Information

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1. Introduction to Rolling Bearings

1.1 Construction

Most rolling bearings consist of an inner ring and an outer ring, rolling elements (either balls or rollers) and a retainer (cage). The cage separates the rolling elements at regular intervals holds them in place within the inner and outer raceways, and allows them to rotate freely. (fig 1.1-1.8)

Rolling elements come in two basic shapes: ball or rollers. Rollers come in four basic types: cylindrical, needle, tapered, and spherical.

Balls geometrically contact the raceway surfaces of the inner and outer rings at points, while the contact surface of rollers is a line contact.

Theoretically, rolling bearings are so constructed as to allow the rolling elements to rotate orbitally while also rotating on their own axes at the same time.

While the rolling elements and the bearing rings take any load applied to the bearings (at the contact point between the rolling elements and raceway surfaces), the case takes no direct load. It only serves to hold the rolling element at equal distances from each other and prevent them from falling out.

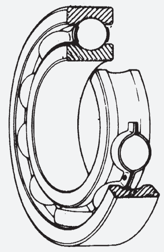
1.2 Classification

Rolling bearings fall into two main classifications: ball bearings and roller bearings. Ball bearings are Classified according to their bearing ring configurations: deep groove, angular contact and thrust types. Roller bearings on the other hand are classified according to the shape of the rollers: cylindrical, needle, taper and spherical.

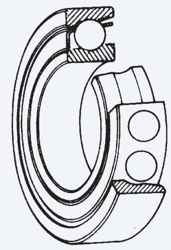
1.3 Characteristics

Rolling bearings come in many shapes and varieties, each with its own distinctive features.

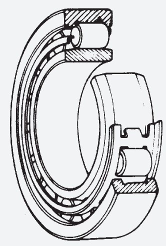
However, when compared with sliding bearings, in rolling bearings the starting friction coefficient is lower and only a litte difference between this and the dynamic friction coefficient is produced. They are internationally standardized, interchangeable and readily obtainable. Lubrication is easy and consumption is low.



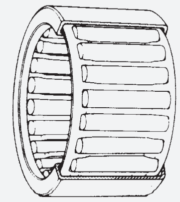
Deep groove Ball Bearing
Fig. 1.1



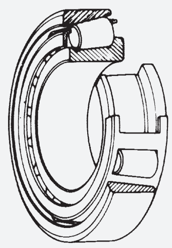
Angular contact Ball Bearing
Fig. 1.2



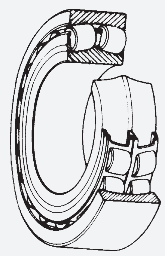
Cylindrical Roller Bearing
Fig. 1.3



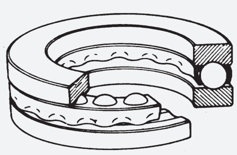
Needle Roller Bearing
Fig. 1.4



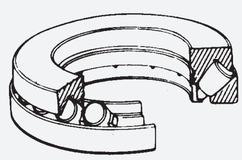
Tapered Roller Bearing
Fig. 1.5



Spherical Roller Bearing
Fig. 1.6



Thrust Ball Bearing
Fig. 1.7



Spherical Thrust Roller Bearings
Fig. 1.8

Ball bearings and roller bearings

When comparing ball and roller bearings of the same dimensions, ball bearings exhibit a lower frictional resistance and lower face run-out in rotation than roller bearings.

This makes them more suitable for use in applications which require high speed, high precision, low torque and low vibration. Conversely, roller bearings have a larger load carrying capacity which makes them more suitable for applications requiring long life and endurance for heavy loads and shock loads.

Radial and thrust bearings

Most of rolling bearings can carry both radial and axial loads at the same time.

Bearings with a contact angle of less than 45º have a much greater radial load capacity an classified as radial bearings. Bearings which have a contact angle over 45º have a greater axial load capacity and are classified as thrust bearings. There are also bearings classified as complex bearings which combine the loading characteristics of both radial and thrust bearings.

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2. Calculation of Service Life

To select an appropriate rolling contact bearing, it is necessary to know operating conditions, i.e., the magnitude and the direction of loads, the nature of loading applied, rotational speeds of one, or both, rings, the required service life, the working temperature of the bearing unit, and other requirements dependable on th structureal features of the machine in question.

The beaing service life, is understood to mean the time expressed in total number of revolutions made by one of the bearing rings relative to the other rings, before fatigue failure sets in at one of the rings or any other rolling elements. It can be expressed in million of revolutions or operating hours. The basic rated resource (i.e., the estimated service life) means the operating life of a batch of bearings wherein not less than 90% of identical bearings would operate without any indication of fatigue failure on their bearing surfaces under similar loads and rotational speeds. The main certified characteristic of a bearing - the basic dynamic load-carrying rating, denoted with symbol C,-is a load to be sustained by a rolling contact bearing over the time it makes one million revolutions. Depending on the bearing design, the dynamic load-carrying capacity of bearings as estimated in accordance with the ISO Recommendations on Rolling Bearings, is given in Tables of the present Catalogue.

The relationship between the basic rated resource, the dynamic load-carrying capacity rating, and the load acting on the bearing at a rotational speed of n>20 min-1 is calculated with the formula:

$$L_{10} = \left(\frac{C_r}{P}\right)^p \text{ million rotations .... 2.1}$$

Where L_{10} – is the basic rated resource, in million of revolutions;

C_r – is the basic dynamic load carrying capacity rating, N;

P – is the equivalent dynamic load, N;

P – is the exponent of a power, for ball bearings: $P = 3$;

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

The basic rated resource is mainly expressed in operating hours:

$$L_{10h} = \frac{1000000}{60n}\left(\frac{C_r}{P}\right)^p \text{, hour .... 2.2}$$

where L_{10} – is the basic rated resource, hour;

P – the rotational frequency, min-1

For vehicles, the basic rated resource of hub bearings is sometimes more convenient to express in total kilometers running:

$$L_{10s} = \frac{\pi D_1}{1000} L_{10}$$

Where L_{10s} –is the basic rated resource, million kilometers (mln.km);

D_1 – is the wheel diameter in meters, m.

Under normal operating conditions, the basic rated resource calculated at 90% reliability level (L_{10}) satisfies the majority of cases of bearings employment, since actually attainable life is more than calculated one, Also, at 50% reliability the serviice life (L_{50}) is, as a rule, five times as that of the basic rated resource (L_{10}). To imporve the compactness of bearing units and to reduce their weight, it is not recommended to overestimate the basic rated resource. However, in a number of technical fields another level of reliability is required. Besides, due to the extensive research and development activity, it has been found that the conditions of lubrication greatly affect the bearing service life. Hence, ISO has introduced a notation of basic rated resource, the formula of which is of the following form:

$$L_{na} = a_1 a_2 a_3 \left(\frac{C_r}{P}\right)^p \text{ or .... 2.3}$$

$$L_{na} = a_1 a_2 a_3 L_{10},$$

Where L_{na} – is the adjusted rated resource/million revolutions, Factor n means the difference between the given reliability and 100% level (e.g., at 95% reliability, $L_{na} = L_{5a}$);

a_1 – is the reliability factor;

a_2 – is the material factor;

a_3 – is the operating conditions factor.

For the generally adopted 90% reliability, as well as for proper bearing steel quality and lubrications conditions which ensure the separation of bearing surfaces in contact within the recommended limits, $a_1 = a_2 = a_3 = 1$ and the formula for the adjusted rated resource (3) becomes identical to the main formula 2.1.

Table 2.1 Values of the Reliability Factor

Reliability, percent	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

Whenever there is a necessity to carry out calculations for bearings with the reliability level in excess of 90%, the values of the reliability factor, a_1, shall be taken form Table 2.

