S.C. "Rulmenti" S.A. Barlad – Romania
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SELECTION OF BEARING TYPE

Selection of bearing type considering the magnitude and direction of load

The magnitude and direction of loads greatly influence the selection of bearings.

Generally, for the same dimensions, the cylindrical roller bearing stands heavy loads than the deep groove balls bearing. The bearings with more rows of rolling elements, especially rollers have heavy load carring capacity.

According to the load acting direction, the following situation are distinguished:

a) Radial load

Cylindrical roller bearings without ribs at one of the rings, with one row of rollers (type N or NU) or with two rows of rollers (type NN or NNU) and needle roller bearings are to be used.

b) Axial load

Thrust balls or roller bearings according to the load magnitude, are to be used. The simple effect thrust roller bearings can be loaded only in a single direction and the double effect thrust roller bearings can be loaded in both directions.

c) Combined load

The simultaneous action of radial and axial load means that on the roller bearing acts a combined load.

For light axial loads together with radial loads are used:

- deep groove ball bearings, single row. (Combined load supported rises if the radial clearance is greater than normal);
- Cylindrical roller bearings of the NUP and NJ+HJ types and spherical roller bearings.
- NJ type cylindrical roller bearings can only accommodate axial loads acting in a single direction and for axial displacement of the shaft in both directions it is recommended to mount roller bearings of the same type.

If the axial load is heavy, a thrust bearing must be mounted together with a radial roller bearing. The angular contact balls bearing or four-point contact bearings (Q or QJ type) used when axial load predominates are mounted with clearance fit for housing.

In case of combined loads in which heavy axial load predominates, angular contact ball bearings single or double rows taper roller bearings or spherical roller thrust bearings. The above are presented in figure 1. in which the black triangles indicate the loads direction for which the respective bearing was designed and the white triangles are indicating the possible loads.

The size of the bearings is selected considering the condition of life requirements ensuring for imposed conditions of load, rating life and reliability of operation. Selection is done on the basis of a characteristic variable: ***basic load ratings***.

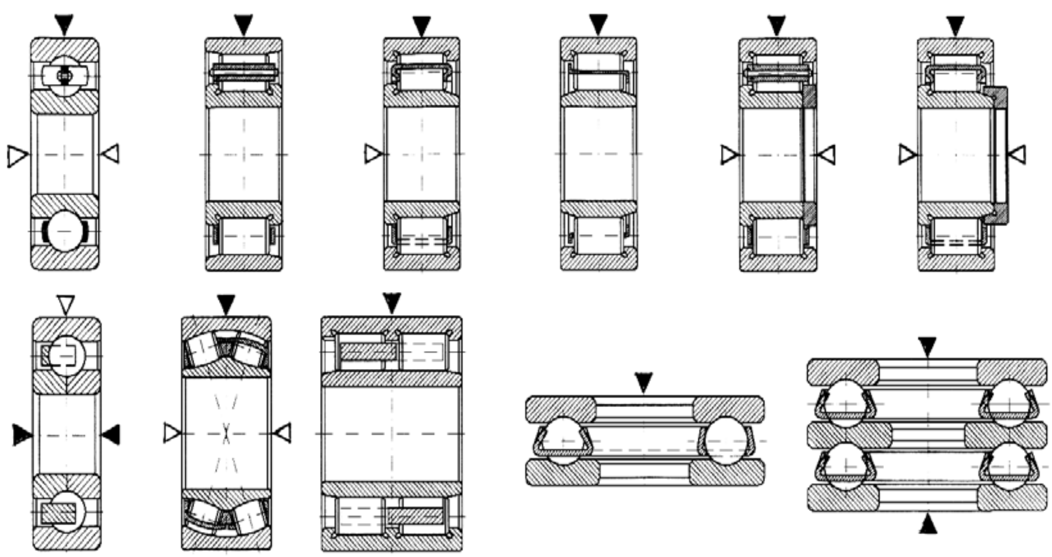

Figure 1

Selection of bearing type considering the alignment between shaft and housing

Angular misalignments occur generally when the shaft bends under the operating load or when bearings adjoint parts have form or position deviations.

In such cases, self-aligning ball bearings, spherical roller bearings or spherical roller thrust bearings should be used.

A certain bearing bent angle can compensate for errors of alignment and maximum angle values are shown for each type in the introductory texts of the table sections.

When misalignments should be compensated, radial and axial clearance are important. The larger the clearance, the greater the possibility of self-aligning.

If the misalignment exceeds the permissible values shown in the introductory texts of the bearing tables, the bearing rating life decreases. The greater the ratio F_r/C_{0r}, the shorter the rating life. If 0.1 <F_{0r}/C_{0r}< 3, the rating life decreases with about 25%.

Selection of bearing type considering the operating temperature

Maximum operating temperature to which the bearings designed for normal applications can be used is of 120°C. Over this temperature in the material of the contact elements (rings and rolling elements) there are produced structural transformations with *negative implications* over the dimensional stability and physical and mechanical characteristics which determine the resistance to contact fatigue and, through implication, to the life of the bearing.

Thus, at higher temperatures it is recommended to use special bearings having the component parts made of special steel brands or stabilized through thermal treatments. These bearings have special symbols.

Remark: When the working conditions of bearing allow great temperature differences in operation for the two rings (interior and exterior) we recommend the use of bearings with radial clearance greater than normal (groups C3, C4, C5)

Selection of bearing internal clearance

In most cases, while operating, bearings should have a small radial clearance that can be defined as "the possible value of displacement in radial direction of one bearing ring in relation to the other without parts deformations".

While operating, bearing internal clearance is diferent from the one at delivery, since the latter is reduced when mounting bearings with a certain tight fit.

Under operating conditions, internal clearance change is also caused by different temperatures between the outer and inner ring. Bearings are generally delivered with a normal radial or axial clearance according to the values shown for each rolling bearing group.

The decrease in radial clearance due to the tight fit and operating temperature is considered to be between 60-80% of the tightening value, depending on bearing series and size.

After the clearance in bearings has been decreased, a large enough operational clearance should remain, so that the lubricant film shouldn't be destroyed.

Deep groove ball bearings should have an operational clearance close to zero. There may be often a light-preload, due to the point-contact between the rolling elements and raceways.

Small-sized cylindrical roller and needle roller bearings should have an operational clearance of 5-10 μm and larger-sized bearings a clearance of 10-30 μm.

Bearing producers can also manufacture - at request-bearings with radial and axial clearance smaller (C1 and C2) or larger (C3, C4 and C5) than normal, so that the most favorable operating conditions for bearings should be asured.

Cylindrical and needle roller bearings can be manufactured with interchangeable rings (no special designation) and with non interchangeable rings (suffix NA).

Bearings with non interchangeable parts have a smaller radial clearance than bearings with interchangeable parts.
Changing rings from one bearing to another is not allowed.

In case of bearings with interchangeable parts, the rings may be changed and the values of radial clearance will be not altered.

Bearing types and technical characteristics

Table 1 shows qualitative results of each group of bearings, considering the main technical characteristics.

Bearing type is selected depending on the technical characteristics required by a certain application.

A suggestive graphic symbol has been determined for each main technical characteristics. Thus, a proper bearing for each purpose can be easily chosen. According to the specifications in this catalogue, the proper type and size of bearing can be selected, together with all manufacturing and operating technical conditions.

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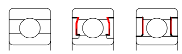
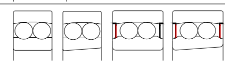
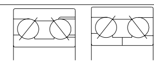
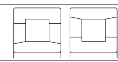
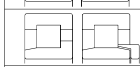
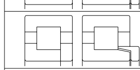
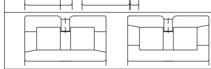
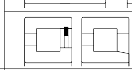
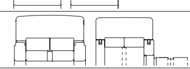
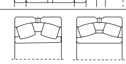
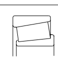
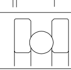
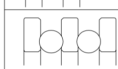
Bearing types and their characteristics

● – excellent
◕ – good
◑ – fair
◔ – poor
○ – unsuitable
→ – single direction
↔ – double direction

Bearing type		Purely radial load	Purely axial load	Combined load	Moment load	Tolerance class	Quiet running	High speed	High stiffness	Compensation of misalignement	Low friction	Shock resistence	Located bearing	Non located bearing	Axial displacement possible in bearing
Deep groove ball bearings: –single row		◕	◑ ↔	◑	◔	●	●	●	◑	◔	●	◑	◕	◑	○
–double row		◕	◑ ↔	◑	◑	◑	◑	◑	◑	○	◕	◑	◑	◑	○
Self–aligning ball bearings		◑	◔	◔	○	◑	◕	◕	◑	●	◑	○	◑	◑	○
Double row angular contact ball bearings		◕	◕ ↔	◕	◑	◑	◑	◑	◕	○	◑	◑	◕	◕	○
Cylindrical roller bearings: NU, N		●	○	○	○	◕	◕	●	◕	◔	◕	●	○	○	●
NJ, NU+HJ		●	◑ →	◑	○	◕	◑	●	◕	◔	◕	●	◑	◑	◑
NUP, NJ+HJ		●	◑ ↔	◑	○	◕	◑	●	◕	◔	◕	●	◑	◑	◑
NNU, NN		●	○	○	○	●	◕	●	●	○	◕	●	○	●	●
NCFV, NJ23VH		●	◑ →	◑	○	◑	◔	◔	●	◔	◔	●	◑	◑	◑
Support rollers		●	◔	◔	○	◑	◑	◔	◑	◑	◔	◑	○	◑	◑
Spherical roller bearings		●	◕ ↔	◕	○	◑	◑	◑	◕	●	◑	●	◕	◑	○
Taper roller bearings –single row		●	◑ →	●	◔	◕	◑	◕	◕	◔	◑	●	◕	○	○
Thrust ball bearings: –single direction		○	◕ ↔	○	○	◕	◔	◑	◑	○	◑	○	○	○	○
–double direction		○	◕ ↔	○	○	◕	◔	◑	◑	○	◑	○	○	○	○

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	Purely radial load	Purely axial load	Combined load	Moment load	Tolerance class	Quiet running	High speed	High stiffness	Compensation of misalignment	Low friction	Shock resistence	Located bearing	Non located bearing	Axial displacement possible in bearing
Angular contact thrust ball bearings: -single direction, RY	◔	◕ —→—	○	◔	◐	◔	◐	◐	○	◐	○	○	○	○
-double direction, 2344	○	◕	○	○	●	◕	◕	◐	○	◐	○	○	○	○
Cylindrical roller thrust bearings 811, 893	○	● —→—	○	○	◐	◔	◔	◕	○	◔	●	○	○	○
Rulmenti axiali cu ace	○	● —→—	○	○	◐	◔	◔	◕	○	◔	●	○	○	○

SELECTION OF BEARING SIZE

Basic load ratings

The size of a bearing is selected considering the load in the used rolling bearing and also depends on the operational rating life and prescribed operating safety.

The basic dynamic load rating C_r is used to calculate bearing dimensions while rotating under load. It expresses the bearing admissible load which will give a basic rating life up to 1000 000 revolutions.

The basic dynamic load ratings of the bearings have been determined in accordance with ISO 281. The values are given in bearing tables.

Considering the basic dynamic load rating, is calculated the service time until the fatigue of the material appears, determining this way the calculated rating life.

Basic static load rating C_{0r} is considered in case of low speeds, low oscillating movements or in the stationary case.

The basic static load rating is defined in accordance with ISO 76, as the load acting upon the stationary bearing. It corresponds to a calculated contact stress in the center of the contact area between the most heavily loaded rolling element and the raceway, of:

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

This stress produces a permanent deformation of the rolling element and raceway which is about 0.0001 of the rolling element diameter. The loads are pure radial for radial bearings and pure axial for thrust bearings.

Bearing live

The life of a rolling bearing is defined as the number of revolutions or the number of operating hours, which the bearing is capable to endure, before the first sign of fatigue occurs on one of its rings, on the raceway or the rolling elements.

If we want to consider only the fatigue on the bearing operating surfaces, the following conditions have to be observed:

1. The forces and speeds considered when calculating the bearing should correspond to the real operating conditions.
2. Proper lubrication should be assured during the entire operating period.
3. If the bearing carries a light load, its failure is generated by the wear.
4. Experience showed that the failure of many bearings was caused by other reasons than fatigue, such as: selection of an inadequate bearing type in a bearing joint, improper operation or lubrication, outer particles in bearing etc.

Basic rating life

The basic rating life of a single bearing or of a group of apparently identical bearings operating under identical conditions, is the life corresponding to a reliability of 90%.

The average life of a group of bearings is approximately five times longer than the basic rating life.

Basic rating life is marked with L_{10} (milions of revolutions) or L_{10h} (operating hours).

L_{10} can be calculated using the equation:

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

L_{10}	- basic rating life, milions of revolutions,
C	- basic bearing load, kN,
P	- equivalent dynamic bearing load, kN,
p	- exponent of the life equation with the following values:
$p = 3$	- for ball bearings
$p = 10/3$	- for roller bearings

The equivalent dynamic bearing load, respectively the radial and axial load, acting simultaneously can be calculated using the following equations (applicable to ball and roller radial bearings):

$P = F_r$, kN - for pure radial load,
$P = XF_r + YF_a$, kN - for combined load,

For thrust ball bearings, the following equations can be used:

$P = F_a$, kN - for pure radial load,
$P = XF_r + YF_a$, kN - for combined load,

where:
F_r - the radial component of the load, kN,
F_a - the axial component of the load, kN,
Values of the coefficients X and Y can be found in tables.

For bearings operating at constant speed, the basic rating life expressed in operating hours can be calculated using the equation:

$L_{10h} = 1\ 000\ 000/60\ n\ (C/P)^p$ sau $L_{10h} = 16\ 666/n\ (C/P)^p$,

where:
n =rotational speed, r/min.

When determining the bearing size it is necessary to base the calculations on the rating life corresponding to the purpose of operation.

It usually depends on the machine type, service life and the requirements regarding operational safety.

Approximate values of the service life for various classes of machines and equipments for general purposes are given in table 1.

Table 1

Application	**Recommended basic rating life L_{10h} (operating hours)**
Household machines, technical apparata for medical use, instruments,agricultural machines:	300...3 000
Machines used for short periods or intermittently: electric hand tools, cranes, lifting tackles in workshops, building machines:	3 000...8 000
Machines used intermittently or for short periods with high operational reliability lifts, small cranes:	8 000...12 000
Machines for use 8 hours/day but not always at full capacity: machines for general purposes, electric motors for industrial use, rotary crushes, gear drives for general purposes:	10 000...25 000
Machines operating 8 hours/day at full capacity: machine tools, woodworking machines, large cranes, printing equipment ventilators, separators, centrifuges:	20 000...30 000
Machines for continuous use 24 hours/day: Rolling mill gear units, medium sized electrical machinery, compressors, pumps, textile machines, mine hoists:	40 000...50 000
Hydraulic machines, rotary fumaces, capstans, propulsion machinery for sea vessels (propellers for sea vessels):	50 000...100 000
Machines for continuous use 24 hours/day with high reliability: large electric machinery, mine pumps and mine ventilators, power station plants, machines for cellulose industry, pumping units:	100 000...

The basic rating life of road and rail vehicle bearings, for wheel-axle bearings, is expressed as a function of the wheel diameter and covered distance (km), using the equation:

$L_{10} = (1000/pD)L_{10s}$, respectively: $L_{10s} = (pD/1000)L_{10}$.

where:
L_{10} - basic rating life, millions of revolutions
L_{10s} - service live distance, millions of kilometers
D - wheel diameter, m.

Approximate values for the service life distance (kilometers covered), in case of light loaded cars and rail vehicles are given in table 2

Table 2

Type of vehicle	$L_{10s}/10^6$, km
Wheel hub bearings for road vehicles - lightly loaded cars - trucks, buses	 0.3 0.6
Axlebox bearings for rail vehicles: - goods wagons (according to UIC) - suburban vehicles, trams - long distance passenger carriages - motorailers - Diesel and electric locs	 0.8 1.5 3 3-4 3-5

In case of bearings which do not rotate but oscillate from a central position through an angle, basic rating life can be determined as follows:

$L_{10OSC} = (180/2g)L_{10}$

where:

L_{10osc} - basic rating life, millions of cycles
g - oscillation amplitude (angle of maximum deviation from center position), degrees.

If the amplitude of oscillation is very small, it can be ignored for basic rating life determination.

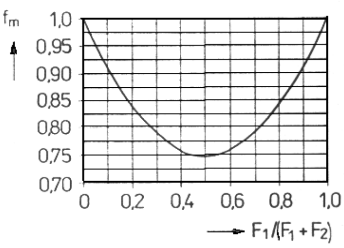


Figure 1

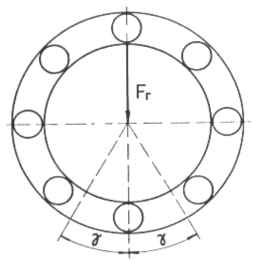


Figure2

Table. 1

γ^0	f_o	
	p=3	p=10/3
10	0,47	0,53
20	0,61	0,65
30	0,69	0,72
45	0,79	0,81
60	0,87	0,89
75	0,94	0,95
90	1,0	1,0

Dynamic load and variable speed

In many operation cases, the size of the load and rotational speed are variable, when an average constant radial F_{mr} or axial F_{ma} load must be calculated for the calculus of the dynamic equivalent load.

1) If at constant rotational speed the force acting on the rolling bearing linearly varies between a minimum value F_{mr}, a_{min} and a maximum value F_{mr}, a_{max}, keeping its direction within a certain interval of time, the medium load results from the following relation:

$$F_{mr,n} = (F_{r,amin} + 2 \cdot F_{r,amax})/3, \quad [KN]$$

2) If the radial load that acts on a rolling bearing is made of a force F_{r1} which is constant in size and direction (for example the weight of a rotor) and a constant rotation force F_{r2} (for example the unbalancing phenomena), the average load results from

$$F_{rm} = f_m(F_{r1} + F_{r2}), \qquad [KN]$$

The values for fm coefficient are obtained from the figure 1:

3) For a radial load, F_r applied on bearing which oscillates from a central position through an angle 2g(see figure 2), the average radial load is done by the following relation:

$$F_{mr} = f_0F_r, \ [KN]$$

with values for f_0 coefficient given in table 1 according to the oscillation angle g and the exponent of the life formula, p.

For variable loads in size, time and direction and for different rotational speeds, the average dynamic loads given by the following formula:

$$F_{mr,a} = [\,\Sigma(F_{ir,a}{}^{p} \cdot n_i)/n]^{1/p},\ [KN]$$

where: $F_{mr,a}$ - average constant load, radial or axial, [KN]

$F_{ir,a}$ - constant loads applied on the duration of effecting of rotations n_i, [KN]

n_i - the no. of rotations corresponding to $F_{ir,a}$ loads

$n = \Sigma n_i$, rot/min

p = 3 for ball bearings and
p = 10/3 for roller bearings

Basic dynamic load of a bearing group

Assembling two or even more identical bearings, the base dynamic load of the *i* bearings assembly being calculated with the following relation:

$C_r = i^{0.7} C_{ri}$ [KN] for punctual contact bearings;

$C_r = i^{7/9} C_{ri}$ [KN] for linear contact bearings.

To take evenly the loads, these bearings must be paired so that the diameter and radial clearance deviations are max. 1/2 of the allowed tolerance field. Relation which refers to the basic dynamic load of the roller bearing indicated in roller bearing tables depends on "basic rating life" (L_{10}) which, according to ISO 281 means the life attained or exceeded by 90% of the bearing group of the same type dimension operating in the same conventional conditions (good mounting, protection against foreign particles penetration, correct lubrication, correct loading, non exposure to extreme speeds and temperatures).

Adjusted rating life

After bearing selection (according to the basic dynamic load), it is recommended to determine its effective life (adjusted rating life for conditions different from those mentioned in ISO 281) with the following relation:

$$L_n = a_1 a_2 a_3 f_t (C_r/P_r)^p$$

where: L_n – adjusted rating life (mil. of rotations),

a_1 - correction factor that takes into account the reliability (table 1);

a_2 - correction factor that takes into account the quality of the material and manufacturing technology (for materials and technologies used for manufacturing URB bearings $a_2 = 1$)

a_3 - correction factor that takes into account the operating conditions and lubrication quality.

f_t – correction factor according to the operation temperature (table 3)

The connection between these two last connection factors leads to their fusion into a single factor, a_{23}, whose value is given in table. 2 and depends on the ratio between the cinematic viscosity of the oil at 40°C, ν initial in cSt or mm²/s (see figure1) and the viscosity required for a correct lubrication at the operation temperature, ν_1 see figure 2.

Table. 1

Reliability, %	L_{na}	a_1
90	L_{10a}	1
95	L_{5a}	0,62
96	L_{4a}	0,53
97	L_{3a}	0,44
98	L_{2a}	0,33
99	L_{1a}	0,21

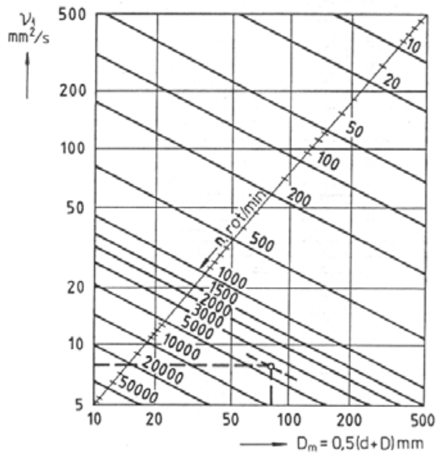


Figure 1

Example of calculation of the kinematic viscosity of the oil:

For a bearing with D_m = 85 mm which operates at a rotational speed of 3500 rot/min, it results from figure 2, ν = 8 mm²/s. From figure 2 results that for the operating temperature of 70°C to obtain ν_1 viscosity it is required an initial viscosity of ν= 20 mm²/s.

