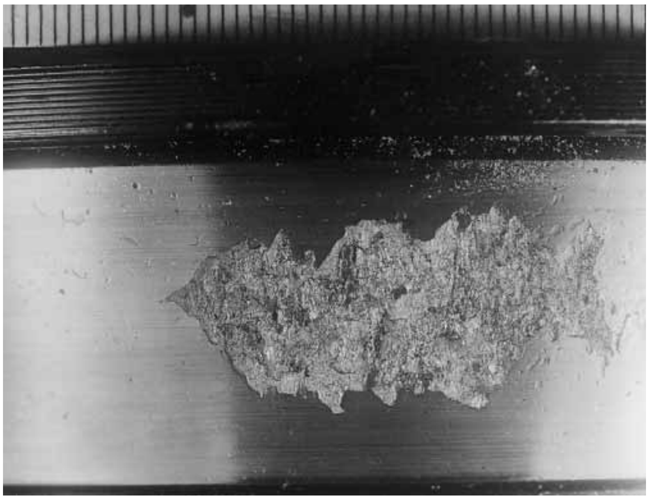


Complete cage destruction of a cylindrical roller bearing due to insufficient lubrication.

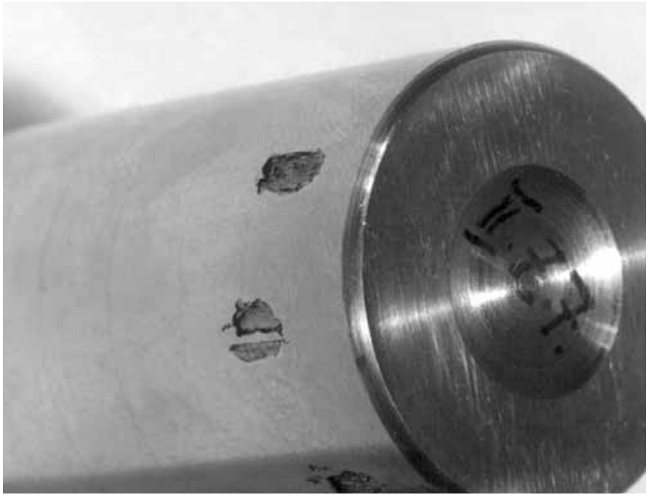
A bearing ring broken into two pieces due to a large axial overload.

Material Fatigue


Fatigue of tapered roller bearing raceways - pitting. The pitting location is proof of a large edge load possibly due to misalignment of the surfaces.

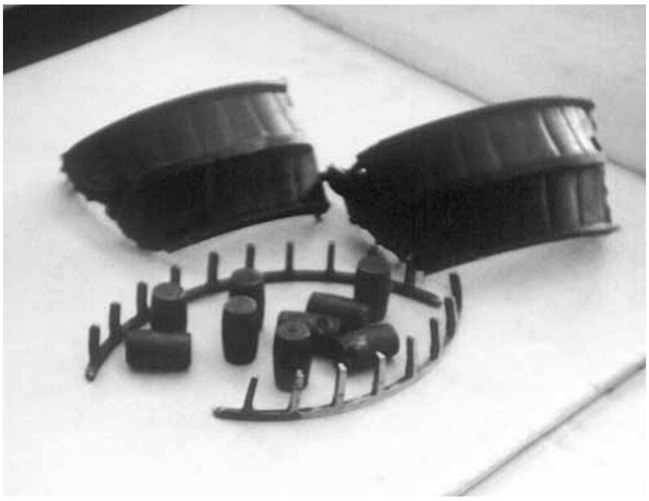
Fatigue of tapered roller bearing raceways - the initial and developed stage of pitting.




Rolling element surface fatigue.

Overheating

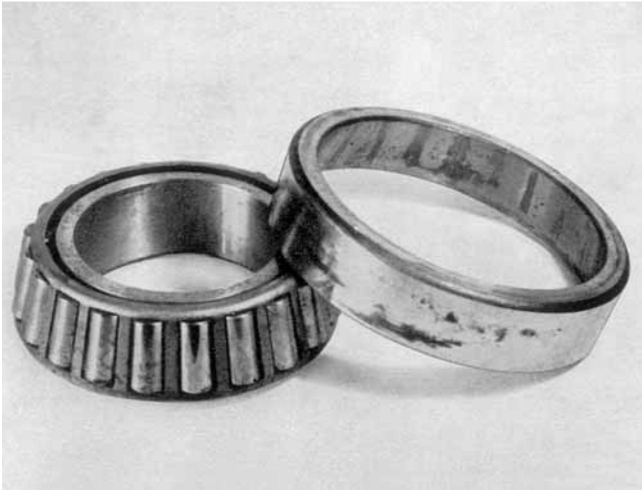
A detail picture of a cylindrical roller raceway fatigue - surface spalling due to an unsuitable lubricant.