Table 2.2 Factors a_{23}

Type of Bearing	Vacuum Treated Steel				
	Values of Viscosity Coefficient $c = n / n_1$				
	0.1-0.2	0.2-0.5	0.5-1	1-2	2-3
	Values of Factor a_{23}				
Radial and Angular Contact Ball Bearings	0.1-0.3	0.3-0.7	0.7-1.0	1.0-1.5	1.5-2.0
Roller Spherical Bearings, Double-row	0.1-0.2	0.2-0.4	0.4-0.7	0.7-1.0	1.0-1.2
Roller Berings, with Short Cylindrical Rollers or Needles	0.1-0.4	0.4-0.6	0.6-1.0	1.0-1.5	1.5-1.8
Spherical Roller Angular Contact Thrust Bearings	0.1-0.2	0.2-0.4	0.4-0.7	0.7-1.0	1.0-1.2

Notes :

1. For the case of ESR steel used and clean lubricants, factor a_{23} may be increase at $\chi > 2$,
2. In case of exececssive lubricant contamination wtih hard particles or poor oil circulation, a_{23} shall be taken to be 0.1.

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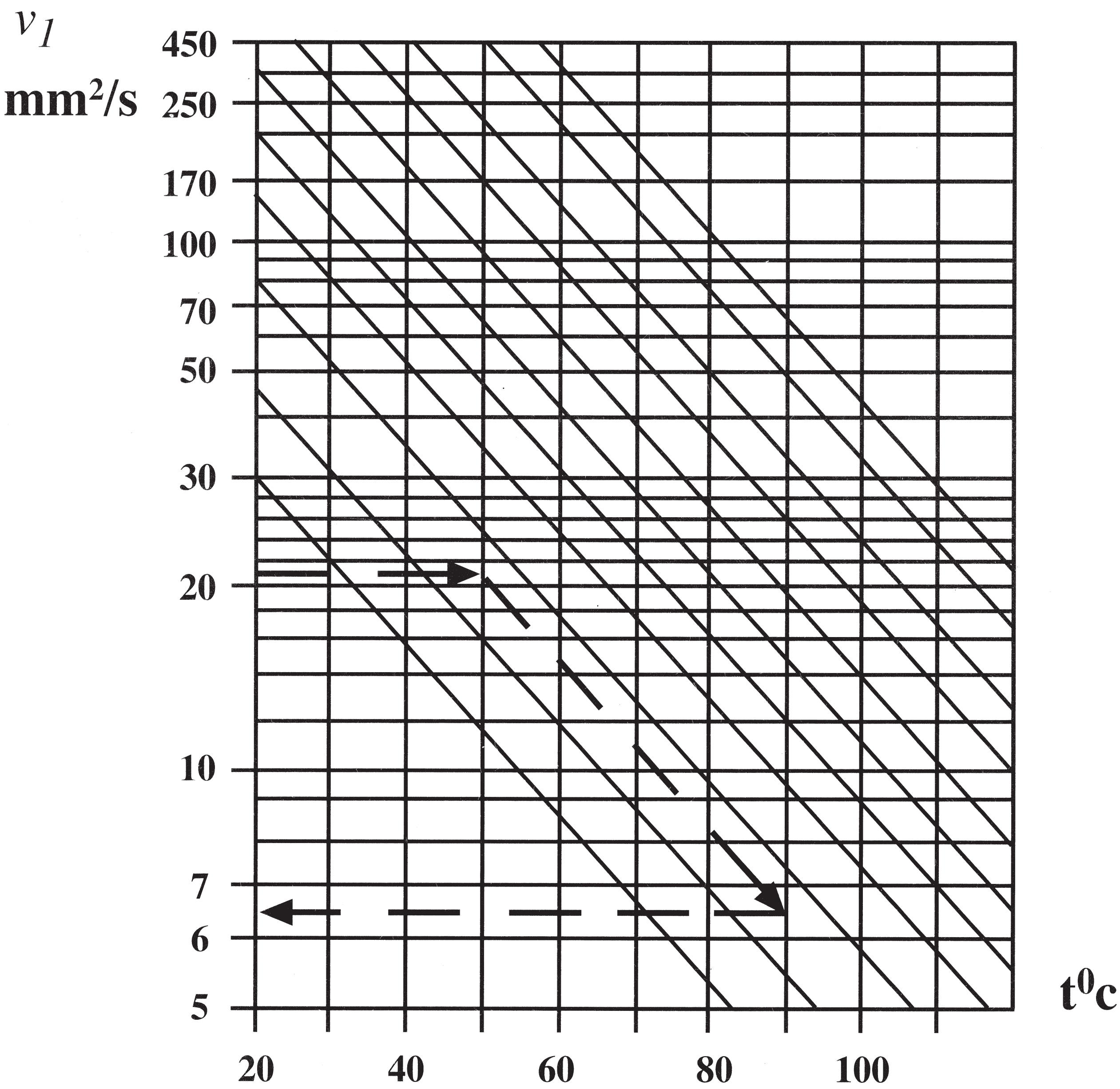

Fig. 1. A nomograph chart to determine lubricant viscosity at operating temperatures when its viscosity at basic temperature is known

However, it is expedient to use factor a_1 only in case of an increase in factor a_2 and a_3; otherwise, an increase in overall dimensions of the bearing results, hence, a reduction in its speed, and increase in its weight and sluggishness of the rotating parts of the machine associated with this bearing.

The operating conditions factor, a_3, specifies mainly lubricant conditions, as well misalignment, housing and shaft rigidity, bearing arrangement; clearances in bearings. Considering the fact that the use of special, higher-grade steels do not compensate the adverse effect of lubricant shortage, factore a_2, and a_3 are combined in one, with the notation a_{23}.

The factor a_{23} is selected form Table 3, by the ratio of normative and actual kinematic viscosity of the lubricant used:

$$\chi = \frac{v}{v_1} \quad .... \, 2.4$$

Where χ – is the viscosity coefficient;

v – is the kinematic viscosity of the oil actually used, at the bearing unit operating temperature, $mm^2;s^{-1}$;

v_1 – is the normative kinematic viscoity of oil as required to ensure lubrication conditions at a given velocity, $mm^2.s^{-1}$;