Table 2

ν/ν_1	0,1	0,2	0,5	1	1,5	2	3	4	5
a_{23}	0,45	0,55	0,75	1	1,3	1,6	2	2,5	2,5

Table 3

Running temperature [°C]	150	200	250	300
f_t	1	0,73	0,42	0,22

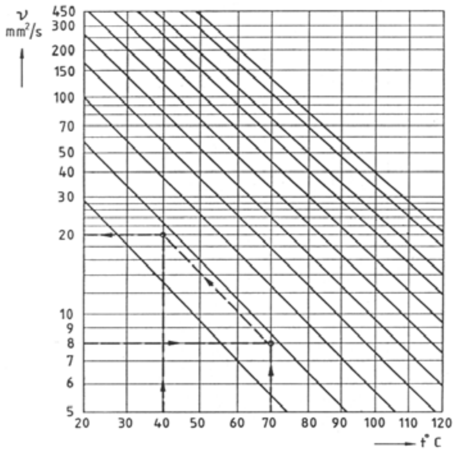


Figure 2

Static load

a) Basic static load

Basic static load, C_{or}, is given in the bearings catalogues for every size and it is taken in consideration when the bearing is stationary, has slow oscillations, low speed (n<10 rot/min) or when during the rotating these must support heavy shock loads. In this case, the operation safety is determined by the size of the deformations of the raceway of the bearing.

Basic static load is determined according to ISO 76 and represents the load that produces a permanent deformation of 0.0001 from the diameter of the rolling element, the load being purely radial for radial bearings and purely axial for thrust bearings.

The combined static load (radial and axial loads that are acting simultaneous on a bearing) must be converted into an equivalent load which results from the general formula:

$$P_0 = X_0F_r + Y_0F_a \text{ , KN}$$

where:
- P_o equivalent static load of the roller bearing, KN
- F_r the radial component of the heavy static load in KN
- F_a the axial component of the heavy static load in KN
- X_o the radial load factor of the bearing
- Y_o the axial load factor of the bearing

X_o and Y_o are given in the bearings tables and catalogues according to the type of the bearing and ratio F_a/F_r

Knowing the shaft diameter “d”, the size of the bearing is determined from the condition of inequality:

$$C_or \geq s_of_{ot}P_o \text{ , KN}$$

where: s_o is a static safety factor coefficient chosen according to table. 1 (for the non-rotating bearings or for the bearings with oscillating movements) and table 2 (for rotating-bearings subject to oscillating loads or heavy shock loads with short duration).

The heavy shock loads that are higher than the basic static load of the bearing are producing remanent deformations not uniformly distributed on raceway which negatively influence the good operation of the roller bearing.

At high operating temperatures C_{or} is corrected with the following factor:

f_{ot} = - 1 for a temperature of 150 ℃
-0,95 for a temperature of 200 ℃
-0,85 for a temperature of 250 ℃
-0,75 for a temperature of 300 ℃

Table 1

Application	s_0
Adjustable pitch airscrew for planes	0,5
Gates of barrages, dams, flood gates	1
Mobile bridges	1,5
Crane hooks for:	
- small cranes, with additional dynamic loads	1,6
- large cranes, without additional loads	1,5

Table 2

Load type	Silent running requirements (without noise)					
	Low		Normal		High	
	Bearings balls	Bearings roller	Bearings balls	Bearings roller	Bearings balls	Bearings roller
Smooth, without vibrations	0,5	1	1	1,5	2	3
Normal	0,5	1	1	1,5	2	3,5
Heavy shock loads	>1,5	>2,5	>1,5	>3	>2	>4

In the situation when more bearings of the same type are mounted close together, the amount of the static load supported by them will be determined with the following relation:

$$C_{0ri} = C_{0r} \cdot i, \text{ KN,}$$

where: C_{0ri} - basic static load of the bearing group, KN
C_{0r} - basic static load of the single bearing taken from tables, KN
i - number of bearings.

FRICTION IN BEARINGS

Friction in rolling bearings is considerably lower than in sliding bearings. Power lost through friction in bearing is generally negligible, in various bearing joints and mechanisms. If a certain frictional moment is required in some applications, the coefficient of friction for the bearing should be known.

If depends on many factors such as: bearing design, speed, direction and magnitude of load, finishing quality of active surfaces, operating temperature, lubricant, bearing material etc.

The frictional moment can be calculated accurately enough using the following equation:

$M = 0.5\mu\, P\, d$ - for radial bearings,
$M = 0.5\mu\, P\, D_m$ - for thrust bearings,

where:
M - frictional moment, N mm,
μ - coefficient of friction, table 1,
P - bearing load, N,
d - bearing bore diameter, mm,
D_m - thrust bearing mean diameter 0.5(d+D), mm.

The values of the friction coefficient μ for various bearing types are given in table 1.
The frictional moment can be more accurately determined with the equation:

$M = M_0 + M_1$,

where:
M_0 - frictional moment which is independent of the bearing load and depends on the hydrodynamic friction
M_1 - resistance moment depending on the bearing load and the size of the elastic contact surfaces,

M_0 can be calculated from:

$M_0 = f_0\,(\nu_1 n)^{2/3} \cdot D^3_m \cdot 10^{-7}$, for $n > 2000$,
$M_0 = 16\, f_0 \cdot D^3_m \cdot 10^{-6}$, for $n \le 2000$,
Where:
M_0 - frictional moment which is independent of the bearing load, N · mm
f_0 - factor which depends on the bearing type and lubricant, table 1,
n - rotational speed, r/min,
ν_1 - kinematic viscosity of lubricant at operating temperature, mm^2/s. In case of grease lubrication, calculation should be done considering the basic oil viscosity,
D_m - bearing mean diameter, mm,

M_1 can be calculated using the equation:

$M_1 = f_1 \cdot P_1 \cdot D_m$

where:
M_1 - load-dependent resistance moment, N ·mm,
f_1 - factor which depends on the bearing type and load, table 1,
P_1 - bearing combined load, determined using the equation in the table 1, N,
D_m - bearing mean diameter = 0.5 (d+D), mm.

Frictional moment for cylindrical roller bearings which also have to support axial loads

In case of these bearings, the total frictional moment is obtained by adding the frictional moment which depends on the magnitude of the axial load F_a:

$M = M_o + M_1 + M_2$
The frictional moment, can be calculated from:
$M_2 = f_2\, F_a\, D_m$, N mm

where:
M_2 - axial frictional moment, N mm,
f_2 - factor depending on bearing design and lubrication,
F_a - axial load, N,
D_m - bearing mean diameter = 0.5 (d+D), mm.

The values of the friction coefficient μ for various bearing types and factors f_o and f_1,

Table 1

Bearing type		Friction coefficient μ	Factor f_o Lubrication: grease [1)]	oil spot	oil bath	oil bath with vertic. shaft, oil jet	Factors for calculating M_1: f_1	P_1 [5)], N
Deep groove ball bearings	single row	0.0010÷0.0020	0.75-2 [2)]	1	2	4	$(8\text{-}9)10^{-4}(P_{0r}/C_{0r})^{0.55}$ [2)]	$3F_a-0.1F_r$
	double row		3	2	4	8		
Self - aligning ball bearings		0.0010÷0.0020	1.5 -2 [2)]	0.7 -1 [2)]	1.5 -2 [2)]	3-4 [2)]	$3\ 10^{-4}(P_{0r}/C_{0r})^{0.4}$	$1.4Y_2F_a-0.1F_r$
Angular contact ball bearings	single row	0.0010÷0.0025	2	1.7	3.3	6.6	$10^{-3}(P_{0r}/C_{0r})^{0.33}$	$F_a-0.1F_r$
	double row		4	3.4	6.5	13	$10^{-3}(P_{0r}/C_{0r})^{0.33}$	$1.4F_a-0.1F_r$
Four-point contact ball bearings		0.0025÷0.0045	6	2	6	9	$10^{-3}(P_{0r}/C_{0r})^{0.33}$	$1.5F_a+3.6F_r$
Cylindrical roller bearings	with cage	0.0010÷0.0025	0.6-1	1.5-2.8	2.2-4	2.2-4 [2)3)]	$(2\text{-}4)x10^{-4}$	F_r [6)]
	without cage	0.0020÷0.0040	5-10 [4)]	-	5-10	-	$5.5x10^{-4}$	F_r [6)]
Spherical roller bearings		0.0020÷0.0025	3.5-7	1.75 -3.5	3.5-7	7-14	$(1.5\text{-}8)x10^{-4}$	$1.35Y_2F_a$, $F_r/F_a<Y_2$ $F_r[1+0.3(Y_2F_a/F_r)^3]$, $F_r/F_a>Y_2$
Taper roller bearings	single row	0.0017÷0.0020	6	3	6	8-10 [2)3)]	$4x10^{-4}$	$2YF_a$
	paired	0.0030÷0.0040	12	6	12	16-20 [2)3)]	$4x10^{-4}$	$1.2Y_2F_a$
Thrust bearings	with balls	0.0010÷0.0025	5.5	0.8	1.5	3	$8x10^{-4}(F_a/C_{or})^{0.33}$	F_a
	with rollers	0.0050÷0.0070	9	-	3.5	7	$1.5x10^{-3}$	F_a
Spherical roller thrust bearings		0.0020÷0.0030	-	-	2.5-5	5-10	$(2.3\text{-}5)x10^{-4}$	F_a, $F_{rmax}<0.55F_a$

1) The values apply to normal operating conditions. In case of bearing relubrication, they apply after 2...4 operating hours.
2) The low values apply to small series bearings, the high values to large series bearings.
3) The values are valid for oil jet lubrication. For oil bath lubrication and a vertical shaft, the value should be doubled.
4) The values for low speeds up to 20% of the speed values given in the catalogue. At higher speeds they should be doubled.
5) If $P_1 < F_r$ then $P_1 = F_r$
6) For bearings which are also axially loaded, specifications for f_2 should be considered.

Simbols:
P_{or} = Equivalent static load,
C_{or} = Basic static load,
F_r = Radial component of dynamic bearing load,

F_a = Axial component dynamic bearing load,
Y, Y_2 = Axial load factors.

Values for factor f_2 Table 2

Bearing type	f_2 Lubrication oil	f_2 Lubrication grease
Bearings with cage: - E design - other bearings	 0.002 0.006	 0.003 0.009
Bearings without cage: - single row - double row	 0.003 0.009	 0.006 0.015

The values of factor f_2 in the table 2 are valid only if the value of ratio F_a/F_r doesn't exceed:

- 0.5 - for single row cylindrical roller, E design,
- 0.4 - for bearings with cage and without cage, normal design,
- 0.25 - for double row cylindrical roller bearings, without cage.

Frictional moment for sealed bearings

In case of sealed bearings, the seal washers produce additional frictions which usually exceed those arising from the bearing.
The frictional moment M_3 for a bearing which is sealed on both sides can be calculated using the following equation:

$$M_3 = (d+D)/f_3 + f_4,$$

where:
M_o - Frictional moment caused by seals, N mm,
d - Bearing bore diameter, mm,
D - Bearing outside diameter, mm,
f_3, f_4 - Factors, table 3.

Table 3

Type	Factors f_3	Factors f_4
Deep groove ball bearings 2RSR, 2RS	20	10
Single row deep groove ball bearings with extended inner ring (UC, UE, US etc.)	20	20

Starting torque

The starting torque of a rolling bearing is defined as the bearing resistance moment which must be overcome so that the bearing should start rotating from the stationary condition.

Generally, the value of the starting torque is approximately twice the load dependent moment M_1.

SPEED LIMIT

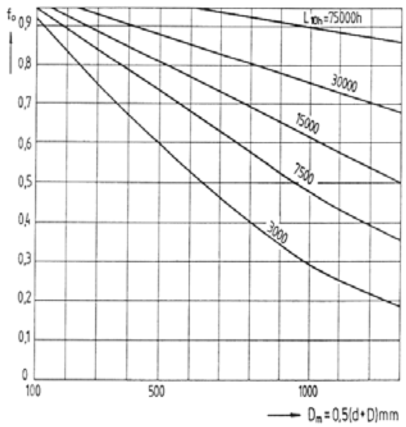


Figure 1

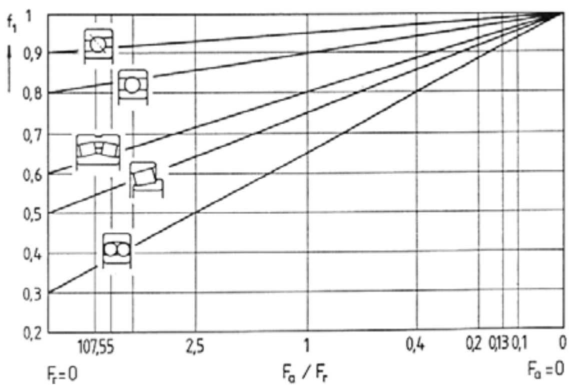


Figure 2

Maximum (limit) rotational speed to which a bearing may be subject to is indicated in bearings tables and catalogues for greasing and oil lubrication.

The values are approximate and valid if bearings are used for loads corresponding to a rating life L_h < 150.000 hours and operate in the following conditions:

- good rigidity of the shaft and housing
- adequate greasing conditions
- heat dispersing conditions (maximum operating temperature 70°C)
- adequate sealing

In the case in which the bearings operating conditions are not known it is recommended that the effective rotational speed not to exceed 75% of the rotational speed indicated in the catalogue.

For heavy loads applied to bearings with the mean diameter greater than 100 mm when the rating life L_h is smaller than 75.000 hours, the limit rotational speed indicated by the catalogue shall be multiplied with the factor f_0 from figure 1.

For combined loads applied to bearings, the rotational speed indicated in the catalogue shall be multiplied with the factor f_1 from figure 2

The increase of the maximum rotational speed above the limit value mentioned in the catalogue could be realized both through the use of higher precision bearing classes in the same time with the increase of the shaft and housing manufacturing precision and through the improvement of the greasing and cooling conditions.

Table 1 presents the multiplication factors of the limit rotational speed.

For high rotational speed the bearings of small size series are to be preferred.

In case of pure radial loads carrying, radial ball or roller bearings can support the higher rotational speeds.

In the case of combined loads carrying, even in the case in which the axial loads are foremost will be preferred radial - axial ball bearings.

In the case of spherical (ball or roller) bearings when axial loads are predominating, it is recommended to reduce the upper limit of the maximum rotational speed.

For all bearings used at high rotational speeds it is recommended the use of a radial clearance greater than the normal one (groups C3, C4, C5).

Table 1

Constructive features/ Precision class	Greasing type	Bearing type	
		Radial ball bearing. Radial roller bearing	Axial ball bearing
Special cage / P6	Oil circulation	1,6 - 1,8	1,1 - 1,3
Special cage / P5	Cooled oil circulation Oil mist	1,8 - 2,1	1,3 - 1,4
Special cage / P4	Cooled oil circulation Oil spot	2,1 - 2,4	1,3 - 1,4

TAPERED BORE BEARINGS

Taper 1:12

Deviations µm — Table 15

d mm		Normal tolerance class,P6					Tolerance class P5				
		Δ_{dmp}		V_{dp} 1)	$\Delta_{d1mp}-\Delta_{dmp}/2$		Δ_{dmp}		V_{dp} 1)	$\Delta_{d1mp}-\Delta_{dmp}/2$	
over	up to	high	low	max.	high	low	high	low	max.	high	low
18	30	+21	0	13	+21	0	+13	0	13	+13	0
30	50	+25	0	15	+25	0	+16	0	15	+16	0
50	80	+30	0	19	+30	0	+19	0	19	+19	0
80	120	+35	0	25	+35	0	+22	0	22	+22	0
120	180	+40	0	31	+40	0	+25	0	25	+25	0
180	250	+46	0	38	+46	0	+29	0	29	+29	0
250	315	+52	0	44	+52	0	+32	0	32	+32	0
315	400	+57	0	50	+57	0	+36	0	36	+36	0
400	500	+63	0	56	+63	0	+40	0	-	+40	0
500	630	+70	0	-	+70	0	+44	0	-	+44	0
630	800	+80	0	-	+80	0	+50	0	-	+50	0
800	1000	+90	0	-	+90	0	+56	0	-	+56	0
1000	1250	+105	0	-	+105	0	+66	0	-	+66	0
1250	1600	+125	0	-	+125	0	+78	0	-	+78	0
1600	2000	+150	0	-	+150	0	+92	0	-	+92	0

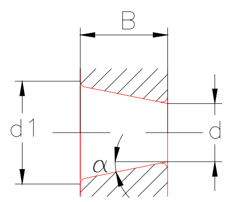
1) Applies in all single radial planes of the bore

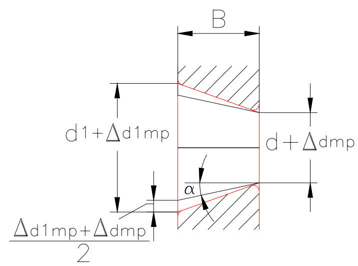
Taper 1:30

Deviations in µm — Table 16

d mm		Normal tolerance class				
		Δ_{dmp}		V_{dp} 1)	$\Delta_{d1mp}-\Delta_{dmp}/2$	
over	up to	high	low	max.	high	low
80	120	+20	0	25	+40	0
120	180	+25	0	31	+50	0
180	250	+30	0	38	+55	0
250	315	+35	0	44	+60	0
315	400	+40	0	50	+65	0
400	500	+45	0	56	+75	0
500	630	+50	0	63	+85	0
630	800	+75	0	-	+100	0
800	1000	+100	0	-	+100	0
1000	1250	+125	0	-	+115	0
1250	1600	+160	0	-	+125	0
1600	2000	+200	0	-	+150	0

1) Applies in all singular planes





Tapered bore
Half angle of taper, α

α=2°23'9.4"(taper 1:12)
α=0°57'17.4"(taper 1:30)

Nominal diameter, d1 at the theoretical large end of bore

$d_1 = d + \frac{1}{12}B$ (taper 1:12)
$d_1 = d + \frac{1}{30}B$ (taper 1:30)

GENERALITIES. SYMBOLS.

Generalities

Bearing tolerances have been internationally standardized in accordance with ISO 492, 199, 582 and ISO 1132.

Bearings are generally manufactured to the tolerance class P0. At request, they can also be manufactured to the tolerance classes P6, P6X, P5, P4 and P2. These bearings are used for special applications, such as very accurate shaft guidance or very high speeds.

The values of the limit deviation for these tolerance classes are given for:

-the overall dimensions of:

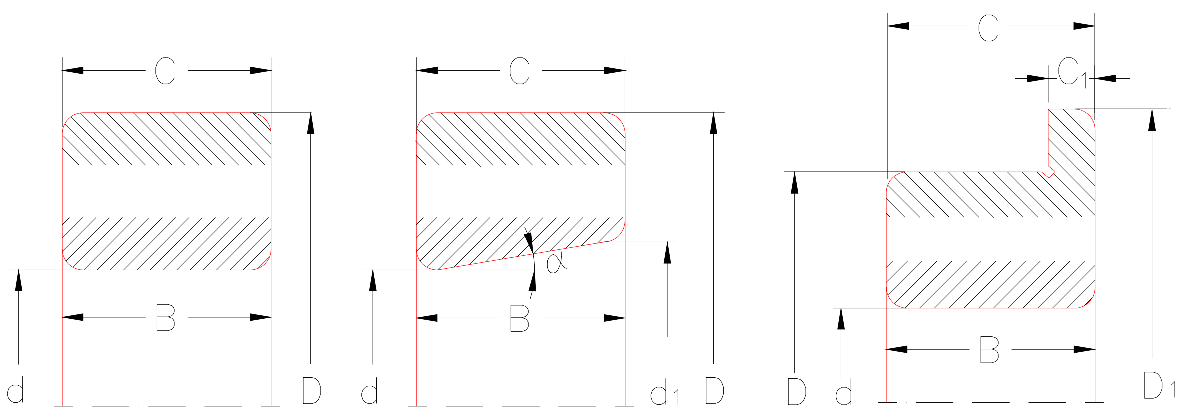
- deep groove ball bearings, spherical roller bearings, cylindrical roller bearings,
- rulmentilor radiali-axiali cu role conice, dimensiuni in mm si in inci,
- tapered bore bearings,
- thrust ball bearings, angular contact thrust ball bearings, cylindrical roller thrust bearings,
- mounting chamfer.