Overheating and following blockage of a spherical roller bearing due to insufficient radial operating clearance and absence of lubrication.

Corrosion

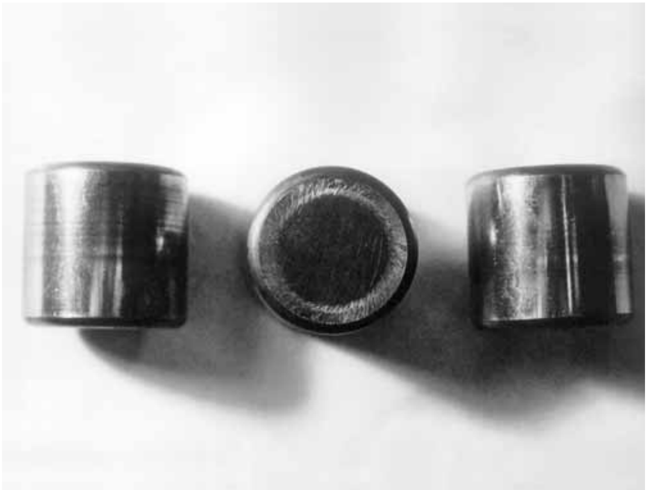
Chemical corrosion of an outer bearing ring raceway.



Contact corrosion of both the raceways and surfaces of a tapered roller bearing.

Other Wear

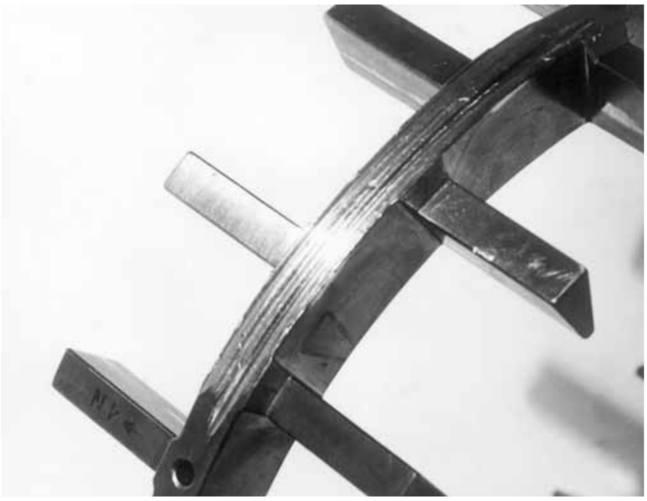
Seizing of cylindrical rollers on the rolling surface.



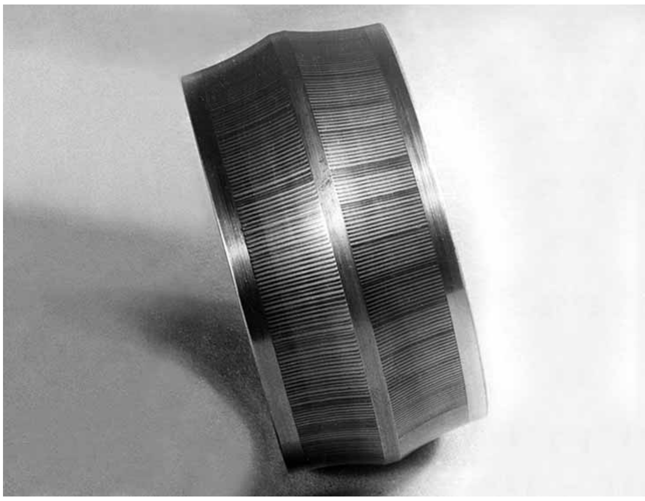
Cylindrical roller seizing on the faces.

Other Wear

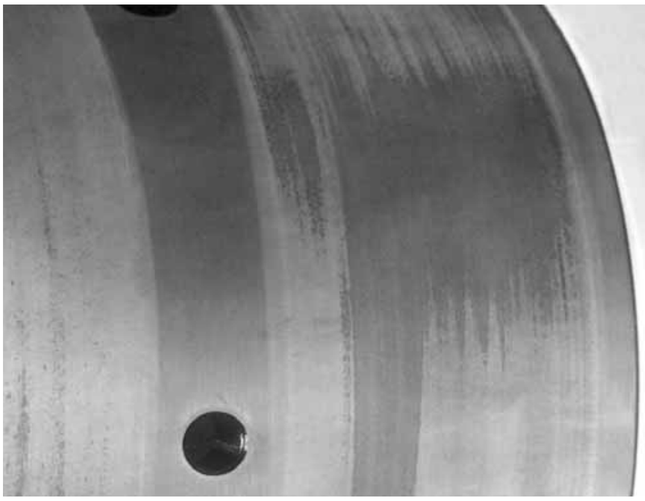
Seizing of guiding surface of a loose rib.