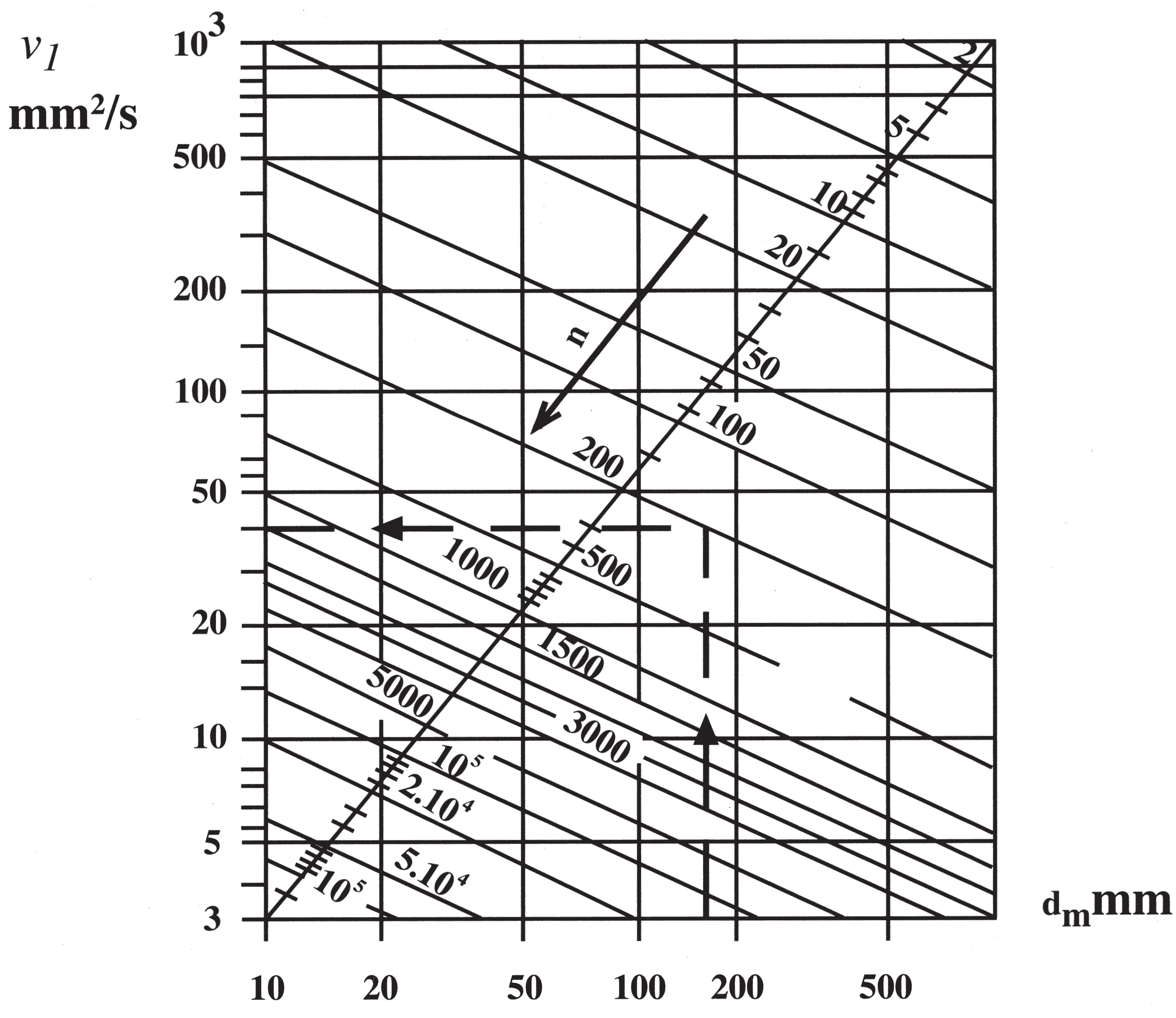

Fig. 2. A nomograph chart to determine normatice lubricant viscosity, v_1

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Bearing Data

Table 2.3. Recommended Values of the Basic Rated Resource for Machines of Different Type

Machine Type and Employment	L_{10h}, hr	L_{10s}, mln, km
Devices and mechanisms used at regular intervals, agricultural machines, household appliances	500-4000	
Machanisms used for a short periods of time, erecting cranes,building machines	4000-8000	
Critical mechanisms used intermittently (accessory mechanisms at power plant stations, conveyors for series production, elevators, metal-cutting machine tools used from time to time)	8000-12000	
One-shift operated machines, underloaded (stationery electric moters, reduction gears, crushers (mills)	12000-20000	
One-shift operated machines, under full load (metal-cutting machine-tools, wood-cutting machines), general-type machine-tools used in machine-building, lifting cranes, ventilators, separators, centriguges, polygraph equipment.	20000-30000	
Machines to be used on a round-the-clock basis (compressors, pumps, mine lifters, stationery electric motors, equipment used in textile industry)	40000-50000	
Hydropower stations, rotary funaces, deep-sea vessels engines	60000-100000	
Continuous-operation heavy-duty machines (paper working equipment, power plants, mine pumps, flexible shafts of deep-sea vessels)	100000	
Wheel-hubs of cars		0.2-0.3
Wheel-hubs of buses, industrial-type vehicles		0.3-0.5
Railway freight-car journal boxes		0.8
Suburbian car and tram journal boxes		1.5
Passenger-car journal boxes		3.0
Locomotive journal boxes		3.0-5.0

The values of the kinematic viscosity of oil, i.e., the operating viscosity, is determined with the help of a nompgraph, Fig 1. To obtain the operating viscosity, it is necessary to know the bearing temperature and the initial kinematic viscosity of the oil used. Fig 2. Contains a nomographic chart which is based on resilient hydrodynamic conditions of the lubricant, wherefrom we determine the normative (or standard) kinematic viscosity, v_1. This arbitrary kinematic viscosity of oil is chosen as function of the speed of motion of the contact element; the latter is obtained based on the following two parameter: the mean diameter and the rotational speed. For example, to calculate the standard viscosity of oil, v_1, for a bearing with a rotational speed of n = 200 min^{-1} and a mean diameter of dm = 150 mm, it is necessary-from the X-axis of mean diameters-to pass over to the corresponding rotational speed which is represented by an inclined line, and choose on the Y-axis the respective value of v_1 (v_1 = 44 mm2S-1 in Fig. 2. Indicated with the arrow).

The discussed procedure of determination of the viscosity coefficient is related to oil. For greases, this coefficient is found for a disperse media, i.e., on the base of the kinematic viscosity of the basic liquid oil which is a component of the grease. However, grease lubrication possesses certain special features of its own.

Most often than not, the designer knows the desired service life of the machine component in question. If these data are not available, the basic design life may be recommended from Table 4.

Equivalent Dynamic Load Calculation

Equivalent dynamic load (P) applied to radial and angular contact ball and roller bearings is a constant radial load that, when applied to a bearing with the inner ring running and the outer ring fixed, ensures the same design service life as that under actual load and rotation conditions. For bearings of the above-mention type, the equivalent load is found from the formula:

$$P_r = XF_r + YF_a \quad .... 2.5$$

Where P_r – is the equivalent dynamic load, H;
F_r – is the radial load constant in direction and value, H;
F_a – is the axial load constant in direction and value, H;
X – is the coefficient of radial load;
Y – is the coefficient of axial load;

In case of $F_a/F_r \leq e$, is assumed,

$$P_r = F_r \quad .... 2.6$$

Where e-is the limited value of F_a/F_r which determines the choice of factors X and Y.
Values of X, Y and e are specified in this Catalogue.

Accordingly, for an angular contact thrust bearing the equivalent dynamic load (P_a) is a constant axial load to be found in the same way:

$$P_a = XF_r + YF_a \quad .... 2.7$$

while for a thrust bearing it has the following form:

$$P_a = F_a \quad .... 2.8$$

The resultant load, F, acting upon th bearing can be determined rather accurately from laws of motion, if external forces are know, For example, loads transferred to the shafts/by machine elements are to be calculated as the reaction of the supports in accordance with equations for beams subjected to static loads. A shaft is regarded as a simply two supported beam resting in bearing supports. Using the momental equation and those for the sum of forces acting upon the beam, teh reaction of the supports is obtained; the latter, if taken with an opposite sign, represents the load applied to the bearing. The load is generated by the forces of the weight sustained by the bearing; by forces arising due to power transmission via the geartrain and/or belt transmission; by cutting forces in metal-cutting machine-tools; by inertial forces; by impact loads, etc.

The resultant load on the bearing, F, directed at any angle to the bearing axis of rotation, may be resolved into a radial (F_r) and axial (F_a) components, Sometimes, it is rather difficult to determine this load because of teh variety of force factors and application of incidental forces. Hence, any mathermatical techniques are applicable to calculated the same. For practical purposes, there may be recommended certain approved procedures for calculation of the resultant force, F.

If the force acting upon a bearing fluctuates linearly within P_{min} to P_{max} (e.g., at the supports of single-sided winding drums, then, the value of F has the form:

$$F = \frac{P_{min} + 2P_{max}}{3} \quad .... 2.9$$

If operating duties are of a varying nature, i.e., load F_1 acts within the period t_1, at a rotational speed n_1, while during the period t_2, at a rotation speed n_2 acts the load F_2 and so on, then, the amount F takes the form:

Equivalent Dynamic Load Calcuation

$$F=\left(\frac{n_1t_1F_1^{\,p}+n_2t_2F_2^{\,p}+n_it_iF_i^{\,p}}{n_1t_1+n_2t_2+...n_it_i}\right)^{\frac{1}{p}} \quad .... \ 2.10$$

where $p = 3$ for ball bearings, and

$p = \dfrac{10}{3}$ for roller bearings.

The assessment of average values of loads in accordance with the aobe-mentioned relationships is valid not only for radial loads but, also, for any load of constant diretion of application relative to the bearing radial plane. For radial bearing, a radial load is calculated, and for a thrust bearing the load applied along the bearing axis. Whenever the force generated by the load is applied at an angle to teh radial plane of teh bearing, radial and axial components are to be calculated. An equivalent load (radial one in case of radial bearings and axial for thrust bearings) is assessed with these components accounted for.