Symbols

d -nominal bore diameter or shaft washer nominal bore diameter for thrust bearings
d_1 -nominal diameter at the theoretical large end of the tapered bore,
d_2 -nominal bore diameter of the shaft washer for double direction thrust bearings,
d_s -deviation of single bore diameter,
d_{psmax} -maximum bore diameter, in a single radial plane
d_{psmin} -minimum bore diameter, in a single radial plane,
Δ_{ds} -deviation of a single bore diameter, $\Delta_{ds} = d_s - d$,
d_{mp} -mean bore diameter, in a single radial plane $d_{mp} = (d_{psmax} + d_{psmin})/2$,
Δ_{dmp} -deviation of the mean bore diameter in a single radial plane; or deviation of the mean diameter at the theoretical small end of the tapered bore, in case of tapered bore bearings; or deviation of the mean bore diameter of the shaft washer in a single radial plane for single direction thrust bearings $\Delta_{dmp} = d_{mp} - d$,
d_{1mp} -mean diameter at the large theoretical end of the tapered bore in a single plane,
Δ_{d1mp} -deviation of the mean diameter at the theoretical large end of the tapered bore, $\Delta_{d1mp} = d_{1mp} - d$,
Δ_{d2mp} -deviation of the mean bore diameter of the shaft washer for a double direction thrust bearing, in a single radial plane
V_{dp} -bore diameter variation in a single radial plane, or bore diameter variation of the shaft washer in a single radial plane, for single direction thrust bearings, $V_{dp} = d_{psmax} - d_{psmin}$,
V_{d2p} -bore diameter variation of the shaft washer for double direction thrust bearings, in a single radial plane
V_{dmp} -mean bore diameter variation (valid only for cylindrical bore), $V_{dmp} = d_{mpmax} - d_{mpmin}$,
-nominal half-angle of the tapered bore
D -nominal outside diameter or housing washer nominal diameter,
D_1 -nominal outside diameter of the outer ring rib,
D_s -single outside diameter,
D_{psmax} -maximum outside diameter in a single radial plane,
D_{psmin} -minimum outside diameter in a single radial plane,
Δ_{Ds} -deviation of the single outside diameter, $\Delta_{Ds} = D_s - D$,
D_{mp} -mean outsite diameter, in a single plane $D_{mp} = (D_{psmax} + D_{psmin})/2$,
Δ_{Dmp} -deviation of the mean outside diameter in a single radial plane; or deviation of the mean diameter of housing washer in a single radial plane, for thrust bearings $\Delta_{Dmp} = D_{mp} - D$,
V_{Dp} -outside diameter variation in a single radial plane; or housing washer diameter variation in a single radial plane for double direction thrust bearings, $V_{Dp} = D_{psmax} - D_{psmin}$,
V_{Dmp} -mean outside diameter variation $V_{Dmp} = D_{mpmax} - D_{mpmin}$,
B -nominal width of the inner ring,
B_s -single width of the inner ring,
Δ_{Bs} -inner ring single width deviation, $\Delta_{Bs} = B_s - B$,
V_{Bs} -inner ring single width variation, $V_{Bs} = B_{smax} - B_{smin}$,
C -nominal width of the outer ring,

C_s -single width of the outer ring

Δ_{Cs} -deviation of outer ring single width, $\Delta_{Cs} = C_s - C$,
V_{Cs} -single width variation of the outer ring, $V_{Cs} = C_{smax} - C_{smin}$,
K_{ia} -radial runout of assembled bearing inner ring,
K_{ea} -radial runout of assembled bearing outer ring,
S_d -side face runout with reference to bore of the inner ring,
S_D -variation in inclination of outsite cylindrical surface to outer ring side face
S_{ia} -side face runout of assembled inner ring with reference to raceway,
S_{ea} -side face runout of assembled outer ring with reference to raceway,
S_i -thickness variation measured from middle of raceway to back seating face of shaft washer,
S_e -thickness variation measured from middle of raceway to back face of housing washer
Δ_{Hs} -deviation of mounting height of single direction thrust ball and roller bearings,
Δ_{H1s} -deviation of mounting height of thrust ball bearings with sphered housing washer,
Δ_{H2s} -deviation of mouting height of double direction thrust ball and roller bearings,
Δ_{H3s} -deviation of mounting height of double direction thrust ball bearings with sphered housing washer
Δ_{H4s} -deviation of mounting height of spherical roller thrust bearings.



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RADIAL BEARINGS (EXCEPTING TAPERED ROLLER BEARINGS)
Tolerance class P0

Inner ring

Deviations in µm — Table 1

d mm		Δ_{dmp}		V_{dp} Diameter series			V_{dmp}	K_{ia}	Δ_{Bs}			V_{Bs}
				7,8,9	0,1	2,3,4			all	normal	modified[2)]	
over	up to	high	low	max.	max.	max.	max.	max.	high	low	low	max.
0.6 [1)]	2.5	0	-8	10	8	6	6	10	0	-40	-	12
2.5	10	0	-8	10	8	6	6	10	0	-120	-250	15
10	18	0	-8	10	8	6	6	10	0	-120	-250	20
18	30	0	-10	13	10	8	8	13	0	-120	-250	20
30	50	0	-12	15	12	9	9	15	0	-120	-250	20
50	80	0	-15	19	19	11	11	20	0	-150	-380	25
80	120	0	-20	25	25	15	15	25	0	-200	-380	25
120	180	0	-25	31	31	19	19	30	0	-250	-500	30
180	250	0	-30	38	38	23	23	40	0	-300	-500	30
250	315	0	-35	44	44	26	26	50	0	-350	-500	35
315	400	0	-40	50	50	30	30	60	0	-400	-630	40
400	500	0	-45	56	56	34	34	65	0	-450	-	50
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

1) This value included,
2) If refers to isolated bearing ring for paired mounting or stack mounting.

Outer ring

Deviations in µm — Table 2

D mm		Δ_{Dmp}		V_{Dp}[3)] Open bearings			Shielded Bearings[2)]	V_{Dmp}[3)]	K_{ea}	Δ_{Cs}		V_{Cs}
				Diameter series								
				7,8,9	0,1	2,3,4	2,3,4					
over	up to	high	low	max.	max.	max.	max.	max.	max.	high	low	max.
2.5 [1)]	6	0	-8	10	8	6	10	6	15	Values are identical to Δ_{Bs} for the inner ring of the same bearing.		
6	18	0	-8	10	8	6	10	6	15			
18	30	0	-9	12	9	7	12	7	15			
30	50	0	-11	14	11	8	16	8	20			
50	80	0	-13	16	13	10	20	10	25			
80	120	0	-15	19	19	11	26	11	35			
120	150	0	-18	23	23	14	30	14	40			
150	180	0	-25	31	31	19	38	19	45			
180	250	0	-30	38	38	23	-	23	50			
250	315	0	-35	44	44	26	-	26	60			
315	400	0	-40	50	50	30	-	30	70			
400	500	0	-45	56	56	34	-	34	80			
500	630	0	-50	63	63	38	-	38	100			
630	800	0	-75	94	94	55	-	55	120			
800	1000	0	-100	125	125	75	-	75	140			
1000	1250	0	-125	-	-	-	-	-	160			
1250	1600	0	-160	-	-	-	-	-	190			
1600	2000	0	-200	-	-	-	-	-	220			
2000	2500	0	-250	-	-	-	-	-	250			

1) This value included,
2) For bearings of diameter series 7,8,9,0, and 1 values are not indicated,
3) Values are valid before mounting the snap ring or shields or after their dismounting.

Tolerance class P6

Inner ring

Deviations in µm Table 3

d mm		Δ_{dmp}		V_{dp} Diameter series			V_{dmp}	K_{ia}	Δ_{Bs}			V_{Bs}
				7,8,9	0,1	2,3,4			all	normal	modifiedt [2)]	
over	up to	high	low	max.	max.	max.	max.	max.	high	low	low	max.
0.6 [1)]	2.5	0	-7	9	7	5	5	5	0	-40	-	12
2.5	10	0	-7	9	7	5	5	6	0	-120	-250	15
10	18	0	-7	9	7	5	5	7	0	-120	-250	20
18	30	0	-8	10	8	6	6	8	0	-120	-250	20
30	50	0	-10	13	10	8	8	10	0	-120	-250	20
50	80	0	-12	15	15	9	9	10	0	-150	-380	25
80	120	0	-15	19	19	11	11	13	0	-200	-380	25
120	180	0	-18	23	23	14	14	18	0	-250	-500	30
180	250	0	-22	28	28	17	17	20	0	-300	-500	30
250	315	0	-25	31	31	19	19	25	0	-350	-500	35
315	400	0	-30	38	38	23	23	30	0	-400	-630	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

1) This value included,
2) It refers to isolated bearing ring for paired mounting or stack mounting.

Outer ring

Deviations in µm Table 4

D mm		Δ_{Dmp}		V_{Dp} [3)] Open bearings			Shielded Bearings [2)]	V_{Dmp} [3)]	K_{ea}	Δ_{Cs}		V_{Cs}
				Diameter series								
				7,8,9	0,1	2,3,4	2,3,4					
over	up to	high	low	max.	max.	max.	max.	max.	max.	high	low	max.
2.5 [1)]	6	0	-7	9	7	5	9	5	8	Values are identical to Δ_{Bs} and V_{Bs} for the inner ring		
6	18	0	-7	9	7	5	9	5	8			
18	30	0	-8	10	8	6	10	6	9			
30	50	0	-9	11	9	7	13	7	10			
50	80	0	-11	14	11	8	16	8	13			
80	120	0	-13	16	16	10	20	10	18			
120	150	0	-15	19	19	11	25	11	20			
150	180	0	-18	23	23	14	30	14	23			
180	250	0	-20	25	25	15	-	15	25			
250	315	0	-25	31	31	19	-	19	30			
315	400	0	-28	35	35	21	-	21	35			
400	500	0	-33	41	41	25	-	25	40			
500	630	0	-38	48	48	29	-	29	50			
630	800	0	-45	56	56	34	-	34	60			
800	1000	0	-60	75	75	45	-	45	75			

1) This value included,
2) For bearings of diameter series 7,8 and 9 values are not indicated,
3) Values are valid before mounting the snap ring or shields or after their dismounting.

Tolerance class P5

Inner ring

Deviations in µm — Table 5

d mm		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} 2)	Δ_{Bs} all	Δ_{Bs} normal	Δ_{Bs} modified 3)	V_{Bs}
over	up to	high	low	max.	max.	max.	max.	max.	max.	high	low	low	max.
0.6 1)	2.5	0	-5	5	4	3	4	7	7	0	-40	-250	5
2.5	10	0	-5	5	4	3	4	7	7	0	-40	-250	5
10	18	0	-5	5	4	3	4	7	7	0	-80	-250	5
18	30	0	-6	6	5	3	4	8	8	0	-120	-250	5
30	50	0	-8	8	6	4	5	8	8	0	-120	-250	5
50	80	0	-9	9	7	5	5	8	8	0	-150	-250	6
80	120	0	-10	10	8	5	6	9	9	0	-200	-380	7
120	180	0	-13	13	10	7	8	10	10	0	-250	-380	8
180	250	0	-15	15	12	8	10	11	13	0	-300	-500	10
250	315	0	-18	18	14	9	13	13	15	0	-350	-500	13
315	400	0	-25	25	18	12	15	15	20	0	-400	-630	15

1)This value included,
2) Applies only to ball bearings.
3) If refers to single bearing ring for paired mounting or stack mounting.

Outer ring

Deviations in µm — Table 6

D mm		Δ_{Dmp}		V_{Dp} 2) 7,8,9	V_{Dp} 2) 0,1,2,3.4	V_{Dmp} 3)	K_{ea}	S_D	S_{ea} 3)	Δ_{Cs}		V_{Cs}
over	up to	high	low	max.	max.	max.	max.	max.	max.	high	low	max.
2.5 1)	6	0	-5	5	4	3	5	8	8	Identical to Δ_{Bs} and V_{Bs} for the inner ring		5
6	18	0	-5	5	4	3	5	8	8			5
18	30	0	-6	6	5	3	6	8	8			5
30	50	0	-7	7	5	4	7	8	8			5
50	80	0	-9	9	7	5	8	8	10			6
80	120	0	-10	10	8	5	10	9	11			8
120	150	0	-11	11	8	6	11	10	13			8
150	180	0	-13	13	10	7	13	10	14			8
180	250	0	-15	15	11	8	15	11	15			10
250	315	0	-18	18	14	9	18	13	18			11
315	400	0	-20	20	15	10	20	13	20			13
400	500	0	-23	23	17	12	23	15	23			15
500	630	0	-28	28	21	14	25	18	25			18
630	800	0	-35	35	26	18	30	20	30			20

1) This value included,
2) Do not apply to shielded bearings.
3) Apply to ball bearings.

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Tolerance class P4

Inner ring

Deviations in µm Table 7

d mm		Δ_{dmp},Δ_{ds} 2)		V_{dp} Diameter series 7,8,9	V_{dp} Diameter series 0,1,2,3,4	V_{dmp}	K_{ia}	S_d	S_{ia} 3)	Δ_{Bs} all	Δ_{Bs} normal	Δ_{Bs} modified 4)	V_{Bs}
over	up to	high	low	max.	max.	max.	max.	max.	max.	high	low	low	max.
0.6 1)	2.5	0	-4	4	3	2	2.5	3	3	0	-40	-250	2.5
2.5	10	0	-4	4	3	2	2.5	3	3	0	-40	-250	2.5
10	18	0	-4	4	3	2	2.5	3	3	0	-80	-250	2.5
18	30	0	-5	5	4	2.5	3	4	4	0	-120	-250	2.5
30	50	0	-6	6	5	3	4	4	4	0	-120	-250	3
50	80	0	-7	7	5	3.5	4	5	5	0	-150	-250	4
80	120	0	-8	8	6	4	5	5	5	0	-200	-380	4
120	180	0	-10	10	8	5	6	6	7	0	-250	-380	5
180	250	0	-12	12	9	6	8	7	8	0	-300	-500	6

1) This value included.
2) Apply only to bearings of diameter series 0,1,2,3,4.
3) Apply only to ball bearings.
4) It refers to single bearing ring for paired mounting or stack mounting.

Outer ring

Deviations in µm Table 8

D mm		Δ_{Dmp},Δ_{Ds} 2)		V_{Dp} 3) Diameter series 7,8,9	V_{Dp} 3) Diameter series 0,1,2,3,4	V_{Dmp}	K_{ea}	S_D	S_{ea} 4)	Δ_{Cs}		V_{Cs}
over	up to	high	low	max.	max.	max.	max.	max.	max.	high	low	low
2.5 1)	6	0	-4	4	3	2	3	4	5	Identical to Δ_{Bs} and V_{Bs} for the inner ring		2.5
6	18	0	-4	4	3	2	3	4	5			2.5
18	30	0	-5	5	4	2.5	4	4	5			2.5
30	50	0	-6	6	5	3	5	4	5			2.5
50	80	0	-7	7	5	3.5	5	4	5			3
80	120	0	-8	8	6	4	6	5	6			4
120	150	0	-9	9	7	5	7	5	7			5
150	180	0	-10	10	8	5	8	5	8			5
180	250	0	-11	11	8	6	10	7	10			7
250	315	0	-13	13	10	7	11	8	10			7
315	400	0	-15	15	11	8	13	10	13			8

1) This value included,
2) Apply to bearings of diameter series 0,1,2,3 and 4.
3) Do not apply to sealed and shielded bearings.
4) Apply only to ball bearings.

Tolerance class SP

Inner ring

Deviations in µm — Table 11

d mm		Cylindrical bore			Tapered bore					Δ_{Bs}		V_{Bs}	K_{ia}	S_d	S_{ia}
		$\Delta_{dmp}, \Delta_{ds}$		V_{dp}	Δ_{ds}		V_{dp}	$\Delta_{d1mp}-\Delta_{dmp}$							
over	up to	low	high	max.	low	high	max.	low	high	low	high	max.	max.	max.	max.
-	18	-5	0	3	-	-	-	-		-100	0	5	3	8	8
18	30	-6	0	3	0	+10	3	0	+4	-100	0	5	3	8	8
30	50	-8	0	4	0	+12	4	0	+4	-120	0	5	4	8	8
50	80	-9	0	5	0	+15	5	0	+5	-150	0	6	4	8	8
80	120	-10	0	5	0	+20	5	0	+6	-200	0	7	5	9	9
120	180	-13	0	7	0	+25	7	0	+8	-250	0	8	6	10	10
180	250	-15	0	8	0	+30	8	0	+10	-300	0	10	8	11	13
250	315	-18	0	9	0	+35	9	0	+12	-350	0	13	10	13	15
315	400	-23	0	12	0	+40	12	0	+13	-400	0	15	12	15	20
400	500	-28	0	14	0	+45	14	0	+15	-450	0	25	12	18	23
500	630	-35	0	18	0	+50	18	0	+17	-500	-	30	15	20	25

Outer ring

Deviations in µm — Table 12

D mm		$\Delta_{Dmp}, \Delta_{Ds}$		V_{Dp}	K_{ea}	S_D	Sea	Δ_{Cs}	V_{Cs}
over	up to	low	high	max.	max.	max.	max.		
30	50	-7	0	4	5	8	8	Identical to Δ_{Bs} and V_{Bs} for the inner ring	
50	80	-9	0	5	5	8	10		
80	120	-10	0	5	6	9	11		
120	150	-11	0	6	7	10	13		
150	180	-13	0	7	8	10	14		
180	250	-15	0	8	10	11	15		
250	315	-18	0	9	11	13	18		
315	400	-20	0	10	13	13	20		
400	500	-23	0	12	15	15	23		
500	630	-28	0	14	17	18	25		
630	800	-35	0	18	20	20	30		

TAPERED BORE BEARINGS

Taper 1:12

Deviations µm — Table 15

d mm		Normal tolerance class,P6					Tolerance class P5				
		Δ_{dmp}		V_{dp} 1)	$\Delta_{d1mp}-\Delta_{dmp}/2$		Δ_{dmp}		V_{dp} 1)	$\Delta_{d1mp}-\Delta_{dmp}/2$	
over	up to	high	low	max.	high	low	high	low	max.	high	low
18	30	+21	0	13	+21	0	+13	0	13	+13	0
30	50	+25	0	15	+25	0	+16	0	15	+16	0
50	80	+30	0	19	+30	0	+19	0	19	+19	0
80	120	+35	0	25	+35	0	+22	0	22	+22	0
120	180	+40	0	31	+40	0	+25	0	25	+25	0
180	250	+46	0	38	+46	0	+29	0	29	+29	0
250	315	+52	0	44	+52	0	+32	0	32	+32	0
315	400	+57	0	50	+57	0	+36	0	36	+36	0
400	500	+63	0	56	+63	0	+40	0	-	+40	0
500	630	+70	0	-	+70	0	+44	0	-	+44	0
630	800	+80	0	-	+80	0	+50	0	-	+50	0
800	1000	+90	0	-	+90	0	+56	0	-	+56	0
1000	1250	+105	0	-	+105	0	+66	0	-	+66	0
1250	1600	+125	0	-	+125	0	+78	0	-	+78	0
1600	2000	+150	0	-	+150	0	+92	0	-	+92	0

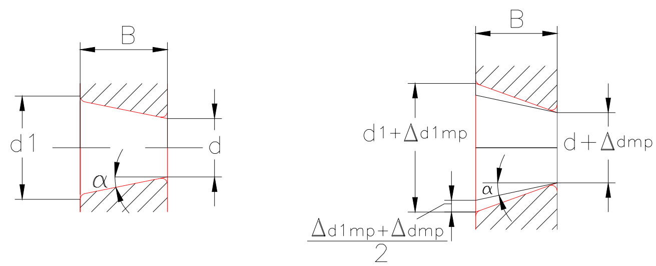
1) Applies in all single radial planes of the bore

Taper 1:30

Deviations in µm — Table 16

d mm		Normal tolerance class				
		Δ_{dmp}		V_{dp} 1)	$\Delta_{d1mp}-\Delta_{dmp}/2$	
over	up to	high	low	max.	high	low
80	120	+20	0	25	+40	0
120	180	+25	0	31	+50	0
180	250	+30	0	38	+55	0
250	315	+35	0	44	+60	0
315	400	+40	0	50	+65	0
400	500	+45	0	56	+75	0
500	630	+50	0	63	+85	0
630	800	+75	0	-	+100	0
800	1000	+100	0	-	+100	0
1000	1250	+125	0	-	+115	0
1250	1600	+160	0	-	+125	0
1600	2000	+200	0	-	+150	0

1) Applies in all singular planes



Tapered bore
Half angle of taper, α

α=2°23'9.4"(taper 1:12)
α=0°57'17.4"(taper 1:30)

Nominal diameter, d1 at the theoretical large end of bore

$d_1 = d + \frac{1}{12} B$ (taper 1:12)
$d_1 = d + \frac{1}{30} B$ (taper 1:30)

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THRUSTBEARINGS

Shaft washer

Deviations in µm Table 17

d si d_2 mm		P0;P6;P5			P4;P2		
		Δ_{dmp} Δ_{d2mp}		V_{dp} V_{d2p}	Δ_{dmp} Δ_{d2mp}		V_{dp} V_{dp}
over	up to	high	low	max.	high	low	max.
-	18	0	-8	6	0	-7	5
18	30	0	-10	8	0	-8	6
30	50	0	-12	9	0	-10	8
50	80	0	-15	11	0	-12	9
80	120	0	-20	15	0	-15	11
120	180	0	-25	19	0	-18	14
180	250	0	-30	23	0	-22	17
250	315	0	-35	26	0	-25	19
315	400	0	-40	30	0	-30	23
400	500	0	-45	34	0	-35	26
500	630	0	-50	38	0	-40	30
630	800	0	-75	-	0	-50	-
800	1000	0	-100	-	-	-	-
1000	1250	0	-125	-	-	-	-

Housing washer

Deviations in µm Table 18

D mm		P0;P6;P5			P4;P2		
		Δ_{Dmp}		V_{Dp}	Δ_{Dmp}		V_{Dp}
over	up to	high	low	max.	high	low	max.
10[1)]	18	0	-11	8	0	-7	5
18	30	0	-13	10	0	-8	6
30	50	0	-16	12	0	-9	7
50	80	0	-19	14	0	-11	8
80	120	0	-22	17	0	-13	10
120	180	0	-25	19	0	-15	11
180	250	0	-30	23	0	-20	15
250	315	0	-35	26	0	-25	19
315	400	0	-40	30	0	-28	21
400	500	0	-45	34	0	-33	25
500	630	0	-50	38	0	-38	29
630	800	0	-75	55	0	-45	34
800	1000	0	-100	75	-	-	-
1000	1250	0	-125	-	-	-	-
1250	1600	0	-160	-	-	-	-

1) This value included,

Variation of shaft washer and housing washer thickness

Abateri in µm — Table 19

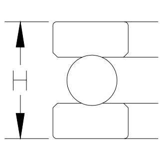
d* mm		S_i					S_e
		P0	P6	P5	P4	P2	P0,P6,P5,P4,P2
over	up to	max.	max.	max.	max.	max.	max.
-	18	10	5	3	2	1	Identical to Si for the shaft washer
18	30	10	5	3	2	1,2	
30	50	10	6	3	2	1,5	
50	80	10	7	4	3	2	
80	120	15	8	4	3	2	
120	180	15	9	5	4	3	
180	250	20	10	5	4	3	
250	315	25	13	7	5	4	
315	400	30	15	7	5	4	
400	500	30	18	9	6	-	
500	630	35	21	11	7	-	
630	800	40	25	13	8	-	
800	1000	45	30	15	-	-	
1000	1250	50	35	18	-	-	

* The values of S_i and S_e admitted for double direction thrust bearings are equal to the coresponding values of the single direction thrust bearings and are functions of the bore diameter d, of the single direction bearings.