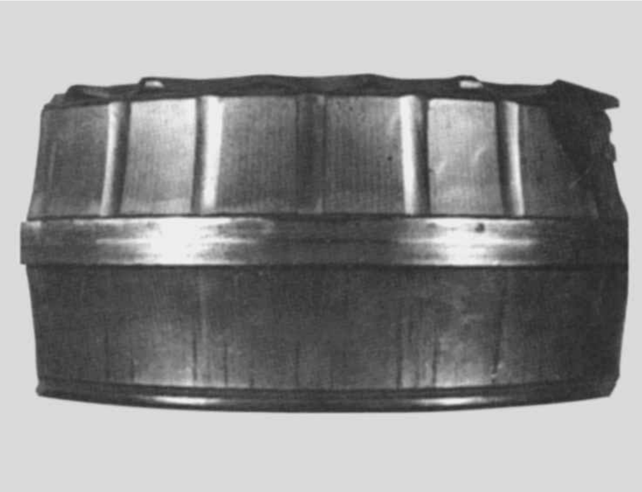
Seizing of a cage guiding surface.



Passage of electric current.



Abrasion - circumference signs on the raceway.

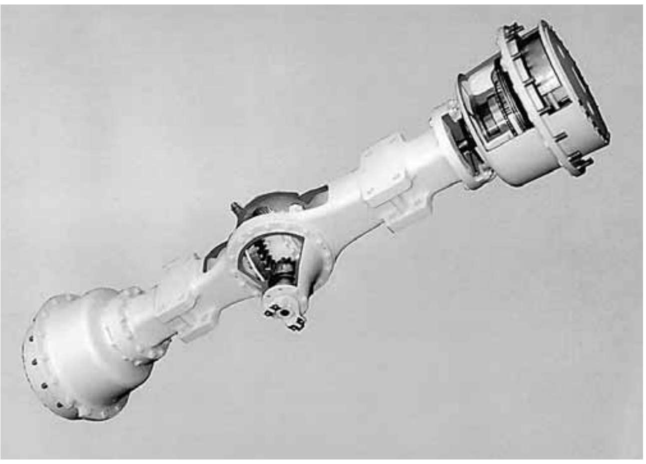


Damage of raceways of a spherical roller bearing due to vibrations, so called brinelling (the bearing does not rotate).

Craters and slipping marks on the raceway.

Conversion Equivalents for U.S. and Metric Measurements

Measurement	When you Know	Multiply by	To get an equivalent in
Lenght	[inch]	25,4	[mm]
	[mm]	0,03937	[inch]
	[ft]	0,3048	[m]
	[m]	3,2808399	[ft]
	[mile]	1,609	[km]
	[km]	0,6214	[mile]
Area	[sq. inch]	645,16	[mm^2]
	[mm^2]	0,001550003	[sq. inch]
	[sq. ft]	92903,04	[mm^2]
	[mm^2]	0,00001076391	[sq. ft]
Volume	[c. inch]	16387,064	[mm^3]
	[mm^3]	0,000061023744	[c. inch]
Weight	[lb]	0,4536	[kg]
	[kg]	2,2046	[lb]
	[lb]	0,0004536	[t]
	[t]	2204,6	[lb]
Force	[lbf]	4,448222	[N]
	[N]	0,22480892	[lbf]
	[lbf]	0,004448222	[kN]
	[kN]	224,80892	[lbf]
Torque	[lbf.inch]	0,1129848	[Nm]
	[Nm]	8,850748	[lbf.inch]
	[lbf.ft]	1,3558182	[Nm]
	[Nm]	0,73756207	[lbf.ft]
	[lbf.ft]	0,0013558182	[kNm]
	[kNm]	737,56207	[lbf.ft]
Temperature	[°F]	(°F-32)/1,8	[°C]
	[°C]	1,8.°C+32	[°F]
Presure, Stress	[psi]	0,006894757	[MPa]
	[MPa]	145,03774	[psi]
Power	[hp]	0,7457	[kW]
	[kW]	1,341	[hp]
Velocity	[$ft.s^{-1}$]	0,3048	[$m.s^{-1}$]
	[$m.s^{-1}$]	3,2808399	[$ft.s^{-1}$]
	[mph]	1,609	[$km.h^{-1}$]
	[$km.h^{-1}$]	0,621	[mph]
Acceleraction	[$ft.s^{-2}$]	0,3048	[$m.s^{-2}$]
	[$m.s^{-2}$]	3,2808399	[$ft.s^{-2}$]

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