In case of a rotational load applied to a bearing (Fig. 3), the magnitude of the rotating force is found as follows:

$$F=mrw^2,\ H, \quad .... \ 2.11$$

Where m – is the mass of the rotating element, kg;
r – is the distance from the bearing axis to the centre of gravity of the rotating element, m;
w – is the angular velocity of the rotating element, rad/s.

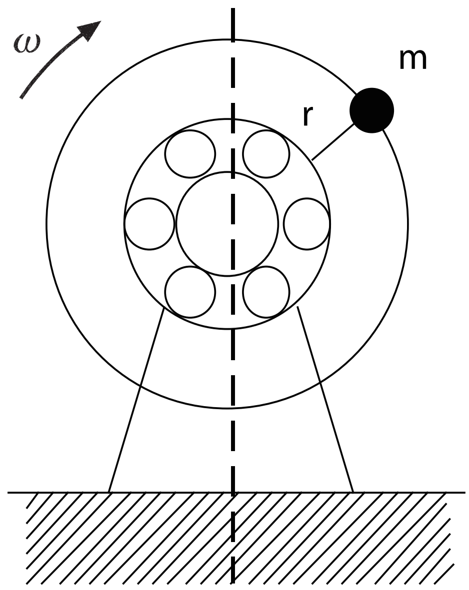

Fig. 3. Diagram of loading a bearing with rotational force.

Bearing Data

Table 2.4 Values of Loading Factor, K_6, as a Function of the Type of Loading and the Fields of Bearing Application

Type of Loading	K_6	Field of Application
Light jerks, short-time overloads up to 125% of the rated (nominal) load	1.0-1.2	Precision gear trains, Metal-cutting machine-tools (with the exception of slotting, planing, and grinding machine- tools). Gyroscopes. Lifting cranes component mechanisms. Electric tackles and monorail trucks. Mechanically-driven winches. Electric motors of low and average power. Light-duty ventilators and blowrs.
Moderate jerks, vibratory load; short-time overloads up to 150% of teh rated (nominal) load	1.2-1.5	Gear trains. Reduction gears of all types. Rail rolling stock journal boxes. Motion echanisms of crane trolleys. Crane swinging mechanisms, and boom overhang control mechanisms. Spindles of grinding machine-tools. Electrical spindles. Wheels of cars, buses, motocycles, motoroller. Agricultural machines.
Same, under conditions of improved reliability	1.5-1.8	Centrifuges and separators. Journal boxes and traction engines of electric locomotives. Machanisms of crane positioning. Wheels of trucks, tractors, prime movers, locomotives, crane and road-building machines. Power electric machines.Power generating plants.
Loads with considerable jerks and vibrations; short-time overloads up to 200% of the rated (nominal) load	1.8-2.5	Gears. Crushers and pile driver. Crank mechanisms. Ball and impact mills. Frame saws. Rolling mill rollers. High-powe ventilators and exhausters.

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In a number of cases, it is not quite easy to perform accurate calculations of loading a bearing. For example, journal boxes of teh rolling stock take up not only the carriage weight force which is easy to determine by calculation. When on move at varying speeds. bearings are subjected to impact loads at rail joints and when passing railroad switches, inertaial loads on turns and during emergency breaking. Whenever these factors cannot be accounted for accurately, one resorts to the experience accrued on the machines of earlier prodution.

Based on teh analysis of their operation, there has been derived a so-called loading factor, k_6, to be multiplied into teh equivalent load as obtained from the equations 2.5 to 2.8 . In the equivalent load the inertial forces, inherent to the vibration machines, sieves, and vibratory mills, have been already accounted for. For smooth mild loads, without jerks, in such mechanisms as low-power kinematic reduction gears and drives, rollers for supporting conveyor belts , pulley tackles, trolleys, controls drives and other similar mechanisms, the magnitude of the loading factor is $k_6 = 1$. The same value of the factor is taken if there is a belief in an accurate match between the calculated and actual loads. Table 2.5 contains recommended values of the loading factor k_6.

With the equivalent load (P) known, the basic rated resource (L_{10}) selected, the basic dynamic load-carring capacity (C) is determinded by computation, and the required standard size is chosen from the Catalogue with due account of Table 2.1.

Equivalent Static Load Calculation

For a bearing at rest, under load P, the service life equation (1) is inapplicable , since at L = 0.p = ∞, the bearing cannot accommodate load as high as is wished. At a low rotational speed $(n \leq 20\ min^6)$, P values turn out to be ovestated. Consequently, for bearings which rotate at low speeds, if at all, -especially when operated under impact loads-the allowable load depends on residual deformation orginating at points of contact of balls/rollers and rings rather than on the fatigue service life. The static load-carrying capacity of a bearing means the allowable load a bearing should withstand with no marked adverse impact on its further employment due to the residual deformation.

Thus, the purely radial load, or purely axial load-depending on whether the radial or angular contact bearings are in question-that results in combined(ring-ball/roller) residual deformation of up to 0,0001 diameter of the rolling elements, is termed the basic static load-carrying capacity, denoted in general as C_0, or C_{0r} or C_{0a} for radial and axial basic load carrying capacity, respectively. In accordance with the ISO Standard, this amount of the residual deformation is caused by a load that generates a maximum rated contact street at the most highly loaded rolling element which is 4200 MPa for bearings (with teh exception of self-aligning double-row bearings), and 4000 MPa for roller bearings. In this Catalogue, values of the basic static load-carring capacity are given as calculated on the above bases.

When testing a stationery (non-rotating) bearing for static load-carrying capacity under a load applied in any direction, it is necessary to calculate teh equivalent static load in that direction with which the static load-carrying capacity of the bearing is associated. This equivalent static load results in the same amount of residual deformation. For radial and angular contact ball and roller bearings the magnitude of the equivalent static load, P_0 is found from teh formula:

$$P_{or} = X_o F_r + Y_o F_a \quad .... 2.12$$

and for angular-contact thrust ball and roller bearings P_o is found as follows:

$$P_{oa} = F_a + 2{,}3 F_r\, tga \quad .... 2.13$$

Where
P_{or} – is the equivalent static radial load, H;
P_{oa} – is teh equvalent static axial load, H;
F_r – is the radial load or the radial component of the load acting upon the bearing, H;
P – is the axial load or the axial component of the load acting upont the bearing, H;
X_o – is the radial load coefficient;
Y_o – is the radial load coefficient;
a – is the nominal contact angle of a bearing, deg.

Thrust ball and roller bearings $(a = 90^o)$ are capable to withstand axial loads, only. The equal load for these types of bearings is calculated from teh formula $P_{oa} = F_a$.

The values of radial and axial load coefficients, as well as particular cases of application of Equations (12) and (13) are given in Tables of teh present Catalogue.

It is necessary that teh load acting upon a bearing not to exceed the tabulated basic load-carrying capacity (C_0). Deviations from this rule are based on experimental data. Thus, if the notion of the static safety coefficient $S_o \left(S_o = \dfrac{C_0}{P_0}\right)$ is introduced, then, for a smooth. i.e., without vibrations and jerk load, low rotational speed, and low accuracy requirements, $s_o > 0{,}5$ overload can be admitted; under normal operating conditionals, $s_o = 1\text{-}1{,}5$ is accepted in the general machine-tool building industry; under impact loads, periodic static loads and strict requirements to the accuracy, the load is limited down to $s = 1{,}5\text{-}2{,}5$.

3. Tolerances

For dimensional accuracy standards prescribe tolerances and allowable error limitations for those boundary dimensions (bore diameter, outside diameter, width, assembled bearing width, chamfer, and taper) necessary when installing bearings on shafts or in housings. For machining accuracy the standards provide allowable variation limits on bore, mean bore, outside diameter, mean outside diameter and raceway width or all thickness (for thrust bearings). Running accuracy is defined as the allowable limits for bearing runout. Bearing runout tolerances are included in the standards for inner and outer ring radial and axial runout; inner ring side runout with bore; and outer ring outside surface runout with side. Tolerances and allowable error limitations are established for each tolerance grade or class.

A comparison of relative tolerance class standards is shown in the Table 3.1.