Assembled thrust bearings
Bearing height

Deviations in µm
Table 20

d* mm		Δ_{Hs}		Δ_{H1s}		Δ_{H2s}		Δ_{H3s}		Δ_{H4s}	
over	up to	high	low	high	low	high	low	high	low	high	low
-	30	+20	-250	+100	-200	+150	-400	+300	-400	+20	-300
30	50	+20	-250	+100	-250	+150	-400	+300	-400	+20	-300
50	80	+20	-300	+100	-300	+150	-500	+300	-500	+20	-400
80	120	+25	-300	+150	-300	+200	-500	+400	-500	+25	-400
120	180	+25	-400	+150	-400	+200	-600	+400	-600	+25	-500
180	250	+30	-400	+150	-400	+250	-600	+500	-600	+30	-500
250	315	+40	-400	+200	-400	+350	-700	+600	-700	+40	-700
315	400	+40	-500	+200	-500	+350	-700	+600	-700	+40	-700
400	500	+50	-500	+300	-500	+400	-900	+750	-900	+50	-900
500	630	+60	-600	+350	-600	+500	-1100	+900	-1100	+60	-1200
630	800	+70	-750	+400	-750	+600	-1300	+1100	-1300	+70	-1400
800	1000	+80	-1000	+450	-1000	+700	-1500	+1300	-1500	+80	-1800
1000	1250	+100	-1400	+500	-1400	+900	-1800	+1600	-1800	+100	-2400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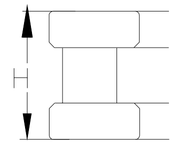
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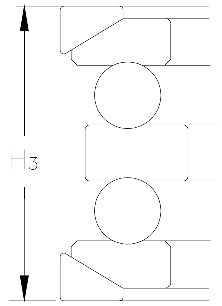
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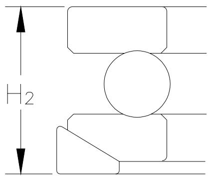
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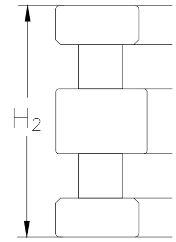
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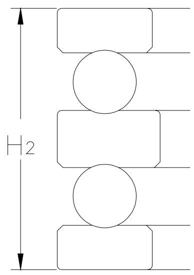
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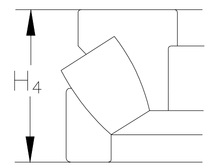
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ANGULAR CONTACT THRUST BALL BEARINGS, DOUBLE DIRECTION

Tolerance class SP

Inner ring

Deviations in µm Table 21

d mm		Δ_{ds}		S_{ia}	H_s	
over	up to	high	low	max.	high	low
0	18	+1	-8	3	+50	-80
18	30	+1	-9	3	+50	-80
30	50	+1	-11	3	+60	-100
50	80	+2	-14	4	+70	-120
80	120	+3	-18	4	+85	-140
120	180	+3	-21	5	+95	-160
180	250	+4	-26	5	+120	-200

Outer ring

Deviations in µm Table 22

D mm		Δ_{Ds}		Δ_{Cs}		S_{ea}
over	up to	high	low	high	low	max.
30	50	-20	-27	0	-30	Identical to S_{ia} for the inner ring
50	80	-24	-33	0	-30	
80	120	-28	-38	0	-30	
120	150	-33	-44	0	-30	
150	180	-33	-46	0	-30	
180	250	-37	-52	0	-30	
250	315	-41	-59	0	-30	

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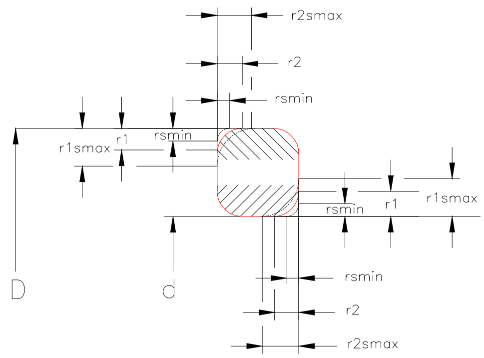
MOUNTING CHAMFER DIMENSION TOLERANCES

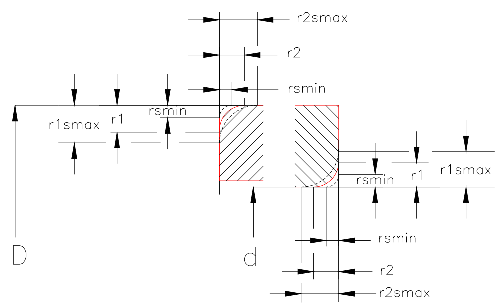
Symbols:
r_1,r_3 - chamfer dimension in radial direction,
r_2,r_4 - chamfer dimension in axial direction,
r_s min - general symbol for minimum limit of r_1,r_2,r_3,r_4,
r_{1s} max., r_{3s} max - maximum dimension in radial direction,
$r_{2s\ max}$, $r_{4s\ max}$ - maximum dimension in axial direction.

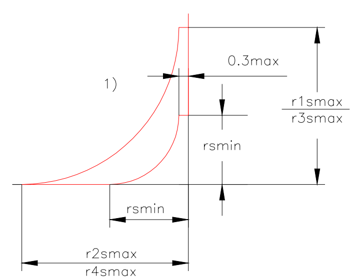
Mounting chamfer dimension limits for radial and thrust bearings

Values in mm Table 23

$r_{s\ min}$	d		Radial bearings		Thrust bearings
			r_{1s},r_{3s}	r_{2s},r_{4s}	r_{1s},r_{2s}
	over	up to	max.	max.	max.
0.1	-	-	0.2	0.4	0.2
0.15	-	-	0.3	0.6	0.3
0.2	-	-	0.5	0.8	0.5
0.3	-	40	0.8	1	0.8
	40	-	0.8	1	0.8
0.6	-	40	1	2	1.5
	40	-	1.3	2	1.5
1	-	50	1.5	3	2.2
	50	-	1.9	3	2.2
1.1	-	120	2	3.5	2.7
	120	-	2.5	4	2.7
1.5	-	120	2.3	4	3.5
	120	-	3	5	3.5
2	-	80	3	4.5	4
	220	-	3.8	6	4
	80	220	3.5	5	4
2.1	-	100	3.8	6	4.5
	-	280	4	6.5	4.5
	280	-	4.5	7	4.5
2.5	100	280	4.5	6	-
	280	-	5	7	-
3	-	280	5	8	5.5
	280	-	5.5	8	5.5
4	-	-	6.5	9	6.5
5	-	-	8	10	8
6	-	-	10	13	10
7.5	-	-	12.5	17	12.5
9.5	-	-	15	19	15
12	-	-	18	24	18
15	-	-	21	30	21
19	-	-	25	38	25







1) Only for d<30mm.

MATERIALS FOR ROLLING BEARINGS

Generalities

Due to various operating conditions and intricate aspects of deterioration phenomena, direct connections between mechanical characteristics and materials used for bearing manufacturing have been ascertained. Experimental studies proved that the following characteristics have to be considered, when appreciating the quality of bearing steels: rating life and contact fatigue loading, hardness at environment temperature and high temperatures, coefficient of expansion, tenacity, corrosion resistance and metallurgical conversion characteristics.

In case of normal applications and operating conditions, only the first two characteristics are of importance, the other being of importance only in case of bearings used for special applications.

Material behavior when being loaded at fatigue contact is difficult to be estimated due to the complexily of the factors involved while hardness can be estimated by classic methods.

These led to the selection of some steels, which are able to satisfy the main demands of normal and special operating conditions. The steels that meet the requirements for rings and rolling elements manufacturing are the following:

Chrome-alloy bearing steels

Steels with high carbon content 1 % and with chrome 1.5% according ISO 683-17 have been chosen for bearing rings and rolling elements.
Table 1 shows the chemical content of bearing steels used in Romania.

Table 1

Standards	Symbol	C	Si	Mn	P max.	S max.	Cr	Mo
		%						
ISO 683-17	100Cr6	0.93-1.05	0.15-0.35	0.25-0.45	0.025	0.015	1.35-1.60	max.0.10
	100CrMnSi6-4	0.93-1.05	0.45-0.75	1.00-1.20	0.025	0.015	1.40-1.65	max.0.10

Case - hardening steels

Although case-hardening steels are not usually selected for bearing manufacturing, for certain applications they can be successfully used.

These steels are generally recommended for large-sized bearings and where bearings are operated under shock, loads and vibrations.

Bearings manufactured of case-hardening steels are less liable to casual failure due to the ductile and soft core of these steels.

The case-hardening bearing steels used and the chemical content are according ISO 683-17.

Bearing cages

Bearing cages are of great importance for bearing design.

The main purpose of the cage is to prevent immediate contact between two neighboring rolling elements and to guide them on raceways. Where bearings are of separable design, the cage also serves to retain the rolling elements when one bearing ring is removed during mouting and dismounting.

Considering the cage manufacturing technologies, they can be classified as follows:

-Pressed cages of steel sheet, low carbon content, for extra-deep drawing.

-Polyamide cages are used for some small and medium-sized bearings due to the following properties:

- low density
- high elasticity
- low wear at sliding movement
- low inertia moment

LOCATING BEARINGS AND NON-LOCATING BEARINGS

Radial and axial loads in bearing units can be transmitted by locating and non-locating bearings.

A locating bearing is generally used for medium and large -sized shafts that can reach high temperatures during operation. It has to support radially the shaft assembly and to locate it axially in both directions.

A non-locating bearing supports the shaft assembly only radially. It also allows axial displacement in relation to the housing to take place so that additional axial loading is avoided.

Axial displacement can take place either in the housing bore seating or in the bearing itself.

In case the shaft is supported by more than two bearings, only one of them will be a locating bearing and it will be the one with the lightest radial load.

In case of small-sized shafts, two non-locating bearings with limited displacement can be used. Each of them can accommodate axial loads in a single direction, having thus mutual location.

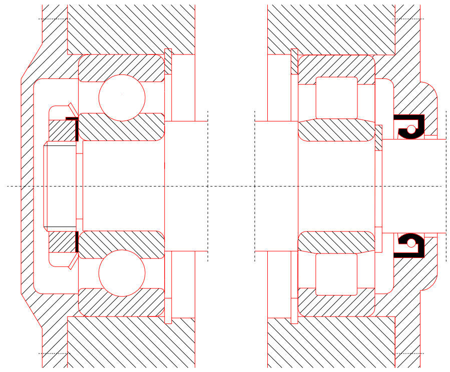
Fig.1 shows a few of the most reprezentative applications of locating and non-locating bearings, as follows:

a) The locating bearing is a single row deep groove ball bearing and the non-locating one is a cylindrical roller bearing with both rings tightly fitted on the shaft and into the housing, respectively.

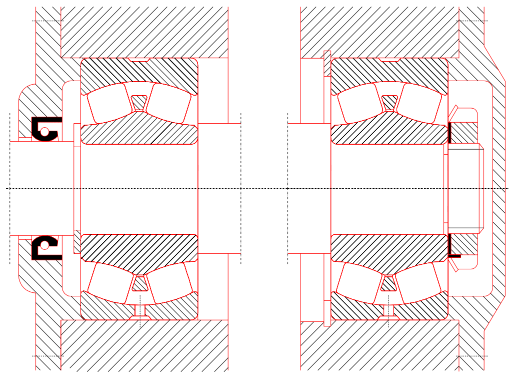
b) Both bearings are supported by spherical roller bearings. The locating bearing is tightly fitted both on the shaft and into the housing. The non-locating bearing has the outer ring mounted with clearance into the housing and thus allows axial displacement in both directions.

c) The locating bearing consists of a cylindrical roller bearing, NUP type and the non-locating bearing consists of a cylindrical roller bearing, NU type.

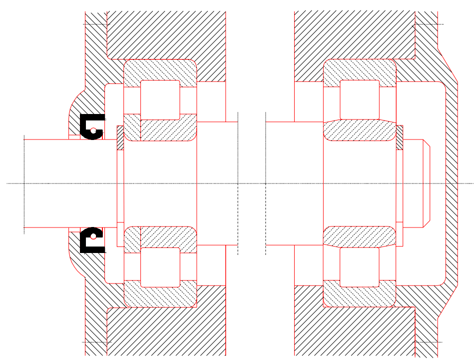
d) The locating bearings consists of a cylindrical roller bearing. NU type which takes over radial loads and of a four- point contact ball bearing (unloaded on the outside). The non-locating bearing consists of a cylindrical roller bearing, NU type.



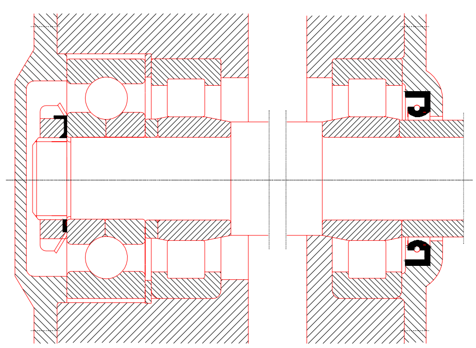
locating bearing   a   non-locating bearing



locating bearing   b   non-locating bearing



locating bearing   c   non-locating bearing



locating bearing   d   non-locating bearing

RECOMMEMDATION FOR BEARING FITS SELECTION

Selecting the bearing fits

Bearing fits are selected on the basis of the following criteria:
a) firm location and uniform support of bearings;
b) simply mounting and dismounting;
c) axial displacement of non-locating bearing.

According as the operation conditions, between the inner ring and the shaft, between the outer ring and the housing, respectively clearance fits, intermediate fits or interference fits may be performed.

When selecting the fit, one has to consider the difference of temperature which may occur between ring and shaft or between ring and housing. The tolerance classes are available for bearing fits which do not exceed +120°C during operation.

A high tightening is recommended for roller bearings and large size bearings in comparison to ball bearings of the same size. In case of a tight fit, the inner ring is supported by the entire shaft contact surface, thus bearing is used at full load carrying capacity.

When selecting a fit, the load of the rotating ring has to be considered and it will be avoided the excessive clearances or tightening.The excessive tightening couldn’t eliminate only the radial mounting clearance of the bearing itself but can determine even the destruction of the ring in the mounting process (because of the tensions stress that are resulting from the ring).

At their turn, the excessive clearances may determine the reduction of the ensemble rigidity and the occurrence of the contact corrosion phenomenon due to the excessive mobility of the joint.

Special fits for some types of bearings

In certain cases the selection of optimum fits for a bearing must be given up. Between these cases there is the design of mechanisms devices which during exploitation must be often dismounted to remove the bearings from their mounting places. For devices of cheap and low-importance machines for which it is not economical to process precisely and complexly the mounting places on the shaft, bearing tight fits will not be used. The same recommendation goes for shafts whose life is much longer than the one of mounted bearings and their frequent replacement would lead to an unavoidable wear of the spindles.

In all these cases it is rational to use larger clearance fits than the theoretical ones namely less tightening fits and to avoid the rings rotation from the mounting places are used pins, keys or other fixing devices.

Normal precision bearings are usually used and only in a few special cases will be used enlarged precision bearings, for example in bearings of main shafts of grinding machines or of high precision lathes and for bearings of high rotation speed shafts where it is required to limit the centrifugal forces resulted from the rotation of the non-balanced elements and in electric motors. In case of using high precision class bearings, the shafts and housings must have a rigid construction and their processing accuracy must be at the level of bearing precision. Form deviations of shafts and housing bores will be two or three classes more accurate than the deviations mentioned for normal precision bearings. Thus, for high precision bearings (P5 or higher), used for machine tools, the housing bore will have to have K6 dimensional deviations and IT3 form deviations.

In the case in which the bearing bore is tapered, the tolerance of the form deviation of the housing bore will be smaller, respective IT2. Often it is recommended the dynamic balancing of the shafts.

Since in the mounting process the bearing rings shape up according to the part form on which these are mounted, it is useless and non-economic to recommend high precision bearings be mounted on the shafts and housings with major form deviations.

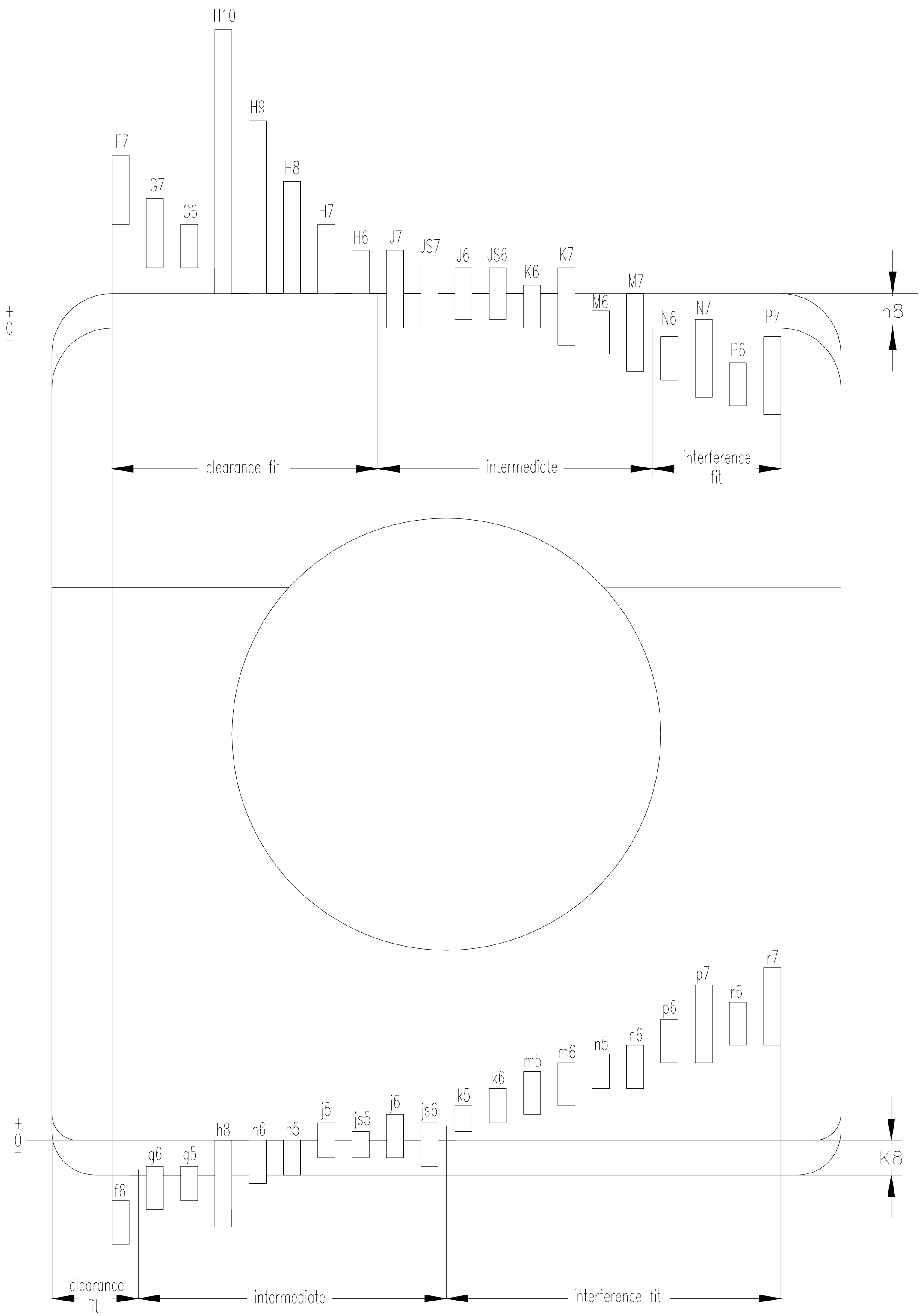
Table 2

Operating conditions	Examples	Tolerance class symbol	Remarks
RADIAL BEARINGS			
SOLID HOUSING			
Rotating outer ring load			
Heavy loads on bearings in twinwalled housings, heavy shock loads (P>0,12C)	Roller bearing wheel hubs, connecting rod bearing	P7	Outer ring cannot be displaced
Normal and heavy loads (P>0,06C)	Ball bearing wheel hubs, connecting rod bearings, crane traveling wheels	N7	
Light and variable loads (P≤0,06C)	Conveyer rollers, rope sheaves, belt tension pulleys	M7	
Direction of load indeterminate			
Heavy shock loads	Traction motors	M7	Outer ring cannot be displaced
Normal and heavy loads (P>0,06C). Outer ring displacement is not necessary	Electric motors, pumps, crankshafts main bearings	K7	
SPLIT OR SOLID HOUSINGS			
Direction of load indeterminate			
Light and normal loads. Desirable outer ring displacement. (P≤0,12C)	Medium sized electric motors, pumps, crankshafts main bearings	J7	Outer ring can be displaced
Stationary outer ring load			
Any type of loads	General mechanical railway axleboxes	H7	Outer ring can be easily displaced
Light and normal loads with simple conditions (P≤0,12C)	Gearing	H8	
Heat conduction thruogh shaft	Drying cylinders, large electric machines with spherical roller bearings	G7	
SPLIT HOUSINGS			
High accuracy rotation, quiet running			
High shiftness at variable loads	Main shafts for machine tools D≤125 with roller bearings D>125	M6 N6	Outer ring cannot be displaced
Light loads, indeterminate direction load	Shaft operating surface for grinding machines with ball bearings, free bearings for high speed superchargers	K6	Outer ring cannot be displaced
Desirable outer ring displacement	Shaft operating surface for grinding machines with ball bearings, free bearings for high speed superchargers	J6	Outer ring can be displaced
Quiet running	Small-sized electrical machines	H6	Outer ring can be easily displaced

Operating conditions	Tolerance class symbol	Remarks
THRUST BEARINGS		
Axial load		
Thrust ball bearings	H8	For less accurate bearing arrangements, clearance in housing can be up to 0,001D
Cylindrical and needle roller thrust bearings	H7 (H9)	
Combined loads on spherical roller thrust bearings		
Local load on housing washer	H7(H9)	
Peripheral load on housing washer	M7	
Axial or combined spherical roller thrust bearings		
Bearing radial location is ensured by another bearing	-	Housing washer fitted with clearance up to 0,001D

Table 4

Operating conditions	Examples	Shaft diameter [mm] Ball bearings	Cylindrical needle and tapered roller bearings	Spherical roller bearings	Tolerance class symbol
Radial bearings with cylindrical bore					
Stationary load on the inner ring					
Easy axial displacement of inner ring on shaft desirable	Wheels on non-rotating shafts (free wheels)	All diameters			g6 (f6)
Axial displacement of inner ring on shaft not necessary	Tension pullyes, sheaves				h6
Rotating inner ring load					
Light and variable loads (P<0,06C)	Conveyers, lightly loaded mechanisms bearings	18÷100 >100÷140	≤40 >40÷100		j6 k6
Normal and heavy loads (P>0,06C)	General mechanical engineering electric motors, turbines, pumps, gearboxes, woodworking machines	≤18 > 18÷100 >100÷140 >140÷200 >200÷280 - - -	- ≤40 >40÷100 >100÷140 >140÷200 >200÷400 - -	- ≤40 >40÷65 >65÷100 >100÷140 >140÷280 >280÷500 >500	j5 k5(k6) m5(m6) m6 n6 p6 r6 r7
Heavy loads and shock loads, ardous working conditions (P>0,12C)	Heavy duty railway vehicles axle bearings, traction motors, rolling mills	- - -	>50÷140 >140÷200 >200	>50÷100 >100÷200 >200	n6 p6 r6
High running accuracy, light loads (P<0,06C)	Machine tools	≤18 > 18÷100 >100÷200 -	- ≤40 >40÷140 >140÷200	- - - -	h5 j5 k5 m5
Axial loads					
	All kind of bearing application	≤250 >250	≤250 >250	<250 >250	j6 js6
Tapered bore bearings with withdrawal or adapter sleeve					
	Axle shaft for railway vehicles	All diameters			h9
	General mechanical engineering				h10
Thrust bearings					
Axial loads					
Thrust ball bearings		All sizes			h6
Cylindrical and needle roller thrust bearings		All sizes			h6(h8)
Cylindrical, needle roller and cage thrust assembly		All sizes			h8
Combined loads spherical roller thrust bearings					
Stationary load on shaft washer		≤250 >250			j6 js6
Rotating load on shaft washer or undetermined load direction		≤200 > 200÷400 >400			k6 m6 n8


H10
H9
F7
H8
G7
G6
H7
H6
J7
JS7
J6
JS6
K6
K7
M6
M7
N6
N7
P7
P6
h8
+
0
−
clearance fit
intermediate
interference fit
r7
p7
r6
p6
n5
n6
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clearance fit
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DEVIATIONS OF FORM AND POSITION

Permisible deviations of form and position for shaft and housing where bearing are to be mounted are given in table 1 and surfaces roughness of shaft and housing seating are given in table 2.

If bearings are mounted with adapter or withdrawal sleeves, shaft surface ruoughness should be of maximum R_a = 1,6 µm.

Basic tolerances for form and position deviations *Table 1*

Tolerance name	Fit	Symbol of deviation	Allowed deviations depending on the tolerance					
				P0 P6X	P6	P5	P4 (SP)	P2 (UP)
Tolerance of dimension	shaft		-	*IT6(IT5)*	*IT5*	*IT4*	*IT4*	*IT3*
	housing			*IT7(IT6)*	*IT6*	*IT5*	*IT4*	*IT4*
Tolerance of roundness and cylindricity	shaft	○ ⌭	t, t_1	*IT4/2 (IT3/2)*	*IT3/2 (IT2/2)*	*IT2/2*	*IT1/2*	*IT0/2*
	housing		t, t_1	*IT5/2 (IT4/2)*	*IT4/2 (IT2/2)*	*IT3/2*	*IT2/2*	*IT1/2*
Tolerance of face runout	shaft	↗	t_2	*IT4(IT3)*	*IT3(IT2)*	*IT2*	*IT1*	*IT0*
	housing			*IT5(IT4)*	*IT4(IT3)*	*IT3*	*IT2*	*IT1*
Tolerance of concentricity	shaft	◎	t_3	*IT5*	*IT4*	*IT4*	*IT3*	*IT3*
	housing			*IT6*	*IT5*	*IT5*	*IT4*	*IT3*
Tolerance of angularity	shaft	∠	t_4	*IT7/2*	*IT6/2*	*IT4/2*	*IT3/2*	*IT2/2*

Remarks: In case of bearings on which adapter or withdrawal sleeves are to be mounted, the shaft tolerances for deviations of form and position should be to IT5/2 tolerance class for shafts with diameter tolerance h9, and IT7/2 for shaft tolerance h10.

Since in the mounting process the bearing rings shape up according to the part form on which these are mounted, it is useless and non-economic to recommend high precision bearings be mounted on the shafts and housings with major form deviations.

The roughness of the mounting surfaces of the shaft and housing *Table 2*

Bearings. Tolerance class	Shaft. Diameter d, mm			Housing. Diameter D,mm		
	≤ 80	>80...500	> 500	≤ 80	> 80... 500	> 500
	Roughness R_a, [µm].					
P0, P6X and P6	0,8 (N6)	1,6 (N7)	3,2 (N8)	0,8 (N6)	1,6 (N7)	3,2 (N8)
P5, SP and P4	0,4 (N5)	0,8 (N6)	1,6 (N7)	0,8 (N6)	1,6 (N7)	1,6 (N7)
P2 and UP	0,2 (N4)	0,4 (N5)	0,8 (N6)	0,4 (N5)	0,8 (N6)	0,8 (N6)

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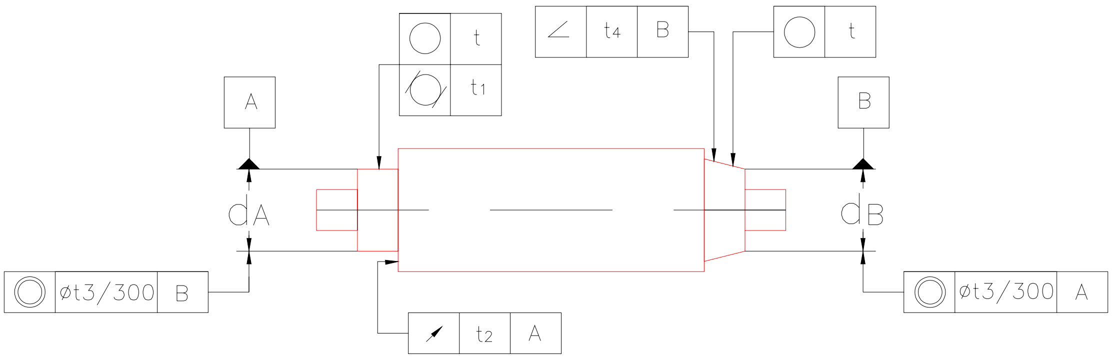

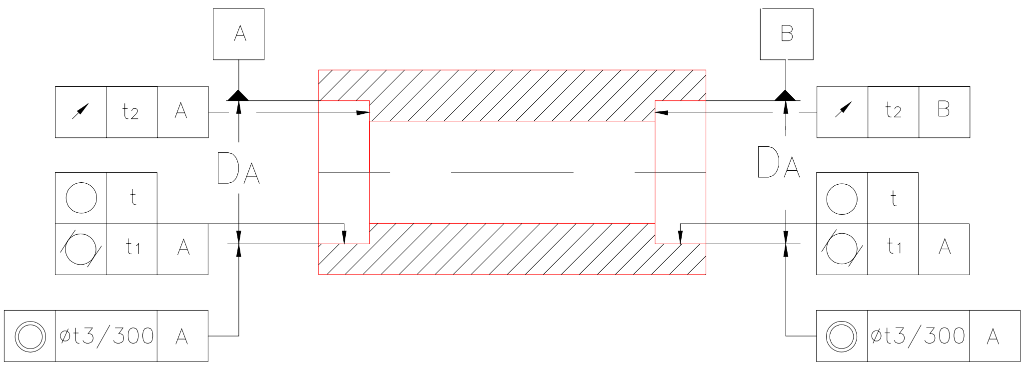

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BEARING AXIAL LOCATION

In order to achieve an axial location of bearings there is a great number of solutions according to the bearing type and magnitude of load to be taken over. Figure 1 presents schematic examples of axial location for fixed bearings and figure 2 presents the same for free bearings..

In cases in which no axial load is transmitted through a certain bearing, a ring could be used of an interference fit only.

The most common axial location system is performed by means of cover, nuts and screw plates etc. through axial supporting of the bearing rings.

For light axial loads, axial location systems could be achieved by means of safety rings.

Due to a low height of the safety rings and of exterior connecting radius of the bearings rings, sometimes intermediate rings are required, mounted between the bearing and the safety ring.

When a low interference fit is used for the inner ring in order to avoid its rotation against the shaft, a lock washer will be introduced between the bearing ring and the lock nut.

The lock washer will have a tenon that entering in a groove in the shaft will disconnect the transmission of the friction forces to the nut thus eliminating the danger of breaking the lock nuts.

Another axial location system is achieved through the taper pressed assemblies, using adapter or withdrawal sleeves.

This system is possible only for tapered bore roller bearings and has the following advantages:

- - heavy axial loads could be taken over in both directions;
- - it is not necessary a high manufacturing precision for the shaft;
- - ensures an easy mounting and dismounting.

It is however necessary to support the inner ring of the bearing on the shaft collar or on a snap ring which in case of using the withdrawal sleeve prevents the axial displacement for heavy axial loads and in the case of adapter sleeve it facilitates bearing dismounting.

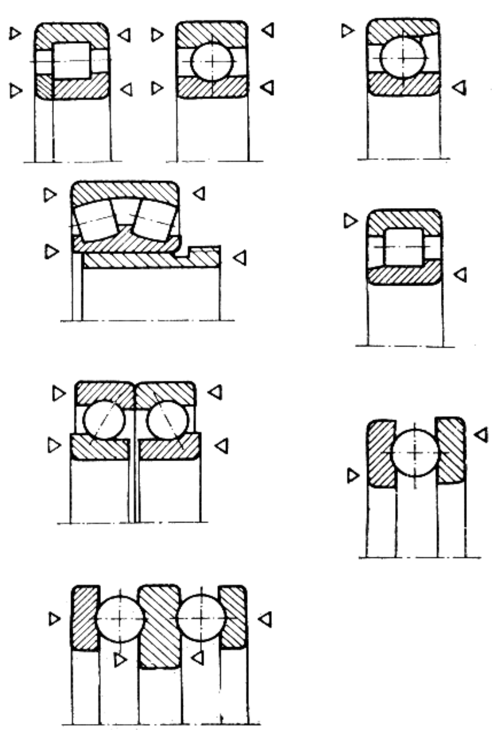


Figure 1

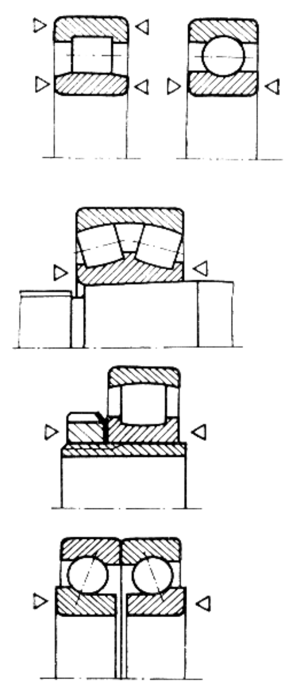


Figure 2

BEARING SEALING

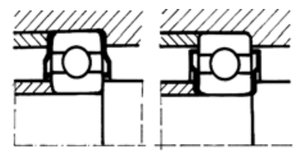
The correct sealing systems ensure a normal operation life to bearings and even the operation of the entire mechanism by their protection against penetration into the bearing of injurious elements (dust, material particles, humidity, acids etc.) and by keeping the lubricant inside the bearing.

In the assemblies with faulty sealing devices or devoid of sealing devices, different foreign bodies penetrate inside the bearing thus inducing an abrasive internal wear of the bearings or the corrosion of their active surfaces.

The leakage of the lubricant from the bearing during its operation determines a useless consumption of lubricants and if the lubricant leak is not discovered in time, it may result a fast heating and/or bearing damage.

The selection of the sealing system depends on the following factors, that feature the assembly operating conditions:

a) bearing rotation speed;
b) type of employed lubricant;
c) lubricating system;
d) assembly operating temperature;
e) environmental conditions;
f) constructive peculiarities of bearing assembly



a *b*
Figure .1

From a functional and constructive viewpoint, the sealing systems are as follows:

- *Stationary seals*, between the stationary elements (housing and cover);
- *Rotary seals*, between the rotating bearing elements;
- *Non-rubbing seals;*
- *Rubbing seals.*

a *b*
Figure . 2

According to the working conditions, the environment in which the lubricated bearing operates, combined (special) sealing system can be used with both types of sealing at the same time. In their description we assume that the inner bearing ring rotates and the outer ring one is fixed.

Stationary seals

The most simple sealings used in bearings are spring washers (shields). This type of sealing is used for peripheral speeds up to 6 m/s in clean and dry environments and their efficiency depends on the clearance between the washer and the shaft in rotation, the housing, respectively or on the radial clearance of the bearing. For this reason, the check washers can be used for bearings with minimal radial clearance. When spring washer are to be mounted, must be ensured their perfect contact on all circumferences of contact surfaces with the shaft (or housing) and bearing.

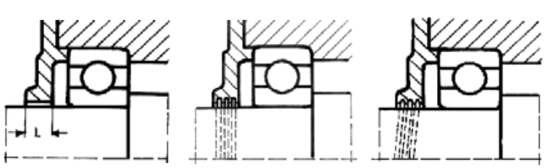
Figure 1 a illustrates the sealing with stationary spring washers (stationary shields) which are used for consistent greases, figure 1b illustrates the sealing with rotary spring washers (rotary shields) used for liquid lubricants ushed onto bearings by centrifugal forces. In the same time the shields laterally shake the dust particles and other impurities from them.

Figure 2 presents two examples of bearings sealed with sealing washers type 2RS (2RSR) - figure 2a, or shielded bearings with shields 2Z (2ZR) - fig 2b, which are delivered together with the entire required grease content for their life and can be successfully operated.

Rotary seals

a) Non-rubbing seals

This type of sealing is used for high temperature and speeds assemblies, their life being considered almost unlimited. There are the following types of rotary non-rubbing seals: gap seals, labyrinths and their combinations.



a *b* *c*
Figure 3.

b) Gap rotary non-rubbing seals

They are used for assemblies less exposed to the danger of impurities and humidity penetration in the working space of the bearing.

In simple cases there are enough the gap seals presented in figure 3a, which are mainly designed to retain grease into the bearing housing. The efficiency of sealing is conditioned by the gap length and by the clearance between shaft (or housing) and the sealing element. Sealing can be improved if on the shaft or in the housing there are one or more circular grooves which are to be filled with grease (see figure 3b), so the leak of grease is reduced and is stoped the penetration of impurities.

In case of oil lubrication, the grooves on the shaft must be helical (figure 3c), and their direction must be the same with the direction of the shaft rotary movement. The allowable peripheral speed for this type of sealing is up to 5 m/s. During operation, the sealing grooves are maintained filled with clean, consistent and good quality grease.

c) Labyrinth seals

They are used at high peripheral speeds for the bearings that are functioning in environments with impurities, for example in dust or when the housing is water sprinkled. These are presented in figure 4.

By increasing the number of gaps the efficiency grows very much. The gaps inside the labyrinths must be filled with a grease based on calcium or lithium soap to ensure the anticorrosive protection in the presence of water. In difficult cases it is recommended from time to time (2...3 times a week) to press in the gaps fresh grease to replace the dirty one or to complete the leakage. The labyrinths can be axially (a), radially (c), oriented or may have sloped steps (b). Sealing efficiency increases where both radial and axial labyrinths are used and the number of gaps is increased.

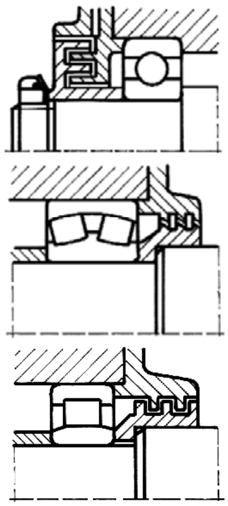


Figure 4

Other types of seals may be executed according to figure 5 through joining iron sheet blades (figure 5a) or processing the labyrinth walls with arc of circle form to avoid grease expulsion from the gaps in the housing in which the shaft is tilted (figure 5b).

Also in order to avoid this phenomenon (which is frequent in the case of high speeds), the radial runout of the surfaces that make the gap must be minimum and their roughness must be Ra > 1.25 μm.

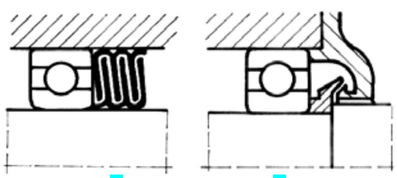


Figure 5

Rubbing seals

When selecting the proper rotary rubbing seal the following factors have to be considered: material and its elasticity (felt, rubber, plastic materials, leather, graphite, asbestos, metals etc.), resistance at different temperatures, maximum peripheral speed on sealing surface, sealing direction etc.

The contact surfaces with ribbing seals must have a very small roughness especially when the peripheral speeds of the contact surfaces are very high.. It must be also assured a mounting that will not damage the sealing elements.

Figures 6a and 6b illustrate sealing with single or double felt rings frequently used for grease or mineral oil lubrication; these are constructively simple, cheap and recommended mainly for peripheral speeds between 4 and 7 m/s (provided the contact surfaces are bright finished) and temperatures over +100^{o}C, efficiently protect the bearing and prevent the leakage of the lubricant.

Before mounting, felt ring is impregnated during an hour with a mixture of mineral oil (66%) and paraffin (33%) at a temperature of +70..+90^{o}C, so that sealings are improved as the friction is reduced. The efficiency of felt sealing depends mainly on the felt quality (with long fibers). The use of low quality felt leads not only to its frequent replacement but can damage the bearing.

Figures 6c and 6e, illustrates rubbing seals with a spring incorporated, which are made of synthetic rubber or special plastic materials and have, in most cases, a metallic hardening fixture.

To maintain as long as possible a tightening force of the seal lip, around the felt ring is mounted a spiral spring. To press the seal in the housing it is recommended to use the press and to avoid the seal lip deterioration.

Rubbing seals with a spring incorporated are most suitable to be used in case of oil lubricated bearings which are operated under peripheral speeds of 5..10 m/s and temperatures between -40^{0}C..+120^{0}C.

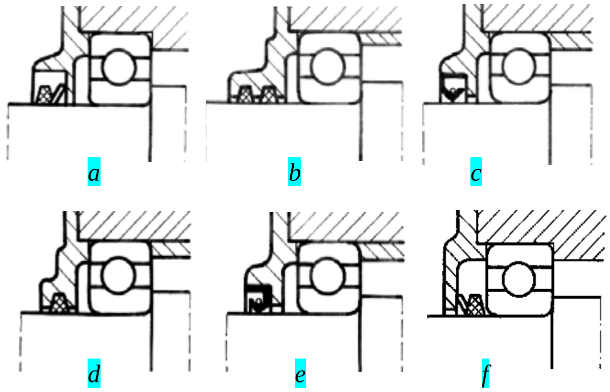


Figure .6

Lubricant outflow can be stopped by mounting the rubbing seal with incorporated spring with the edge inwards (figure 6c) or outwards (figure 6e) if sealing has to prevent dust or other impurities from penetrating into the bearing.

Double sealing with these rubbing seals can also be used, improving thus the reliability and durability of the sealing.

In figure.6e, and f, we are presenting V-rings sealing (figure6f) for grease lubrication and 3.6e for oil lubrication). V-ring seals are used at temperatures of -40^{o}C..+100^{o}C, roughness of sealing surface being Ra=1,5 - 3 μm.

V-ring seals can also be used on the shaft subject to eccentric or angular (2 ...3)^o movements.