Table 3.1 Comparison of tolerance classifications of national standards

Standard		Tolerence Class					Bearing Types
International Organization for Standardization	ISO 492	Normal Class Class 6X	Class 6	Class 5	Class4	Class 2	Radial bearings
	ISO 199	Normal Class	Class 6	Class 5	Class4	-	Thrust ball bearings
	ISO 578	Class 4	-	Class 3	Class 0	Class 00	Tapered roller Bearings (Inch series)
	ISO 1224	-	-	Class 5A	Class 4A	-	Precision instrument Bearings
Japanese Industrial Standard	JIS B 1514	class 0 class 6X	Class 6	Class 5	Class 4	Class 2	All type
Deutsches Institut	DIN 620	P0	P6	P5	P4	P2	All type
American National Standards Institute (ANSI) Anti-Friction Bearing Manufacturers (AFBMA)	ANSI/AFBMA Std.201)	ABEC-1 RBEC-1	ABEC-3 RBEC-3	ABEC-5 RBEC-5	ABEC-7	ABEC-9	Radial bearings (Except tapered Roller bearings)
	ANSI/AFBMA Std. 19.1	Class K	Class N	Class C	Class B	Class A	Tapered roller bearing (Metric series)
	ANSI / B 3.19 AFBMA Std.19	Class 4	Class 2	Class 3	Class 0	Class 00	Tapered roller bearings (Inch Series)
	ANSI/AFBMA Std. 12.1	-	Class 3P	Class 5P Class 5T	Class 7P Class 7T	Class 9P	Precision instrument ball bearings (Metric Series)
	ANSI/AFBMA Std. 12.2	-	Class 3P	Class 5P	Class 7P Class 5T	Class 9P Class 7T	Precision instrument ball bearings (Inch Series)

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Table 3.2 Bearing types and applicable tolerance

Bearing type		Applicable standard	Applicable tolerence				
Deep groove ball bearing		ISO 492	Class 0	Class 6	Class 5	Class 4	Class 2
Angular contact ball bearings			Class 0	Class 6	Class 5	Class 4	Class 2
Self-aligning ball bearings			Class 0	-	-	-	-
Cylindrical roller bearings			Class 0	Class 6	Class 5	Class 4	Class 2
Needle roller bearings			Class 0	Class 6	Class 5	Class 4	-
Spherical roller bearings			Class 0	-	-	-	-
Tapered Roller Bearings	Metric	ISO 492	Class 0,6X	Class 6	Class 5	Class 4	-
	Inc	AFBMA Std.19	Class 4	Class 2	Class 3	Class 0	Class 00
Thrust ball bearings		ISO 199	Class 0	Class 6	Class 5	Class 4	-

Codes and Symbols

Dimension

d : Nominal bore diameter

d_2 : Nominal bore diameter (double direction thrust ball bearing)

D : Nominal outside diameter

B : Nominal inner ring width or nominal center washer height

C : Nominal outer ring width1)

Note 1) For radial bearings (except tapered roller bearings) this is equivalent to the norminal bearings width.

T : Nominal bearing width of single row tapered roller bearing, or nominal height of single direction thrust bearing.

T_1 : Nominal height of double direction thrust ball bearing, or nominal effective width of inner ring and roller assembly of tapered roller bearing

T_2 : Nominal heigh form back face of housing washer to back face of center washer on double direction thrust ball bearings, or nominal effective outer ring width of tapered roller bearing.

r : Chamfer dimensions of inner and oute rings (for tapered roller bearings, large end of inner ring only)

r_1 : Chamfer dimensions of center washer, or small end of inner and outer ring of angular contact ball bearing, and large end of outer ring of tapered roller bearing.

r_2 : Chamfer dimensions of small end of inner and outer rings of tapered roller bearing

Dimension Deviation

- Δ_{ds} : Single bore diameter deviation
- $\Delta_{d\mathrm{mp}}$: Single plane mean bore diameter deviation
- $\Delta_{d2\mathrm{mp}}$: Single plane mean bore diameter deviation (double direction thrust ball bearing)
- Δ_{ds} : Single outside diameter deviation
- $\Delta_{d\mathrm{mp}}$: Single plane mean outside diameter deviation
- $\Delta_{B\mathrm{s}}$: Inner ring width deviation, or Centre washer height deviation
- $\Delta_{C\mathrm{s}}$: Outer ring width deviation
- $\Delta_{T\mathrm{s}}$: Overall width deviation of assembled signle row tapered roller bearing, or height deviation of single direction thrust bearing
- $\Delta_{T1\mathrm{s}}$: Height deviation of double direction thrust ball bearing, or effective width deviation of roller and inner ring assembly of tapered roller bearing
- $\Delta_{T2\mathrm{s}}$: Double direction thrust ball bearing housing washer back face to center washer back face height deviation, or tapered roller bearing outer ring effective width deviation

Dimension Variation

- $V_{d\mathrm{p}}$: Single radial plane bore diameter variation
- $V_{d2\mathrm{p}}$: Single radial plane bore diameter variation (double direction thrust ball bearing)
- $V_{d\mathrm{mp}}$: Mean single plane bore diameter variation
- $V_{D\mathrm{p}}$: Single radial plane outside diameter variation
- $V_{D\mathrm{mp}}$: Mean single plane outside diameter variation
- $V_{B\mathrm{s}}$: Inner ring width variation
- $V_{C\mathrm{s}}$: Outer ring width variation

Chamfer Boundary

- $r_{\mathrm{s\,min}}$: Minimum allowable chamfer dimension for inner/outer ring, or small end of inner ring on tapered roller bearing
- $r_{\mathrm{s\,max}}$: Maximum allowable chamfer dimension for inner/outer ring, or large end of inner ring on tapered roller bearing
- $r_{1\mathrm{s\,min}}$: Minimum allowable chamfer dimension for double direction thrust ball bearing center washer, small end of inner/outer ring of angular contact ball bearing, large end of outer ring of tapered roller bearing
- $r_{1\mathrm{s\,max}}$: Maximum allowable chamfer dimension for double direction thrust ball bearing center washer, small end of inner/outer ring of angular contact ball bearing, large end of outer ring of tapered roller bearing
- $r_{2\mathrm{s\,min}}$: Minium allowable chamfer dimension for small end of inner/outer ring of tapered roller bearing
- $r_{2\mathrm{s\,max}}$: Maximum allowable chamfer dimension for small end of inner/outer ring of tapered roller bearing

Rotation Tolerance

- K_{ia} : Inner ring radial runout
- S_{ia} : Inner ring axial runout (with side)
- S_{d} : Face runout with bore
- K_{ea} : Outer ring radial runout
- S_{ea} : Outer ring axial runout
- S_{d} : Outside surface inclination
- S_{i} : Thrust bearing shaft washer raceway (or center washer receway) thickness variation
- S_{e} : Thrust bearing housing washer raceway thickness variation

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Table 3.3 Tolerance for radial bearings (Except tapered roller bearings)
Inner Rings

Nominal bore diameter d (mm)		Δ_{dmp}										V_{dp} diameter series 7.8.9					V_{dp} diameter series 0.1					V_{dp} diameter series 2.3.4				
		class 0		class 6		class 5		class 4		class 2		class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2
over	incl.	high	low	high	low	high	low	high	low	high	low	Max					Max					Max				
0.61	2.5	0	-8	0	-7	0	-5	0	-4	0	-2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
2.5	10	0	-8	0	-7	0	-5	0	-4	0	-2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
10	18	0	-8	0	-7	0	-5	0	-4	0	-2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
18	30	0	-10	0	-8	0	-6	0	-5	0	-2.5	13	10	6	5	2.5	10	8	5	4	2.5	8	6	5	4	2.5
30	50	0	-12	0	-10	0	-8	0	-6	0	-2.5	15	13	8	6	2.5	12	10	6	5	2.5	9	8	6	5	2.5
50	80	0	-15	0	-12	0	-9	0	-7	0	-4	19	15	9	7	4	19	15	7	5	4	11	9	7	5	4
80	120	0	-20	0	-15	0	-10	0	-8	0	-5	25	19	10	8	5	25	19	8	6	5	15	11	8	6	5
120	150	0	-25	0	-18	0	-13	0	-10	0	-7	31	23	13	10	7	31	23	10	8	7	19	14	10	8	7
150	180	0	-25	0	-18	0	-13	0	-10	0	-7	31	23	13	10	7	31	23	10	8	7	19	14	10	8	7
180	250	0	-30	0	-22	0	-15	0	-12	0	-8	38	28	15	12	8	38	28	12	9	8	23	17	12	9	8
250	315	0	-35	0	-25	0	-18	-	-	-	-	44	31	18	-	-	44	31	14	-	-	26	19	14	-	-
315	400	0	-40	0	-30	0	-23	-	-	-	-	50	38	23	-	-	50	38	18	-	-	30	23	18	-	-
400	500	0	-45	0	-35	-	-	-	-	-	-	56	44	-	-	-	56	44	-	-	-	34	26	-	-	-
500	630	0	-50	0	-40	-	-	-	-	-	-	63	50	-	-	-	63	50	-	-	-	38	30	-	-	-
630	800	0	-75	-	-	-	-	-	-	-	-	94	-	-	-	-	94	-	-	-		55	-	-	-	-
800	1000	0	-100	-	-	-	-	-	-	-	-	125	-	-	-	-	125	-	-	-	-	75	-	-	-	
1000	1250	0	-125	-	-	-	-	-	-	-	-	155	-	-	-	-	155	-	-	-	-	94	-	-	-	
1250	1600	0	-160	-	-	-	-	-	-	-	-	200	-	-	-	-	200	-	-	-	-	120	-	-	-	
1600	2000	0	-200	-	-	-	-	-	-	-	-	250	-	-	-	-	250	-	-	-	-	150	-	-	-	

Table 3.3.2 Outer rings

Nominal bore diameter D (mm)		Δ_{dmp}										V_{dp} diameter series 7.8.9					V_{dp} diameter series 0.1					V_{dp} diameter series 2.3.4				
		class 0		class 6		class 5		class 4		class 2		class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2
over	incl.	high	low	high	low	high	low	high	low	high	low	Max					Max					Max				
2.5	6	0	-8	0	-7	0	-5	0	-4	0	-2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
6	18	0	-8	0	-7	0	-5	0	-4	0	-2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
18	30	0	-9	0	-8	0	-6	0	-5	0	-4	12	10	6	-	4	9	8	5	4	4	7	6	5	4	4
30	50	0	-11	0	-9	0	-7	0	-6	0	-4	14	11	7	-	4	11	9	5	5	4	8	7	5	5	4
50	80	0	-13	0	-11	0	-9	0	-7	0	-4	16	14	9	7	4	13	11	7	5	4	10	8	7	5	4
80	120	0	-15	0	-13	0	-10	0	-8	0	-5	19	16	10	-	5	19	16	8	6	5	11	10	8	6	5
120	150	0	-18	0	-15	0	-11	0	-9	0	-5	23	19	11	-	5	23	19	8	7	5	14	11	8	7	5
150	180	0	-25	0	-18	0	-13	0	-10	0	-7	31	23	13	-	7	31	23	10	8	7	19	14	10	8	7
180	250	0	-30	0	-20	0	-15	0	-11	0	-8	38	25	15	11	8	38	25	11	8	8	23	15	11	8	8
250	315	0	-35	0	-25	0	-18	0	-13	0	-8	44	31	18	13	8	44	31	14	10	8	26	19	14	10	8
315	400	0	-40	0	-28	0	-20	0	-15	0	-10	50	35	20	15	10	50	35	15	11	10	30	21	15	11	10
400	500	0	-45	0	-33	0	23	-	-	-	-	56	41	23	-	-	56	41	17	-	-	34	25	17	-	-
500	630	0	-50	0	-38	0	-28	-	-	-	-	63	48	28	-	-	63	48	21	-	-	38	29	21	-	-
630	800	0	-75	0	-45	0	-35	-	-	-	-	94	56	35	-	-	94	56	26	-	-	55	34	26	-	-
800	1000	0	-100	0	-60	-	-	-	-	-	-	125	75	-	-	-	125	75	-	-	-	75	45	-	-	-
1000	1250	0	-125	-	-	-	-	-	-	-	-	155	-	-	-	-	155	-	-	-	-	94	-	-	-	-
1250	1600	0	-160	-	-	-	-	-	-	-	-	200	-	-	-	-	200	-	-	-	-	120	-	-	-	-
1600	2000	0	-200	-	-	-	-	-	-	-	-	250	-	-	-	-	250	-	-	-	-	150	-	-	-	-
2000	2500	0	-250	-	-	-	-	-	-	-	-	310	-	-	-	-	310	-	-	-	-	190	-	-	-	-

Unit μm

V_{dmp}					K_{ia}					S_D			S_{ia} (1)			Δ_{Bs}						V_{Bs}				
class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 5	class 4	class 2	class 5	class 4	class 2	class 0.6		class 5.4		class 2		class 0	class 6	class 5	class 4	class 2
Max					Max					Max			Max			high	low	high	low	high	low	Max				
6	5	3	2	1.5	10	5	4	2.5	1.5	7	3	1.5	7	3	1.5	0	-40	0	-40	0	-40	12	12	5	2.5	1.5
6	5	3	2	1.5	10	6	4	2.5	1.5	7	3	1.5	7	3	1.5	0	-120	0	-40	0	-40	15	15	5	2.5	1.5
6	5	3	2	1.5	10	7	4	2.5	1.5	7	3	1.5	7	3	1.5	0	-120	0	-80	0	-80	20	20	5	2.5	1.5
8	6	3	2.5	1.5	13	8	4	3	2.5	8	4	1.5	8	4	2.5	0	-120	0	-120	0	-120	20	20	5	2.5	1.5
9	8	4	3	1.5	15	10	5	4	2.5	8	4	1.5	8	4	2.5	0	-120	0	-120	0	-120	20	20	5	3	1.5
11	9	5	3.5	2	20	10	5	4	2.5	8	5	1.5	8	5	2.5	0	-150	0	-150	0	-150	25	25	6	4	1.5
15	11	5	4	2.5	25	13	6	5	2.5	9	5	2.5	9	5	2.5	0	-200	0	-200	0	-200	25	25	7	4	2.5
19	14	7	5	3.5	30	18	8	6	2.5	10	6	2.5	10	7	2.5	0	-250	0	-250	0	-250	30	30	8	5	2.5
19	14	7	5	3.5	30	18	8	6	5	10	6	4	10	7	5	0	-250	0	-250	0	-300	30	30	8	5	4
23	17	8	6	4	40	20	10	8	5	11	7	5	13	8	5	0	-300	0	-300	0	-350	30	30	10	6	5
26	19	9	-	-	50	25	13	-	-	13	-	-	15	-	-	0	-350	0	-350	-	-	35	35	13	-	-
30	23	12	-	-	60	30	15	-	-	15	-	-	20	-	-	0	-400	0	-400	-	-	40	40	15	-	-
34	26	-	-	-	65	35	-	-	-	-	-	-	-	-	-	0	-450	-	-	-	-	50	45	-	-	-
38	30	-	-	-	70	40	-	-		-	-	-	-	-	-	0	-500	-	-	-	-	60	50	-	-	-
55	-	-	-	-	80	-	-	-	-	-	-	-	-	-	-	0	-750	-	-	-	-	70	-	-	-	-
75	-	-	-	-	90	-	-	-	-	-	-	-	-	-	-	0	-1000	-	-	-	-	80	-	-	-	-
94	-	-	-	-	100	-	-	-	-	-	-	-	-	-	-	0	-1250	-	-	-	-	100	-	-	-	-
120	-	-	-	-	120	-	-	-	-	-	-	-	-	-	-	0	-1600	-	-	-	-	120	-	-	-	-
150	-	-	-	-	140	-	-	-	-	-	-	-	-	-	-	0	-2000	-	-	-	-	140	-	-	-	-

(1) To be applied for deep groove ball bearing and angular contact ball bearings.