For peripheral speeds up to 15 m/s the V-ring seal operates as a rubbing seal and for peripheral speeds over 15 m/s the seal lip will lift from the sealing surface operating as a centrifugal sealing.

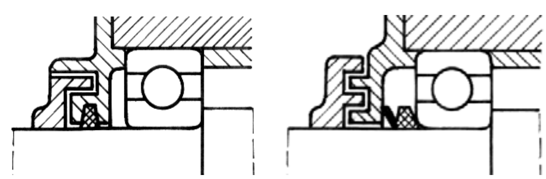
To obtain superior sealing results, the above mentioned sealing are to be combined, example in figure 7a and 7b, or other combined sealing, thus ensuring good sealing for any operating conditions of the bearings assemblies.

The sealing systems for vertical shafts are not generally different from the systems used for horizontal shafts except lower bearings of the vertical shafts that require a special device to prevent lubricant leakage.

The presented sealing systems does not represents all the constructives posibilities.

The dimensions indicated for grooves and clearances at the labyrinth sealing (which have a great importance for the normal operation of the bearing) are recommended for the normal functioning temperatures.

In case of significant temperature variations and for high axial loads acting on the bearing, these dimensions must be increased.



a. Felt ring rubbing seal, combined with axial labyrinth non-rubbing seal.

b. V-rubbing seal, combined with axial labyrinth non-rubbing seal.

Figure.7

GREASE LUBRIFICATION

Selection of consistent lubricants

Except the direct use of the vegetal or animal fats as lubricants in very restrained applications, grease represent the usually consistent lubricant because of the simple housing construction, simplicity of the sealing and the relubrication facility.

The grease are obtained as dispersed mixtures with plastic properties of some thickening agents as dispersed phase (soaps of the fatty acids of Na, Ca, Li, Pb etc or paraffin, bentonite, silicagel etc) in mineral or synthetic oils or in oleaginous fluids as dispersion environment (75 - 90 %). In grease composition may be introduced additives for high loads, corrosion avoiding or thermal stability.

Lubricant quantity and relubrication intervals

The amount of grease to be packed into the bearing at the beginning depends on the rotation speed. The bearing cavities should be filled with grease to capacity to ensure all the functional areas of the bearing are supplied with grease. The housing space should be filled as fallows:

- full at $n/n_{lim} < 0,2$;
- one third at $n/n_{lim} = 0,2 .. 0,8$;
- not at all at $n/n_{lim} > 0,8$ where n represents the working rotation speed and n_{lim} represents the limit rotation speed recommended for greasing in the catalogue.

The grease quantity required for the initial lubrication at the normal bearings can be calculated with the following relations:

$G = d^{2,5} / 900$ [g], for d in mm, at ball bearings;
$G = d^{2,5} / 350$ [g], for d in mm, at roller bearings,
where: d is bore diameter of bearing.

At high rotation speeds of the shaft it is recommendable that befhours introducing grease in bearing this must be immersed in refined mineral oil of mean viscosity, then, after oil release, to be introduced the grease.

In bearings with $n/n_{lim} = 0,2 .. 0,8$ with horizontal axis it is indicated to fill only the lower half of the housing and the cover to remain empty. At vertical axis mountings in which the housing is made of two pieces is recommendable that both pieces are inserted into the grease but only to half of the free space.

For low rotation speed bearings ($n/n_{lim} < 0,2$) that are operating in moisture and dusted environment, the free space of the housing must be fully filled to obtain a satisfactory sealing.

Normally at the beginning of operation of a recently greased bearings, it may notice a temperature increase over the normal working temperature after which the temperature decreases and remains constant at a value of 10 - 50°C over the environment temperature. The persistence of high values of temperature may be due to an excess of grease within the bearing, a too heavy load on the bearing or to misalignement of shafts.

Table1

Bearing type	Value of coefficient α	
	Relubrication interval	Grease life
Spherical roller bearing	1	2
Tapered roller bearing	1	2
Thrust ball bearing	1	2
Cylindrical roller bearing	5	15
Needle roller bearing	5	15
Deep groove ball bearing	10	20-40*

* Reduced values for bearings fitted with seals or shields.

Table 2

Temperature ^{0}C	70^0	85^0	100^0
f_1	1	0,5	0,25

Table 3

Operating conditions	Light	Moderate	Heavy	Very heavy
f_2 (dust), f_3 (vibrations)	1	0,7-0,9	0,4-0,7	0,1-0,4

The relubrication intervals are established by practical observations and adequate recommendations or with experimentally established formulas such as:

$$Tu = \alpha\left(\frac{14 \cdot 10^6}{n\sqrt{d}} - 4d\right) f_1 f_2 f_3, \text{ where:}$$

Tu - relubrication interval or the grease life, in operation hours;
α - coefficient depending on the bearing type (see table 4.8);
n - bearing rotation speed, rpm;
d - bore diameter of the bearing, in mm;
f_1, f_2, f_3 - coefficients depending on the operating and environmental conditions (see tables 2 and 3)

The values resulting from calculus without f_1, f_2, f_3 correction coefficients are valid for working temperatures up to 70°C.

For temperatures above this value the relubrication interval decreases to a half, than the preceding period, for every 15°C without exceeding the temperature limit for every type of grease. The relubrication interval is also reduced when there is not a good environmental condition for grease or in hard operating conditions. Where the grease has a sealing role against water penetration, (where the housing is frequently washed as for the paper processing machines) it is necessary to complete weekly the amount of grease.

For the replenishment, the required quantity is given by the orientative formula:

$$Tc = K_c DB \text{ [g]},$$

where:

D - outer diameter, in mm;
B - roller bearing width, in mm;
K_c - coefficient (table 4)

Replenishment must be done with the same type of grease or compatible ones to avoid the incompatible mixtures.

Table 4

Replenishment interval	K_c
Daily	0,0012-0,0015
Weekly	0,0015-0,002
Monthly	0,002-0,003
Yearly	0,003-0,0045
2-3 years	0,0045-0,055

Consistent lubricants feeding systems

In an increasing no. of applications it is recommended the use of the greasing in bearings with protection devices (shields) or sealing devices (seals).

In conditions of a controlled technological process, the greasing enhances the behavior and durability of bearings. The grease is filled into the cavities in a ratio of 30 - 50%, according to the minimum quantity required for greasing, but also to the heat produced in the bearing and its evacuation in the exterior. Usually in this conditions the greasing is done once for the entire life of the bearing, named "life-time" lubrication.

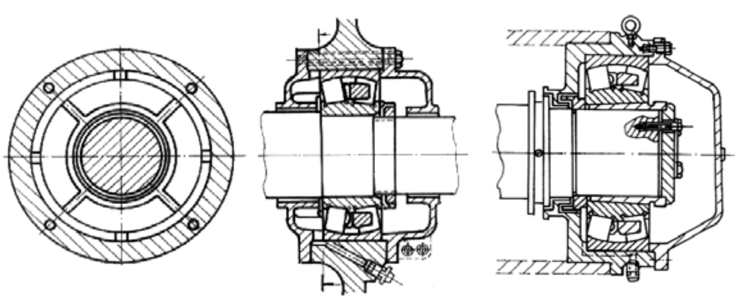


Figure 1

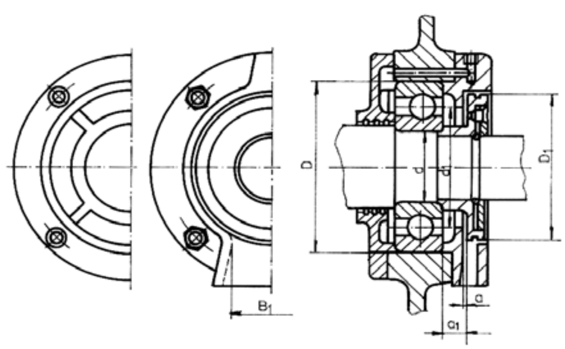


Figure 2

For the rest of greased bearings constructions the best solution is given by the normal application of grease in the housing hollows (figure 1) in the above mentioned ratio. If the relubrication intervals are at least of 6 months, replenishment ducts or devices are not executed. Ribs may be executed on covers to reduce the grease rotation trend together with the shaft or bearing, or adequately dimensioned elements (adjusting washers) to prevent the filling of too much grease within the bearing (figure 2)

The grease replenishment must be done with grease cups, greasing pump ("grease gun") to insert grease under pressure or grease pump to introduce grease in the centralized greasing systems.

OIL LUBRIFICATION

a) Selection of liquid lubricants

Table 1

Bearings operating temperature, (^{0}C)	Viscosity at 50^{0}C	
	Centistokes, (cSt)	Engler degrees, (^{0}E)
30	7	1,52
40	8,5	1,70
50	12	2,06
60	17	2,65
70	25	3,65
80	35	4,96
90	50	6,96
100	70	9,67
>100	150...200	20...30

The sealing difficulties and lubricant leakage limit the applications of liquid lubrication to the applications where the stability and control of the lubricating film, reduced friction and cooling effects (heat evacuation through lubricant is determined by load, rotation speed and operating temperature) are important. Where the lubrication of bearings can be achieved in a common oil bath, liquid lubrication is recommended.

Liquid lubricants for the bearings are both mineral and synthetic oils and mixtures of mineral and synthetic oils. The oils can be added with solid lubricants (molibdenum disulphide, graphite, teflon) or with other substances whose action manifests positively in improving the resistance of the lubricating film under high load, in avoiding the oxidation, the froth or viscosity decreasing at the increase of the temperature, etc.

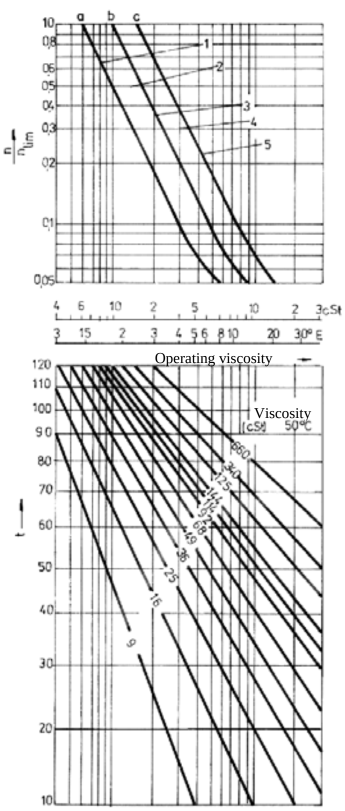


Figure 1

The oil viscosity is highly influenced by the temperature, decreasing at the increase of the temperature. The dependence of viscosity on temperature is evaluated by the viscosity index (IV), a higher value (IV>100) indicates a better stability of viscosity related to temperature. Table 1 presents the recommendations for the choice of oil viscosity at 50°C according to the operating temperature of the bearings.

The viscosity is also influenced by pressure which is important mainly in the case of bearings, and for that reason in the case of special applications with heavier loaded bearings it is necessary to choose the lubricant after a thorough analysis.

The chart in figure 1 may be used to determinate the viscosity at 50°C (in cSt). The upper diagram serves for determining the oil viscosity according to the bearing type, to the type of the load, and to the ratio n/n_{lim}, where n represents the working rotation speed of the bearing and n_{lim} represents the limit rotation speed for oil lubrication (values according to the catalogue). As a general rule, the viscosity must be greater for the axially loaded bearings than for the radially loaded ones.

The lower diagram is used to establish the oil viscosity (in cSt at 50°C) according to the operating temperature, t [°C], and viscosity.

b) Liquid lubricants circulating systems

For the usual lubrication of bearings very low quantities of lubricant are necessary to reach the lubricating film. Considering the fact that the oil is required to act as coolant, it must be taken into account this requirement when determining the lubricant quantity. Heat evacuation from the bearing through the lubricant is more necessary when load and rotation speed increases.

The oil circulating systems, regarding the way in which bearing lubrication is assured are achieved through the following: oil bath lubrication; oil bath with external oil circulation; jet lubrication; sprinkling by conveyance ring; oil driving by thrower ring; oil dropping; oil mist.

The selection of a lubrication system may be also done according to the recommendations given in table 2 that take into account the general working conditions and the product $d_m \bullet n$

The oil circulating systems, regarding the way in which bearing lubrication is assured are achieved through the following: oil bath lubrication; oil bath with external oil circulation; jet lubrication; sprinkling by conveyance ring; oil driving by thrower ring; oil dropping; oil mist.

The selection of a lubrication system may be also done according to the recommendations given in table 2 that take into account the general working conditions and the product d_m•n

Table 2

Circulating systems	Working conditions. Hints.	d_m•n	Oil viscosity at 50^{0}C (m^2/s)	Example in figure
Oil bath	The bearing is filled in oil to the middle of the lower rolling element for horizontal shafts and to 70 - 80% of the roller bearing width for the vertical shafts. Used for road vehicles, machine tools and railway vehicles. Magnetic plugs are recommended for catching the metallic particles. Attention to sealing and level check.	< 200000	(12,5..180)x10^{-6}	2, 3
Oil bath with external circulation	The oil is delivered with a pressure of 0,15 MPa from a central tank fitted with a cooling, heating and filtering and flow regulating system. The diameter of the evacuation pipe will be 2 .. 10 times greater than the one of the feeding pipe.	< 600000	(30..120)x10^{-6}	4
Oil jet	High loads and speeds where cooling is required. The diameter of the evacuation pipe is greater than those of the nozzle. The flow is set to 0,5 .. 10 l/min according to the temperature. Used for machine tools, axial compressors, centrifugal separators.	< 900000	(15..50)x10^{-6}	5
Sprinkling (splashing)	Medium loads and rotation speeds. Used for vehicles, gear boxes. Magnetic plugs are recommended for catching the metallic particles.	< 175000	(20..90)x10^{-6}	6
Oil driving	High speeds. The oil thrower ring must allow also oil mist formation.	< 180000	30x10^{-6}	7
Oil drop	Medium loads and relatively high rotation speeds. Used for machine tools. The lubricant flow must be set at 0,5 .. 6 drops per minute.	< 210000	30x10^{-6}	8
Oil mist	Medium and small bearings with high loads and rotation speeds. The required flow of oil mist is of (0,001 .. 5) cm^3/hr. The pressure is of (0,05 - 0,5 MPa) and the air flow 0,5 - 4 m^3/h.	< 1200000	(16,5..45)x10^{-6}	9

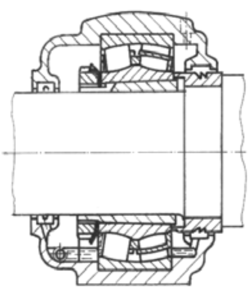


Figure 2

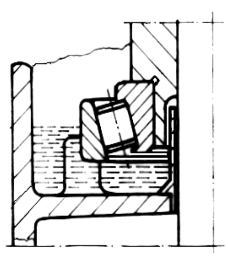


Figure 3

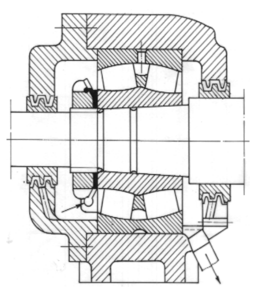


Figure 4

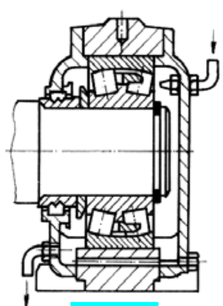
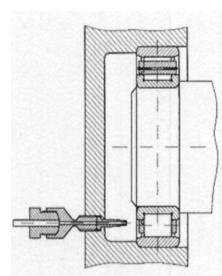


Figure 5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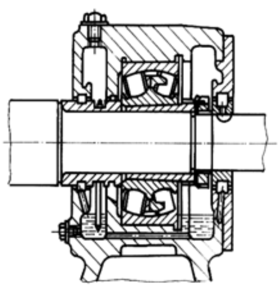
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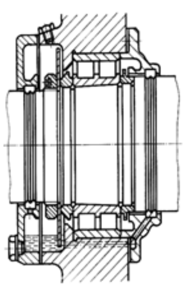
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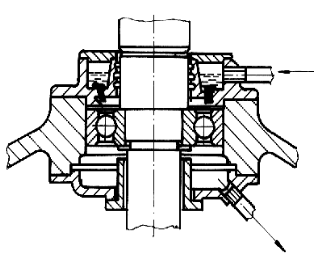
Figure 9

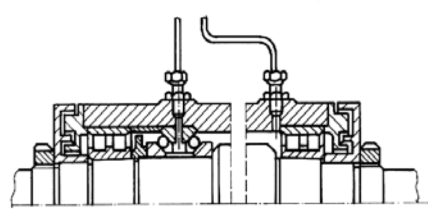
Figure 10

BEARING DESIGNASION

The purpose of designation is that of identification of bearings, so that of bearings with the same designation to be interchangeable both dimensionally and operationally no matter who the producers may be. Designations of URB rolling bearings are in accordance with those used by world-known bearing companies: SKF, INA, KOYO.etc. and they are standardized by national standard STAS 1679.

The complete designation of a bearing consists of a basic design and may include one or more supplementary designations (prefixes and suffixes), as shown in chart fig.1.

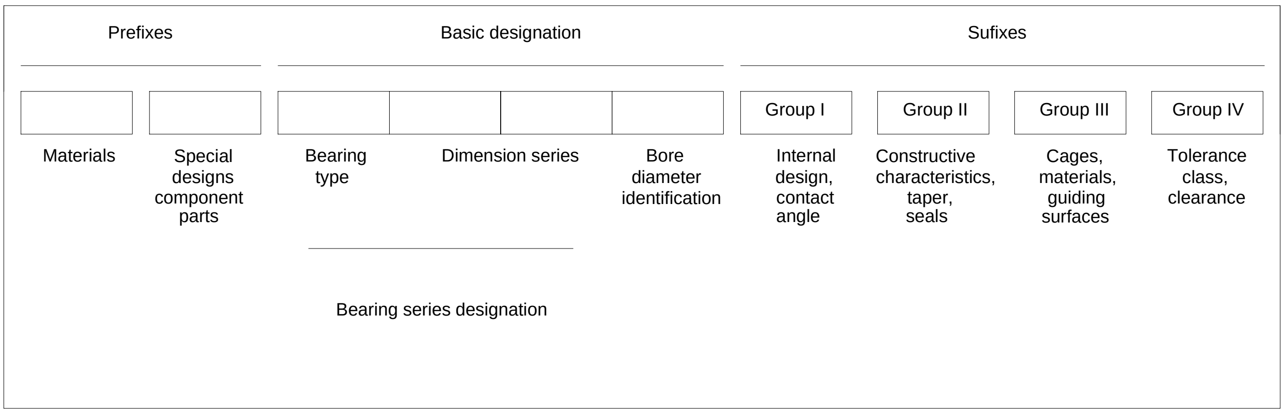

Fig. 1

The basic designation consists of an identification of the type of bearing (figure or letter), the series designation, in accordance with ISO and the bore diameter identification.

The designations of the bearing type and dimension series for main standardized and unstandardized bearing types are given in table 1.

Bore diameter identification consists of one, two or more figures as follows:

- bore diameter from 1 to 9 mm:	- one figure, representing the bore diameter (e.g. 623, 608),
- bore diameter from 10 to 495 mm	- two figures, as follows 00 for 10 mm, 01 for 12 mm, 02 for 15 mm, 03 for 17 mm, 04 and up to 99 for bore diameter from 20 to 495 mm, (bore diameter = bore diameter identification x 5, e.g. 6230, d = 150 mm);

Certain types of special bearings listed in this catalogue(e.g. support rollers) made an exception from this rule.
In this cases, the values of bore diameter are stated (e.g. NUTR25).

Prefixes

Prefixes are letter-identifications which indicate the material, other than steel for bearings or component parts of bearing. The prefix for material is separated by a horizontal line from the rest of designation.

Prefixes for materials

H - heat-resisting steel (e.g. H - NUP 210),
M - copper alloy (e.g. M - 6008),

S - plastics, glass, ceramics etc.(e.g. S - 6204),
T - case - hardening steel (e.g. T - 35352),
X - stainless steel (e.g. X - 6202).

Prefixes for special designs or parts of bearings

K - cage with rolling elements of dismountable bearing (e.g. KNU205),
L - free ring of dismountable bearing(e.g.LNU205) (interchangeable ring, e.g. L 30205),
R - dismountable bearing without free ring (ex. RNU205; RN205; RNU5208).
E - shaft washer of thrust ball bearing (e.g. E 51210),
W - housing washer of thrust ball bearing (e.g. W 51216),
WS - shaft washer of roller thrust bearing (e.g.. WS 81108),
GS - housing washer of roller thrust bearing (e.g. GS 81112),

LS - axial washer, thickness greater than 1 mm (e.g.. LS 2035).
AS - axial washer, thickness less than 1 mm or less (e.g. AS 2035)

Suffixes

Suffixes are used to identify various constructive modifications of the bearing in comparison to normal design. They are classified in four different groups, as follows:

Group I - Modifications of internal design, design with increased basic load (e.g. A, C, E etc.), contact angle (e.g. A, B, C) and others.

Group II - Modifications of external design, tapered bore, groove on outer ring etc. (e.g.21318CK, NUP311ENR, 6304-2RSR),

Group III - Modifications of cage design, material, guiding surfaces etc. (e.g. 6304TN, NU410 MA),

Group IV - Modifications of normal design regarding tolerance classes, bearing radial or axial clearance, stability of dimensions at high temperatures, bearing matching etc.(e.g. 6404P5, 6404P53, NU210SO).

These suffixes for bearing designation are listed considering the groups they belong to, at the beginning of each bearing group.