Unit μm

V_{dp} (2) capped bearings diameter series		V_{Dmp}					K_{ea}					S_D			S_{ea}			Δ_{Cs}	V_{Cs}			
class 0	class6	class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 5	class 4	class 2	class 5	class 4	class 2	all type	class 0.6	class 5	class 4	class 2
Max		Max					Max					Max			Max			Max	Max			
10	9	6	5	3	2	1.5	15	8	5	3	1.5	8	4	1.5	8	5	1.5	identical to	identical to	5	2.5	1.5
10	9	6	5	3	2	1.5	15	8	5	3	1.5	8	4	1.5	8	5	1.5	Δ_{Bs} of inner	Δ_{Bs} and Δ_{Bs}	5	2.5	1.5
12	10	7	6	3	2.5	2	15	9	6	4	2.5	8	4	1.5	8	5	2.5	ring of same	of inner ring	5	2.5	1.5
16	13	8	7	4	3	2	20	10	7	5	2.5	8	4	1.5	8	5	2.5	bearing	of same	5	2.5	1.5
20	16	10	8	5	3.5	2	25	13	8	5	4	8	4	1.5	10	5	4		bearing	6	3	1.5
26	20	11	10	5	4	2.5	35	18	10	6	5	9	5	2.5	11	6	5			8	4	2.5
30	25	14	11	6	5	2.5	40	20	11	7	5	10	5	2.5	13	7	5			8	5	2.5
38	30	19	14	7	5	3.5	45	23	13	8	5	10	5	2.5	14	8	5			8	5	2.5
-	-	23	15	8	6	4	50	25	15	10	7	11	7	4	15	10	7			10	7	4
-	-	26	19	9	7	4	60	30	18	11	7	13	8	5	18	10	7			11	7	5
-	-	30	21	10	8	5	70	35	20	13	8	13	10	7	20	13	8			13	8	7
-	-	34	25	12	-	-	80	40	23	-	-	15	-	-	23	-	-			15	-	-
-	-	38	29	14-	-	-	100	50	25	-	-	18	-	-	25	-	-			18	-	-
-	-	55	34	18	-	-	120	60	30	-	-	20	-	-	30	-	-			20	-	-
-	-	75	45	-	-	-	140	75	-	-	-	-	-	-	-	-	-			-	-	-
-	-	94	-	-	-	-	160	-	-	-	-		-	-	-	-	-			-	-	-
-	-	120	-	-	-	-	190	-	-	-	-	-	-	-	-	-	-			-	-	-
-	-	150	-	-	-	-	220	-	-	-	-	-	-	-	-	-	-			-	-	-
-	-	190	-	-	-	-	250	-	-	-	-	-	-	-	-	-	-			-	-	-

(2) To be applied in case snap rings are not installed on the bearings.

Table 3.4 Tolerance of tapered roller bearings (Metric)
Inner rings

Nominal bore diameter d (mm)		Δ_{dmp}						V_{dp}				V_{dmp}				K_{ia}				S_d	
		class 0.6x		class 5,6		class 4		class 0.6x	class 6	class 5	class 4	class 0.6x	class 6	class 5	class 4	class 0.6x	class 6	class 5	class 4	class 5	class 4
over	incl.	high	low	high	low	high	low	Max				Max				Max				Max	
10	18	0	-12	0	-7	0	-5	12	7	5	4	9	5	5	4	15	7	5	3	7	3
18	30	0	-12	0	-8	0	-6	12	8	6	5	9	6	5	4	18	8	5	3	8	4
30	50	0	-12	0	-10	0	-8	12	10	8	6	9	8	5	5	20	10	6	4	8	4
50	80	0	-15	0	-12	0	-9	15	12	9	7	11	9	6	5	25	10	7	4	8	5
80	120	0	-120	0	-15	0	-10	20	15	11	8	15	11	8	5	30	13	8	5	9	5
120	180	0	-25	0	-18	0	-13	25	18	14	10	19	14	9	7	35	18	11	6	10	6
180	250	0	-30	0	-22	0	-15	30	22	17	11	23	16	11	8	50	20	13	8	11	7
250	315	0	-35	-	-	-	-	35	-	-	-	26	-	-	-	60	-	-	-	-	-
315	400	0	-40	-	-	-	-	40	-	-	-	30	-	-	-	70	-	-	-	-	-
400	500	0	-45	-	-	-	-	45	-	-	-	34	-	-	-	80	-	-	-	-	
500	630	-	-50	-	-	-	-	50	-	-	-	38	-	-	-	90	-	-	-	-	
630	800	0	-75	-	-	-	-	75	-	-	-	56	-	-	-	105	-	-	-	-	
800	1000	0	-100	-	-	-	-	100	-	-	-	75	-	-	-	120	-	-	-	-	

Outer rings

Nominal outside diameter D (mm)		Δ_{Dmp}						V_{Dp}				V_{Dmp}				K_{ea}				S_D	
		class 0.6x		class 5,6		class 4		class 0.6x	class 6	class 5	class 4	class 0.6x	class 6	class 5	class 4	class 0.6x	class 6	class 5	class 4	class 5	class 4
over	incl.	high	low	high	low	high	low	Max				Max				Max				Max	
18	30	0	-12	0	-8	0	-6	12	8	6	5	9	6	5	4	18	9	6	4	8	4
30	50	0	-14	0	-9	0	-7	14	9	7	5	11	7	5	5	20	10	7	5	8	4
50	80	0	-16	0	-11	0	-9	16	11	8	7	12	8	6	5	25	13	8	5	8	4
80	120	0	-18	0	-13	0	-10	18	13	10	8	14	10	7	5	35	18	10	6	9	5
120	150	0	-20	0	-15	0	-11	20	15	11	8	15	11	8	6	40	20	11	7	10	5
150	180	0	-25	0	-18	0	-13	25	18	14	10	19	14	9	7	45	23	13	8	10	5
180	250	0	-30	0	-20	0	-15	30	20	15	11	23	15	10	8	50	25	15	10	11	7
250	315	0	-35	0	-25	0	-18	35	25	19	14	26	19	13	9	60	30	18	11	13	8
315	400	0	-40	0	-28	0	-20	40	28	22	15	30	21	14	10	70	35	20	13	13	10
400	500	0	-45	-	-	-	-	45	-	-	-	34	-	-	-	80	-	-	-	-	-
500	630	0	-50	-	-	-	-	50	-	-	-	38	-	-	-	100	-	-	-	-	
630	800	0	-75	-	-	-	-	75	-	-	-	56	-	-	-	120	-	-	-	-	
800	1000	0	-100	-	-	-	-	100	-	-	-	75	-	-	-	140	-	-	-	-	
1000	1250	0	-125	-	-	-	-	125	-	-	-	84	-	-	-	165	-	-	-	-	
1250	1600	0	-160	-	-	-	-	160	-	-	-	120	-	-	-	190	-	-	-	-	

Unit μm

S_{ia}	Δ_{Bs}						Δ_{Ts}						$\Delta_{B1s}, \Delta_{C1s}$		$\Delta_{B2s}, \Delta_{C2s}$	
class 4	class 0.6		class 6X		class 4,5		class 0.6		class 6X		class 4,5		class 0, 6, 5		class 0, 6, 5	
max	high	low	high	low	high	low	high	low	high	low	high	low	high	low	high	low
3	0	-120	0	-50	0	-200	+200	0	+100	0	+200	-200	-	-	-	-
4	0	-120	0	-50	0	-200	+200	0	+100	0	+200	-200	-	-	-	-
4	0	-120	0	-50	0	-240	+200	0	+100	0	+200	-200	+240	-240	-	-
4	0	-150	0	-50	0	-300	+200	0	+100	0	+200	-200	+300	-300	-	-
5	0	-200	0	-50	0	-400	+200	-200	+100	0	+200	-200	+400	-400	+500	-500
7	0	-250	0	-50	0	-500	+350	-250	+150	0	+350	-250	+500	-500	+600	-600
8	0	-300	0	-50	0	-600	+350	-250	+150	0	+350	-250	+600	-600	+750	-750
-	0	-350	0	-50	-	-	+350	-250	+200	0	-	-	+700	-700	+900	-900
-	0	-400	0	-50	-	-	+400	-400	+200	0	-	-	+800	-800	+1000	-1000
-	0	-450	-	-	-	-	-	-	-	-	-	-	+900	-900	+1200	-1200
-	0	-500	-	-	-	-	-	-	-	-	-	-	+1000	-1000	+1200	-1200
-	0	-750	-	-	-	-	-	-	-	-	-	-	+1500	-1500	+1500	-1500
-	0	-1000	-	-	-	-	-	-	-	-	-	-	+1500	-1500	+1500	-1500