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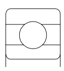
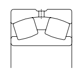
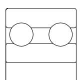
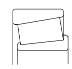
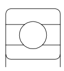
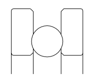
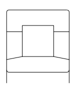
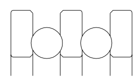
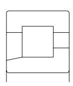
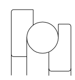
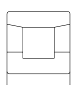
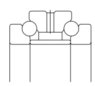
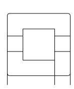
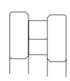
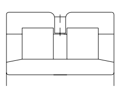
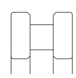
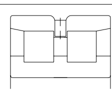
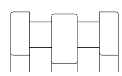
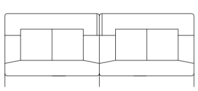
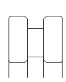
Designation of type and dimension series for the main standardized and non-standardized bearings

Table 1

Bearing design	Bearing type identification	Series design. Standardized	Nestandardized	Example	Bearing design	Bearing type identification	Series design. Standardized	Nestandardized	Example
	6	17, 00, 04 18, 10, 22 19, 02, 23 29, 03	50...69 65305	61952 6208		2	39, 41, 23 30, 22, 13 40, 32 31	51...59	22216 25130
	4	22 23		4204 4305		3	29, 02, 13 20, 22, 23 30, 32 31, 03	40...49	32010 32208 34115
	E	E B0 L0 M		E15 L20		5	11, 13 12, 14	51...59	51115 51212 55144
	NU	19 29 10 02 22 03 23 04 22 03 23 04	51...59	NU208 NU5140		5	22 23 24	61...69	32010 52308 56120
	NJ			NJ2206 NJ5140		R_Y	65	66...69	R_Y 6540 R_Y 6681
	N			N310 N5161M		23	44 47		23420 234720
	NUP			NUP209 NUP5410		8	11 12	51...59	81115 81220 85115
	NNU	49	51...57	NNU4920 NN5124		8	93	51...59	89312 85312
	NN	30	51...57	NN3015		8	22 23 24	61...69	82210 82315 86144
	NNU	69 60		NNU6064 4NNU5146		ANK			ANK 2035

MOUNTING AND DISMOUNTING OF BEARINGS

MOUNTING OF BEARINGS

The preparation of components for the mounting procedure

Before mounting, all components must be verified according to the prescriptions at the dimensional, form precision and surface quality.

The composite elements that are specific to a bearing assembly are: shafts, housings, axial fixing elements (adapter sleeves, shafts and housings, covers) and sealing elements.

The preparation of new bearings

Bearings in their original packing are protected against corrosion and it is not necessary to remove the preserving substance (oil or grease) from them.

Removing of the original packing will be done in the same day when the mounting of bearings is performed.

The preparation of used bearings

Bearings with damaged packing and bearings preserved more than 12 months will be washed and preserved again.

For cleaning, the bearing will be introduced in a lamp oil bath, clean diesel fuel or white-spirit to remove the old preserving substance and the other impurities.

During cleaning the rings must be rotated so that all surfaces are cleaned. After cleaning, the bearing must be immersed in another bath with white-spirit to rinse it and after that it will be dried suspended. Before rinsing, the aspect of the exterior surfaces of the bearing must be verified. If there are noticeable corrosion points with a surface less than 5 mm^2 on the externl surfaces of the outer ring and on the frontal surfaces of both rings, it is allowed to remove them by friction with sandpaper.

It is strictly forbidden to effect this operation on the rolling sides of the bearing (race ways and rollers)!

For the anticorrosive protection of relubricated bearings, the best method is to insert them in a technical Vaseline (petroleum jelly) bath at 50-60°C or consistent grease heated at 70-90°C. After the complete heating of bearings (recognized by the uniform layer of grease on them), they must be removed from the bath, dried and after cooling, they will be packed in plastic sheet or waxed paper and will be stored in cardboard boxes (for small and medium size bearings) or wrapped in textile band (large size bearings).

The insertion and removal of bearings from the bath must be made with metallic hooks without touching the bearings bare-handed.

The grease or the technical Vaseline could be replaced with bearings preservation products.

To prevent the impurity contamination of greases bearings, the following directions must be respected:

- the bowl in which the grease is preserved must be closed with a lid;
- do not take out the grease with dirty hands or instruments.

It is recommended the use of a metallic shovel which can be easily cleaned.

In case the above mentioned directions are not respected, the premature damage of the bearings is facilitated.

Shafts preparation for bearings mounting

There are verified the mounting surfaces, including the frontal part of the shafts ribs which must be clean, smooth, without traces of impact, scores, corrosion etc.

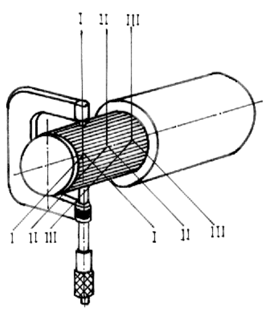


Figure.1

The presence of abrasive particles or metal chips makes the mounting procedure extremely difficult.

If in spite of such shaft damage, a bearing is mounted on it, its position may be incorrect determining a decrease of its durability. If bearing is fixed in an axial location through a lock nut, it is necessary to screw up one time the lock nut to remove the bits that otherwise could reach the inner side of the bearing.

Dimensional precision check according to the execution drawings is done by fixing the axis between the bottoms or in rests with the aid of a limit gauge or with the micrometer. It is necessary to check the shaft diameter with the micrometer in several points situated on the length of the shaft (figure 1). This allow the observation of the processing uniformity on the entire length and points out the tapering of the shaft. The ovality of the shaft must also be checked by means of a micrometer or limit gauge in a few planes and at least in three direction at 120° for every plan. To verify shape deviations in case of long dimension spindles a straight line with ink on it is used. The line is placed on the shaft in longitudinal direction and after that is moved back and forth. If the shaft is straight, on its surface is formed a continuous band of ink and if the spindle has surface deviations the ink band will be interrupted.

To verify the small length taper shaft, a ring gauge covered with ink which allows shape deviation verification through the look of the ink deposits on the shaft surface is used.

At diameters greater than 140 mm the ring gauges are replaced by special measuring devices.

The under-dimensioned shafts **must not be admitted** in the mounting process because in this case the rotation of the inner ring is unsuitable. This phenomenon is accompanied by a fast increase of the bearing temperature determining its rapid destruction. Because of strong heating, the bearing will get dark and the surface of the shaft will be destroyed. This procedure must not be applied because the securing of the interior ring of the bearing against axial displacements does not exclude the danger of ring rotation on the shaft.

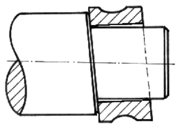


Figure.2

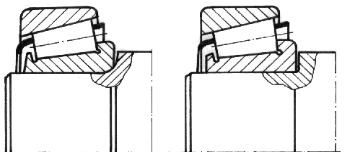


Figure.3

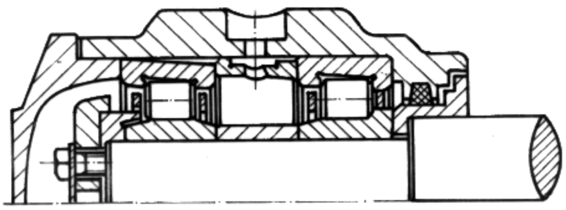


Figure.4

Neither over-dimensioned shafts must be admitted in mounting because of the dilatation of the inner ring leading to the disappearance of the radial clearance required in exploitation. If the clearance between the rings is reduced, the rollers or the balls will be blocked. Such a bearing will have a slow motion, it will be very heated and will soon be destroyed.

Thus, it is important to choose correctly and to respect the tolerance fields for shafts and the tolerances for form and position deviations.

For a normal functioning in bearings, especially in case of high axial efforts and high rotation speeds it is very important that the shaft ribs and connections be correctly designed and executed with maximum precision.

The frontal part of ribs must not disclearance traces of impact and damage because the ring of the bearing must adhere uniformly to the entire frontal surface. The shaft rib must be normal on the spindle axis.

The check of rib perpendicular alignment normality may be done with the gauge through shaft securing between its two sides.

If the ribs are not perpendicular alignment on the surfaces and shaft axis, these are determining additional stresses on the bearing even in the absence of the external loads.

The processing conditions required to the ribs are important for the bearings when the cylindrical roller is axially loaded or when the exterior ring must be pressed onto the housing.

In figure 2 it is illustrated an exaggerated case of deformation of the cylindrical roller bearing inner ring because of the incorrect processing of the rib shaft.

The shafts with tapered ribs will be rejected from mounting because in this case only the exterior or the interior side of the bearing can touch the frontal surface of the rib after the bearing mounting as illustrated in figure 3 and 4 Under high load, the rib will rapidly deform itself and the bearing will not be tight between the lock nut and the shaft rib.

When moved inside the workshops, the shafts will be protected with sleeves. The shafts prepared for utilization and from several reasons are to be kept stored for more time must be protected with a technical Vaseline or grease layer and then will be packed in waterproof paper.

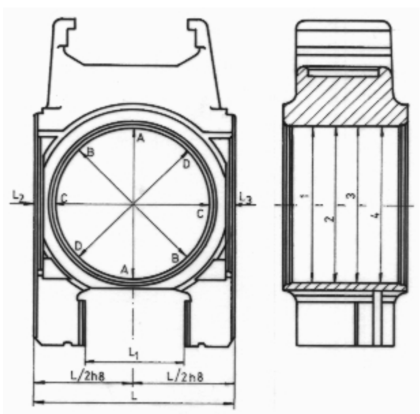


Figure.5

The preparation of housing

Generally, the above mentioned hints for shaft preparation are valid for the housings and for any other pieces which are part of a bearing assembly.

The inside part of the housings is measured with fixed gauges or interior micrometers. The interior surfaces will be examined to be smooth, clean, without impurities, burrs without cracks, scores etc. The outer ring of the bearing mounted in its seat must be uniformly in contact on all its circumference which is possible only when the boring is precisely processed.

In case of two-piece housings it will be verified the finishing degree of the contact surfaces of the two pieces to secure a good adherence eliminating any clearance between them.

The scaling check of the two contact surfaces will be done with the thickness gauge which must not cross nowhere along the contact surface of the two halves.

Form (cylindricity, tapered) deviation check will be done according to figure 5 through measurements effected in several planes and positions (1, 2, 3, 4) in at least 3 places on the circumference (diameters AA, BB, CC, DD), by turning the micrometer with 120°.

Alignment and parallelism of the half housings against their side surfaces can be verified through measurement of the thicknesses L1 and L2.

The correct geometrical form of the housings (even the one of the two pieces housings) is checked with gauges of adequate dimensions. These ones after their painting on the cylindrical surface are pressed onto the surface of the boring and rotated several times in both ways.

The surface of the housing bore is considered to be good if the paint tracks cover at least 75% of the surface.

The preparation for the mounting of the axial fixing elements

Usually, the axial mounting of the inner rings is done through supporting sleeves and covers or axial lock nuts, mounted on the frontal surfaces of the shafts.

At the axial fixing elements there are not accepted the following surface defects:

- local wears
- cracks and pinches
- sticking marks
- inclusions

The supporting sleeves are to be verified with the aid of the micrometer gauge.

The shape deviations of the axial fixing elements must be chosen in the same precision class with the one of the assembled bearings.

The measures taken both on the shafts, housings and fixing elements will be done in clean rooms at an mean temperature of 20°C and a relative humidity of 55%.

Before measuring the devices, the micrometer gauges and precision gauge devices will be introduced in the room waiting four hours to get them to the temperature inside.

The measuring devices and gauges will be carefully kept in their cases, in clean rooms, dust free and will be kept away from chokes and vibrations.

With the "go" and "not-go" gauge will be cecked the inner and the outer threads, the chamfers and grooves.

Bearing mounting devices

Generalities

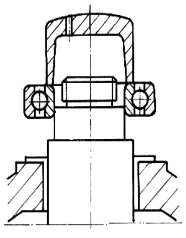


Figure.6

The operational safety of bearings depends highly on their correct mounting and dismounting on the operation site.

The non-observance of the requirements regarding the bearings mounting and dismounting in view of their maintenance often leads to important prejudices that may compromise assemblies perfectly designed from the technical point of view.

The diversity of bearings type dimensions and their operation conditions (load, rotation speed, temperature) require different methods for their mounting and dismounting with adequate tools and devices.

The basic idea in conceiving any bearings mounting device is that the exterior forces applied to the bearing when pressing must not be transmitted through the rolling elements. If this principle is not observed, because of the pressing forces, impressions could appear on the bearings raceways that will make it unusable after a certain running time.

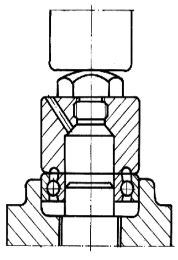


Figure.7

The devices must be as simple as possible constructively and secure a uniform and symmetrical disposition of the pressing forces on the contact surfaces. They must be comfortable and ensure a raised productivity in the mounting process.

Mounting of bearings with cylindrical bore

Mounting of bearings in housings and shafts fitted with interference fit is done by means of mechanical, thermal or hydraulic devices.

The pressing force must be transmitted only through the ring that causes with the shaft or the housing the interference fit avoiding its transmission through the rolling elements.

In small size bearings, mounting (bore diameter less than 50 mm) with intermediate fits or with interference fit in housing and/or shaft are used special sleeves with one or two ribs on which there are applied smooth strokes with a hammer. The use of sleeves guarantees the uniform distribution of force (figures 6, 7)

In order to ensure a progressive and continuous application of the pressing force, mechanical presses or hydraulic presses. If a non-separable bearing must be mounted in the same time on the shaft and in the housing, between the bearing and the mounting sleeve a plate is inserted to transmit uniformly the force on the frontal faces of the rings. If both bearing rings must form interference fittings, a sleeve whose front surface, in the shape of two circular rims, lies simultaneously both on the inner and the outer rings, is used (figure 7).

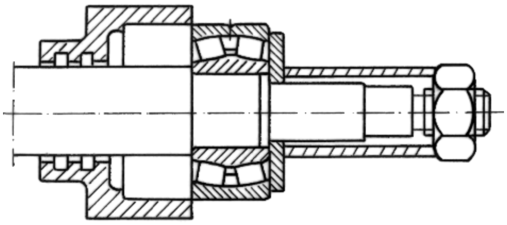


Figure.8

In the case of dismounting bearings, the rings could be mounted separately on the shaft, respective in the housing, which is favourable mainly when interference fits for both rings are practiced. The medium size bearings (with the bore diameter between 50 and 100 mm) and large size ones (with the bore diameter between 100 and 200 mm) cannot be cold pressed on the shaft or in the housing due to the increase of pressing forces with the bearing size increase. Thus, the non-separable bearings or the inner rings of the separable bearings are heated before mounting. If a non-separable bearing must be simultaneous mounted on the shaft and in housing, then it is inserted between bearing and sleeves, a suporting plate to evenly disperse the pressing force. (figure 8). The temperature difference, required between the bearing ring and the interlinked part, depends on the fitting and the size of the bearing.

Bearings **must not be heated over 110°C** in order not to cause alterations within the internal structure of steel thus, resulting in dimensional variations and decrease of their hardness.

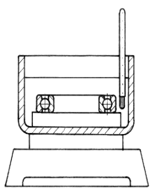


Figure.9

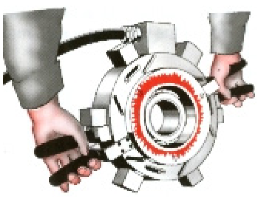


Figure.10

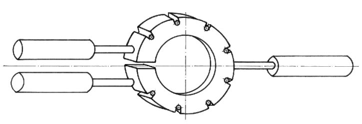


Figure.11

At the determination of the heating temperature must be taken into account the distance between the heating source and the mounting places since bearing cooling takes place during transportation. Local overheating must be avoided. The protected bearings or sealed bearings (2Z and 2RS) are not heated before mounting.

Oil baths, heating housings or electrical heating plate are used in bearings heating. In case of bath heating an anticorrosive oil is used, with low viscosity which easily leaks from the bearing when it is rotated upon removing from bath. An example of heating environment is the transformer oil.

As it can be seen in figure 9, the bowl is seated on an electrical heating plate. The oil temperature is checked with a thermometer, but it is preferred an automatic adjustment of temperature. The bath is fitted with a grill (situated at 60 - 70 cm from the bottom) to prevent the bearing from coming in direct contact with the heating plate and the possible impurities in the bath from entering the bearing at the same time. The oil must cover completely the bearing.

The bearings heated on the electrical plate must be turned several times for a uniform heating and to avoid local overheating. The heating time of the bearing is of 30 - 50 min, according to the bearing dimensions.

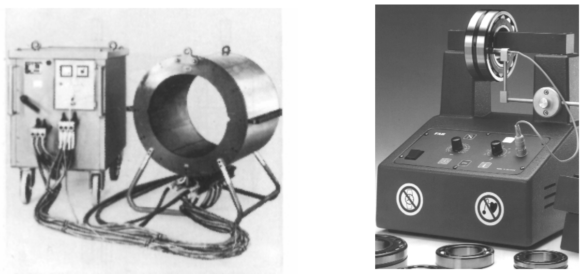
The temperature of the electric heating plate must be adjusted with a thermostat.

To obtain an enhanced productivity in case of a series production, other devices and heating equipment can be used (ex: the inner rings of the cylindrical bearings type NU, NJ and NUP can be heated before mounting with extraction electric devices or with a thermal dismounting ring as in figure 10, or with a dismounting thermal-ring figure 11).

The medium size and large size bearings could be mounted through the induction heating devices. These are, first of all, heated to a temperature greater than the room temperature by around 80°C.

The induction heating devices contain a coil inductor and a power group fitted with several voltage steps, time relays and thermal protection relays for ring heating. The separable inner ring of the bearing inserted into the inductor bore and kept there for a short time (about 80 sec) is heated through Foucault currents formation.

After heating, the demagnetized rings in the same device are removed and mounted on the shaft.



Induction heating devices

Mounting of cylindrical roller bearings

Mounting of the inner ring or of the outer ring into the assembly must not be done in force. If there is a considerable resistance, the ring with the rollers must be alternatively rotated during mounting (figure 12). To facilitate the mounting and avoid the bearing damage, the use of specially constructed auxiliary rings is recommended (figure 13 and 14).

After the mounting of cylindrical bearing (type N or NU), the axial displacement of a ring against the other must be measured and checked.

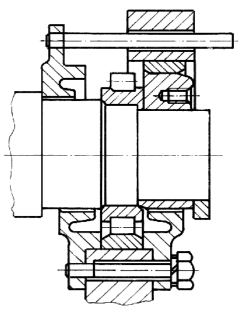


Figure.13

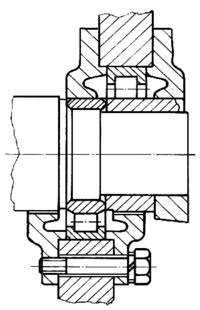


Figure.14

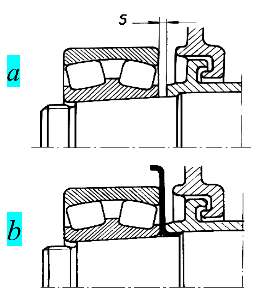
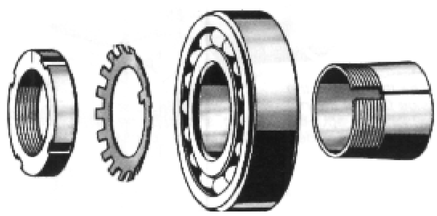


Figure.15



Adapter sleeve

Mounting of tapered bore bearings

The tapered bore bearings could be mounted directly on shafts, adapter sleevees or withdrawal sleevees. The mounting of these bearings is done only with interference fit.

The tightening is achieved through axial displacement of the inner ring of the tapered bore mounted directly on the tapered shaft or through axial displacement of the adapter or withdrawal sleeve. The tightening is evaluated through the reduction of the radial clearance or through the axial displacement. The radial clearance is measured with thickness gauge.

The axial displacement of the

mounted tapered bore bearing is measured with a limit caliber as presented in figure 15 a and b whose width is established with the following relation:

$$m = s - a,$$

where:

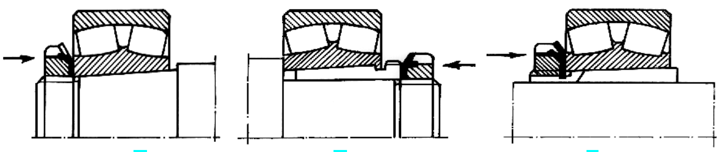
m - caliber width, in mm;

s - the initially measured distance in mm;

a - axial displacement from the table 6.1.

In table 1 are given for the self aligning spherical roller bearings the values of the radial clearance reduction after mounting according to the initial clearance.

The small size tapered bore bearings mounted directly on the shaft, adapter sleeve or withdrawal sleeve after fixing on the tapered shaft are axially displaced through some lock nuts type



a *b* *c*

Figure.16

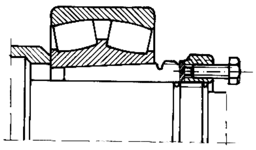


Figure.17

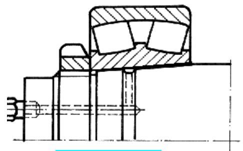
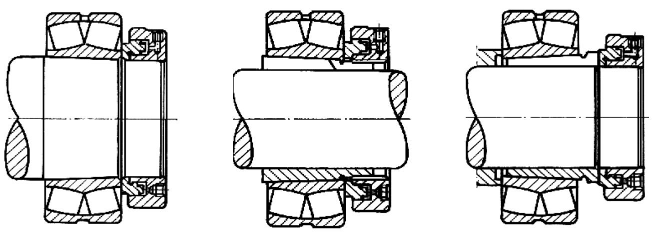


Figure.18



a *b* *c*

Figure.19

KM and washers type MB, (figure 16 a, b, c). The medium size bearings are axially displaced using a special nut (figure 17) with several screws which is afterwards dismounted and replaced with an axial lock nut.

In medium and large size bearings special mounting hydraulic presses (figure 18) or special hydraulic nuts are used, for example in figure19 a. (The mounting of a tapered bore bearing directly on the tapered shaft), figure19 b (the mounting of a bearing with adapter sleeve), figure19 c (the mounting of a bearing with withdrawal sleeve).

Dimensions in mm

Table 1

Inner diameter, d		Reduction of radial clearance		Axial displacement a, taper 1:12 On tapered shaft		Axial displacement a, taper 1:12 On tapered sleeve		Axial displacement a, taper 1:30 On tapered shaft		Axial displacement a, taper 1:30 On tapered sleeve		Minimum radial clearance after mounting, for clearance group		
over	up to	min	max	min	max	min	max	min	max	min	max	Normal	C3	C4
30	40	0,02	0,025	0,35	0,4	0,35	0,45	-	-	-	-	0,015	0,025	0,04
40	50	0,025	0,03	0,4	0,45	0,45	0,5	-	-	-	-	0,02	0,03	0,05
50	65	0,03	0,04	0,45	0,6	0,5	0,7	-	-	-	-	0,025	0,035	0,065
65	90	0,04	0,05	0,6	0,75	0,7	0,85	-	-	-	-	0,025	0,04	0,07
90	100	0,045	0,06	0,7	0,9	0,75	1	1,7	2,2	1,8	2,4	0,035	0,05	0,08
100	120	0,05	0,07	0,7	1,1	0,8	1,2	1,9	2,7	2	2,8	0,05	0,065	0,1
120	140	0,065	0,09	1,1	1,4	1,2	1,5	2,7	3,5	2,8	3,6	0,055	0,08	0,11
140	160	0,075	0,1	1,2	1,6	1,3	1,7	3	4	3,1	4,2	0,055	0,09	0,13
160	180	0,08	0,11	1,3	1,7	1,4	1,9	3,2	4,2	3,3	4,6	0,06	0,1	0,15
180	200	0,09	0,13	1,4	2	1,5	2,2	3,5	4,5	3,6	5	0,07	0,1	0,16
200	225	0,1	0,14	1,6	2,2	1,7	2,4	4	5,5	4,2	5,7	0,08	0,12	0,18
225	250	0,11	0,15	1,7	2,4	1,8	2,6	4,2	6	4,6	6,2	0,09	0,13	0,2
250	280	0,12	0,17	1,9	2,6	2	2,9	4,7	6,7	4,8	6,9	0,1	0,14	0,22
280	315	0,13	0,19	2	3	2,2	3,2	5	7,5	5,2	7,7	0,11	0,15	0,24

315	355	0,15	0,21	2,4	3,4	2,6	3,6	6	8,2	6,2	8,4	0,12	0,17	0,26
355	400	0,17	0,23	2,6	3,6	2,9	3,9	6,5	9	6,8	9,2	0,13	0,19	0,29
400	450	0,2	0,26	3,1	4,1	3,4	4,4	7,7	10	8	10,2	0,13	0,2	0,31
450	500	0,21	0,28	3,3	4,4	3,6	4,8	8,2	11	8,4	11,2	0,16	0,23	0,35
500	560	0,24	0,32	3,7	5	4,1	5,4	9,2	12,5	9,6	12,8	0,17	0,25	0,36
560	600	0,26	0,35	4	5,4	4,4	5,9	10	13,5	10,4	14	0,2	0,29	0,41
630	710	0,3	0,4	4,6	6,2	5,1	6,8	11,5	15,5	12	16	0,21	0,31	0,45
710	800	0,34	0,45	5,3	7	5,8	7,6	13,3	17,5	13,6	18	0,23	0,35	0,51
800	900	0,37	0,5	5,7	7,8	6,3	8,5	14,3	19,5	14,8	20	0,27	0,39	0,57
900	1000	0,41	0,55	6,3	8,5	7	9,4	15,8	21	16,4	22	0,3	0,43	0,64
1000	1200	0,45	0,6	6,8	9	7,6	10,2	17	23	18	24	0,32	0,48	0,7
1200	1250	0,49	0,65	7,4	9,8	8,3	11	18,5	25	19,6	26	0,34	0,54	0,77

Performance test

Test starting must be done after the following verifications have been carried out:

a) tightening screws and nuts;

b) checking the radial clearance of the bearing;

c) hand rotation test of bearings (if the assembly allow this test).

Attention! These two verifications (b and c) must also be performed before bearing mounting, so:

b) Check the radial clearance of bearings using thickness gauges which are inserted:

- between the rollers and the raceway of the outer ring at the upper part for the spherical bearings;
- between the rollers and raceway of the inner ring at the lower part for bearings type NU, NJ;
- between the rollers and the raceway of the outer ring at the lower part for type N bearings.

Note: In case of deep groove ball bearings the clearance cannot be checked after mounting and before mounting, this operation requires the existence of a special device.

c) At the hand rotation test, it must be checked if bearings are easily rotating without interruptions, with low and uniform noise. Before the rotation test, a few drops of transformer oil could be inserted in the bearing to avoid dry friction. If the hand rotation is normal, the sealing is further checked, the required quantity of lubricant is inserted and after that the operation test is performed.

During the latter, the assembly must be verified mainly regarding the noise and easy rotation.

The noise check during operation should be carried out by a specialized worker who is supposed to listen to the bearing noise, a worker capable to make difference between the noise produced by the bearing and the noise of the other moving parts of the machine.

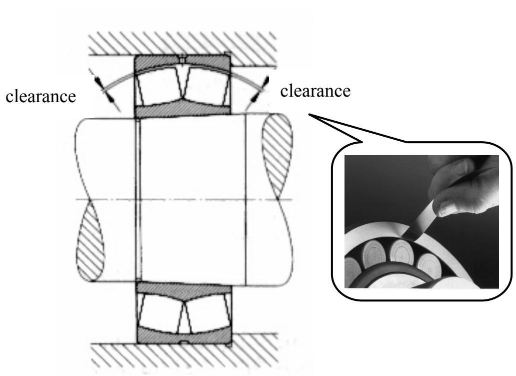
This check can be done with an acoustic tube, long hilted screwdriver.

A correct mounted bearing must operate uniformly, without shockes and unusual noise. A dump interrupted noise shows that the bearing is dirty and a “whistling” sound indicates that the bearing is not enough lubricated or there are frictions between the bearing and its connection pieces.

In both cases it is necessary to stop the test to verify the bearing mounting and its state.

During the performance test the bearing temperature must be frequently verified. Under normal conditions this must not exceed with more than 20 - 30°C the temperature of the environment.

There are not allowed temperatures over 80°C.



Clearance checking

In the first moments of the test, because of the friction between the sealing and the shaft, a sudden increase of the temperature above normal values may occur, but in a certain period of operation this increase is stabilized.

If the increase of the temperature has a pronounced and continuous character, the test must be interrupted to find the reasons for which the bearing heated.

In case in that at the assembly dismounting there are no faulty bearing or adjacent parts and the aspect of the pieces is normal, the correct selection of the bearing for the operating conditions must be checked.

We recommend ensuring a minimal loading (Fr min) in bearing assembly test functioning, mainly in case of high rotation speeds such as:

$Fr_{min} = 0{,}01Cr$ - for radial ball bearings;

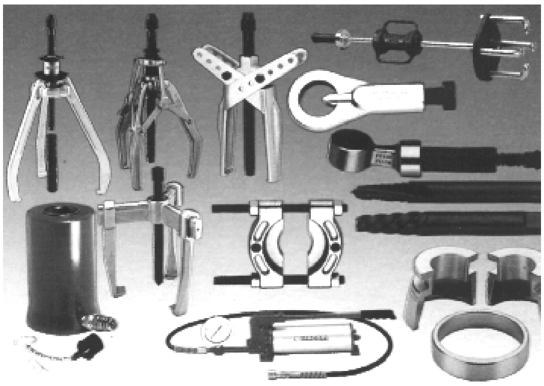
$Fr_{min} = 0{,}02Cr$ - for radial roller bearings;

$Fr_{min} = 0,04Cr$ - for full complement bearings,
where: C_r = basic dynamic load taken from the bearing catalogue.

This is necessary in diminishing the effect of inertial forces which cause sliding effects between the raceway and elements with a negative influence upon the operating conditions (additional friction, temperature increase).

The performance test with the loaded bearings as in the operating conditions will allow to notice the possible faults and to remove them thus ensuring a good running of the assembly.

DISMOUNTING OF BEARINGS



Dismounting kit

Wrong *Right*

Rules in bearings dismounting

Bearings dismounting implies the same measures described in the above section (Bearing Mounting).

Many users do not pay special attention to this operation, they don't know and don't apply dismounting methods.

The reasons that determine the necessity of roller bearings dismounting are different, such as:

a) faulty mounting;
b) regular wear of bearings;
c) complete damage;
d) device repair and general repair of bearings;

Thus, bearing dismounting must be done carefully during its taking out even from the shaft and from the housing. The same attention must be paid to non-operable bearing, whose inspection is useful to determine the causes of damage.

If the bearing is completely damaged, precaution measures at its taking out are pointless, in this case the dismounting method must exclude the possibility of damaging the surrounding parts or affecting the exploitation qualities of the machine in running, device etc.

The problem of ensuring cleanliness conditions of the dismounted bearings must be considered differently according to their situation.

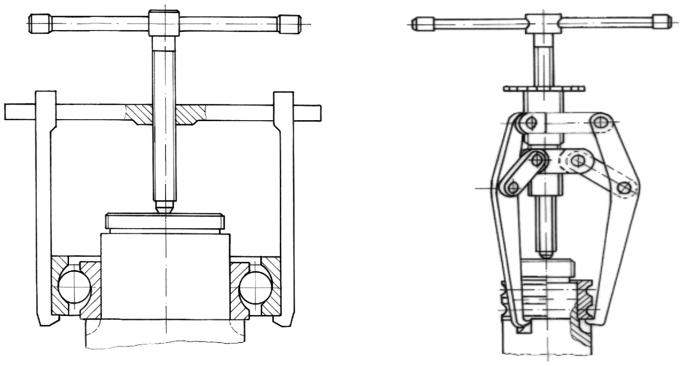


Figure 1

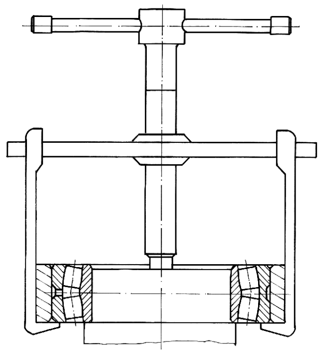


Figure 2

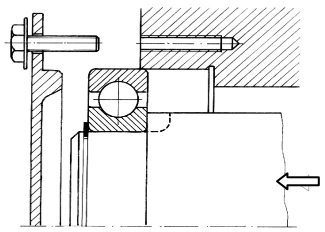


Figure 3

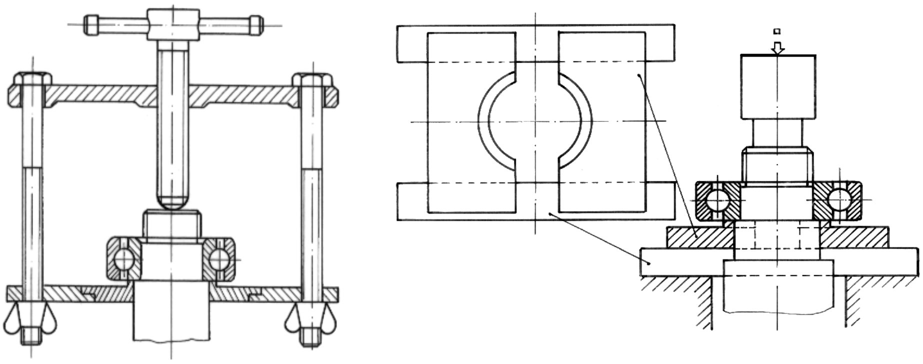


Figure .4

When dismounting a demanged bearing, it is not necessary to observe the maintenance - washing rules, if the user dismounts good bearings or bearings for inspection cleanliness is essential.

Before bearing dismounting, it is recommended to take all the necessary measures to ensure cleanliness at work site. In bearing dismounting, the worker wear gloves and use clean tools.

If the bearing functioned and there are reasons to suppose that impurities penetrated inside it, it is necessary that immediately after dismounting the bearing must be washed by the methods presented.

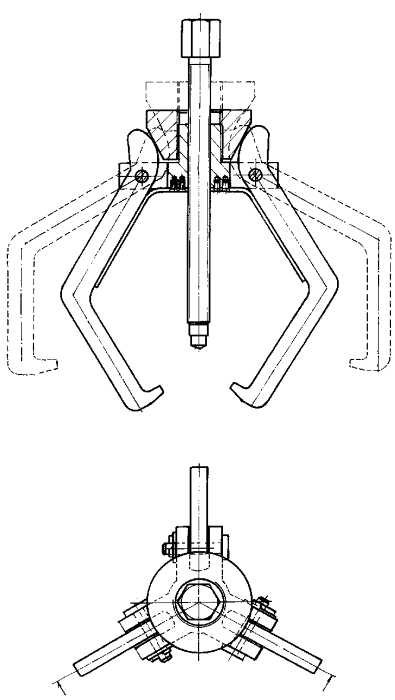


Figure .5

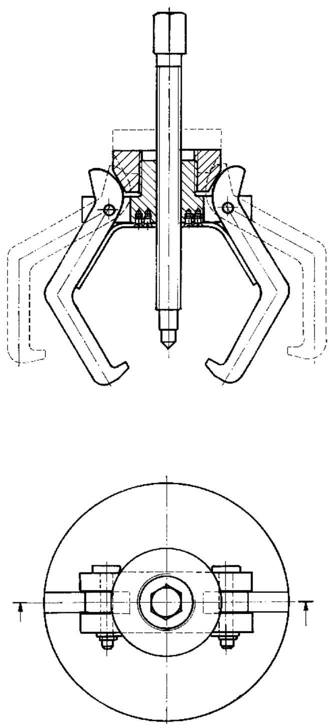


Figure .6

Bearing dismounting devices

Bearings dismounting from housings and shafts is done with specific means according to the fitting type, bearing type and size, necessity of further using the respective bearing, housing or shafts.

There could be distinguished two main categories during the dismounting operations:

1. Cylindrical bore bearing dismounting;
2. Taperred bore bearing dismounting.

The dismounting of bearings from housings and shafts with interference fit is carried out by mechanical, thermal or hydraulic means in reversed order than the mounting operations, dismounting first the elements less tightened.

The extraction force must be transmitted only through the ring in contact with the shaft or the housing of the interference fit, avoiding its transmission through the rolling elements.

Dismounting of bearings with cylindrical bore

For the small dimension bearing (bore smaller than 50 mm) from the separable or non-separable bearings category, tightening mounted, the extraction is done using a soft steel or copper mandrel with rounded edges and a hammer through which are applied strokes on the inner ring on its side face with the aid of the mandrel along the circumference of the ring. It must have care not to touch the shaft or the inner of the bearing, the strokes being able to determine irreparable damages.

If it is not required the further use of the roller bearing, its dismounting may be done using a mechanical press, figure 1 and figure 2, or the specific construction of the shaft, figure 3. The best extraction mode is presented in figure 4 when it is used a small capacity mechanical or hydraulic press, the shaft with the bearing being able to be moved in its working zone.

The elements used for dismounting are made on dimensional ranges.

If the shafts with the bearings cannot be moved, there are used two or three arms mechanical presses (figure 5 and 6). These are adjustable tools allowing a great range of dimensions.

In the case of medium and large size bearings it is recommendable the use of the oil injection method, figure 7, or 8, which develops great extraction forces. Through this method it is followed the creation of a oil film between the shaft and the inner ring facilitating thus the slide. In the case of dismounting after a long operation period in improper lubrication conditions which allowed oxidation between the contact surfaces, to prevent their deterioration it is recommended the use of a oil with antioxidant additives.

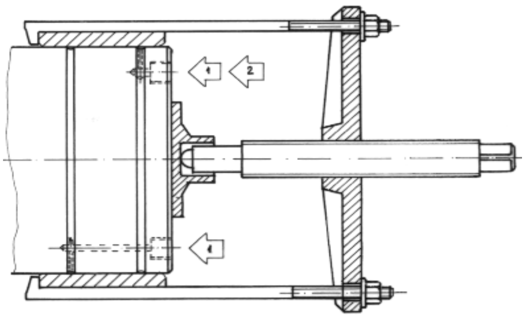


Figure .7

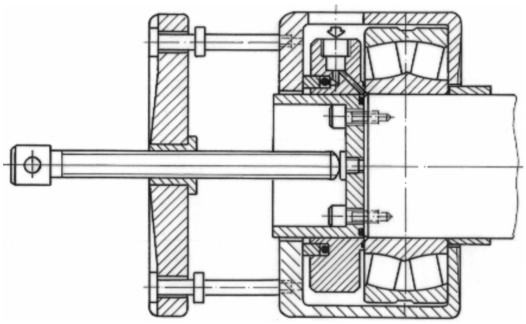


Figure .8

Medium and large size bearings and rings serial dismounting from the shafts is greatly facilitated using the thermal means.

The bearing or the separable inner ring is heated up to 80 - 100°C so that through dilatation it must be easily removed from the shaft.

There are several heating modalities depending on the repeatability of the dismounting operation and to the dimensions of the dismounting element.

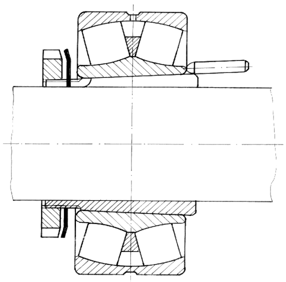


Figure .9

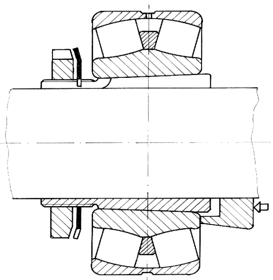


Figure .10

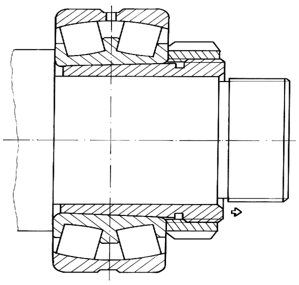


Figure .11

Thus, the heating may be done:

a) in mineral oil baths having the temperature of 80 - 100°C so that there are heated only the ring or the bearing protecting the rest of the shaft with asbestos or cardboard. The extraction is

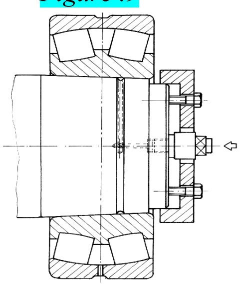


Figure 12

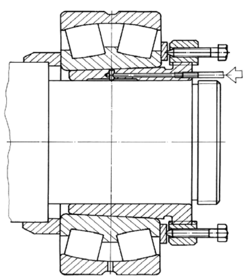


Figure.13

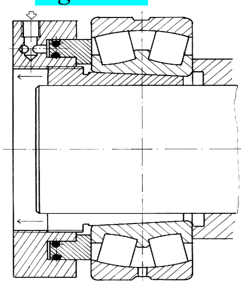


Figure.14

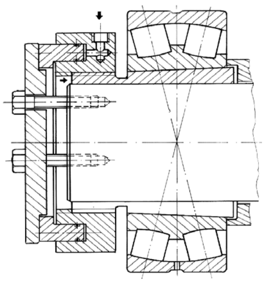


Figure.15

done by mechanical devices.

b) for the rings with medium and large dimensions, it is simple to use an aluminum made extractor ring (see figure 11 heated on the electrical heating plate or through induction up to about 220°C after which the ring to be dismounted is inserted and tightened with the two joined rings.

After about 20 - 30 seconds the ring dilates enough to allow its easy extraction. To protect the race-ways, the ring is smeared with silicone oil (which is resistant to oxidation).

Dismounting of tappered bore bearings

The dismounting of the small tappered bore bearings mounted directly on the shaft is achieved through the application of the tool (mandrel, pipe, half ring) directly on the side face of the inner ring and after that with gently strokes dismounting is produced.

In case of small bearings mounted on sleeves there must be opened first the lock washer and after that unscrew few turns, the lock nut.

The tool, which can be a tool or a mandrel (figure 9) or an intermediary half ring (figure 10) is placed in position and then, with strokes uniformly distributed on the entire circumference, the bearing is dismounted.

The withdrawal sleeve could be also dismounted mechanically. To this purpose it can be used the nut axially fastened after removing the shaft’s nut and lock washer (figure 11) by tightening the nut with the special wrench till the bush extraction.

The greater are the dimensions of the roller bearings (boring greater than 100 mm) the greater must be the dismounting forces.

In this case it is used the method of oil injection between the monting surfaces or the hydraulic nuts.

Oil injection could be applied in the case of the bearings directly mounted on the shaft through the special provided grooves.

Oil injection procedure is also applied to the bearings mounted on the adapter sleeve or withdrawal sleeves (figure and figure 13) which have provided grooves for this purpose.

The decrease of the physical effort and dismounting time are achieved by the use of the hydraulic nuts which are used both for removing from the withdrawal sleeves (figure 14) and for removing from the adapter sleeve (figure 15).

Insertion of pressurized oil into the hydraulic nuts is done with the aid of the manual presses. The oil injection method may be combined with the use of the hydraulic nut in case of large size bearings or for blocking them on the shaft.