Unit μm

S_{ea}	Δ_{Cs}		
class 4	class 0, 6, 5, 4	class 6X	
max	high	low	high
5	Identical to Δ_{Bs} inner ring of same bearing	0	-100
5		0	-100
5		0	-100
6		0	-100
7		0	-100
8		0	-100
10		0	-100
13		0	-100
-		0	-100

Effective width of outer and inner withroller

Unit μm

Nominal bore diameter d (mm)		Δ_{R1s}				Δ_{R2s}			
		class 0		class 6X		class 0		class 6X	
over	incl.	high	low	high	low	high	low	high	low
10	18	+100	0	+50	0	+100	0	+50	0
18	30	+100	0	+50	0	+100	0	+50	0
30	50	+100	0	+50	0	+100	0	+50	0
50	80	+100	0	+50	0	+100	0	+50	0
80	120	+100	-100	+50	0	+100	-100	+50	0
120	180	+150	-150	+50	0	+200	-100	+100	0
180	250	+150	-150	+50	0	+200	-100	+100	0

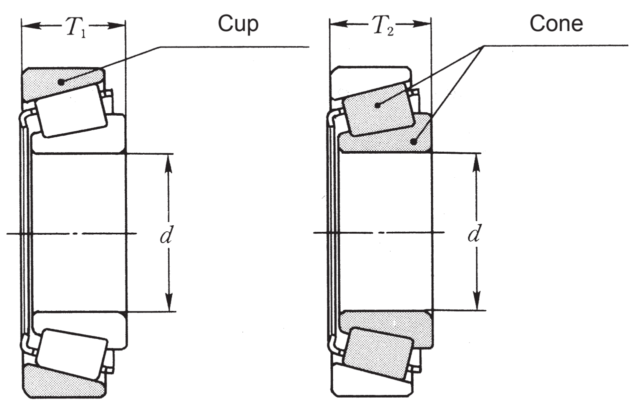

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Table 3.7 Tolerance of thrust ball bearings
Inner rings

		Δ_{dmp}, Δ_{d2mp}				V_{dp}, V_{d2D}		S_i 2)			
		class 0, 6, 5		class 4		class 0, 6, 5	class 4	class 0	class 6	class 5	class 4
over	incl.	high	low	high	low	max		max			
-	18	0	-8	0	-7	6	5	10	5	3	2
18	30	0	-10	0	-8	8	6	10	5	3	2
30	50	0	-12	0	-10	9	8	10	6	3	2
50	80	0	-15	0	-12	11	9	10	7	4	3
80	120	0	-20	0	-15	15	11	15	8	4	3
120	180	0	-25	0	-18	19	14	15	9	5	4
180	250	0-	30	0	-22	23	17	20	10	5	4
250	315	0	-35	0	-25	26	19	25	13	7	5
315	400	0	-40	0	-30	30	23	30	15	7	5
400	500	0	-45	0	-35	34	26	30	18	9	6
500	630	0	-50	0	-40	38	30	35	21	11	7

1) The division of double type bearings will be in accordance with divion "d" of single direction tyep bearings corresponding to th identical nominal outer diameter of bearings, not according to division "d2"

Outer rings

Nominal outside diameter D (mm)		Δ_{Dmp}				V_{Dp}		S_e 2)			
		class 0, 6, 5		class 4		class 0, 6, 5	class 4	class 0	class 6	class 5	class 4
over	incl.	high	low	high	low	max		max			
10	18	0	-11	0	-7	8	5	According to the tolerance of S_1 against "d" or "d_2" of the same bearings			
18	30	0	-13	0	-8	10	6				
30	50	0	-16	0	-9	12	7				
50	80	0	-19	0	-11	14	8				
80	120	0	-22	0	-13	17	10				
120	180	0	-25	0	-15	19	11				
180	250	0	-30	0	-20	23	15				
250	315	0	-35	0	-25	26	19				
315	400	0	-40	0	-28	30	21				
400	500	0	-45	0	-33	34	25				
500	630	0	-50	0	-38	38	29				
630	800	0	-75	0	-45	55	34				

2) To be applied only for bearings with flat

Height of bearings center washer

Unit μm

Nominal bore diameter d (mm)		Single direction type		Double direction type					
		Δ_{Ts}		Δ_{T1s} 3)		Δ_{T2s} 3)		Δ_{T3s} 3)	
over	incl.	high	low	high	low	high	low	high	low
-	30	0	-75	+50	-150	0	-75	0	-50
30	50	0	-100	+75	-200	0	-100	0	-75
50	80	0	-125	+100	-250	0	-125	0	-100
80	120	0	-150	+125	-300	0	-150	0	-125
120	180	0	-175	+150	-350	0	175	0	-150
180	250	0	-200	+175	-400	0	-200	0	-175
250	315	0	-225	+200	-450	0	-225	0	-200
315	400	0	-300	+250	-600	0	-300	0	-250
400	500	0	-350	-	-	-	-	-	-
500	630	0	-400	-	-	-	-	-	-

3) To be in accordance with the division "d" of single direction type bearings corresponding to the identical outer diameter of bearings in the same bearings series.

Table 3.8 Tolerance of sphrical thrust roller bearing

Inner rings

Unit μm

Nominal bore diameter d (mm)		Δ_{dmp}		V_{dp}	S_d	Δ_{Ts}	
over	incl.	high	low	max	max	high	low
50	80	0	-15	11	25	+150	-15
80	120	0	-20	15	25	+200	-200
120	180	0	-25	19	30	+250	-250
180	250	0	-30	23	30	+300	-300
250	315	0	-35	26	35	+350	-300
315	400	0	-40	30	40	+400	-400
400	500	0	-45	34	45	+450	-450

Outer ring

Unit μm

Nominal bore diameter D (mm)		Δ_{Dmp}	
over	incl.	high	low
120	180	0	-25
180	250	0	-30
250	315	0	-35
315	400	0	-40
400	500	0	-45
500	630	0	-50
630	800	0	-75
800	1000	0	-100

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4. Fits

4.1 Interference

Bearing rings are fixed on the shaft or in the housing so that slip or movement does not occur between the mated surface during operation or under load.

This relative movement, creep, between the fitted surfaces of the bearing and the shaft or housing can ocur in a radial direction, or in an axial direction, or in the direction of rotation. This creeping movement under load causes damage to the bearing rings, shaft or housing in the form of abrasive wear, fretting corrosionor friction crack. This can also lead to abrasive particles getting into the bearing, which can cause vibration, excessive heat, and lowered rotational efficiency. To insure that slip does not occur between the fitted surfaces of the bearing rings and the shaft or housing, the bearing is usually installed with an interference fit.

Most effective interference fit is called a tight fit or shrink fit. The advantage of this tight fit for thin walled bearings is that it provides uniform load support over the entire ring circumference without any loss in load carrying capacity.

However, with a tight interference-fit, ease of mounting and dismounting the bearing is lost; and when using a non-separable bearing as a non-fixing bearing, axial displacement is impossible.

4.2 Calculation

Load and interference

The minimum required amount of interference for the inner rings mounted on solid shafts when acted on lby radial load, is found by formulae 4.1 and 4.2.

When $F_r \leqq 0.3\ C_{or}$

$$\Delta_{dF} = 0.08\sqrt{\frac{d \cdot F_r}{B}} \quad \cdots\cdots 4.1$$

When $F_r > 0.3\ C_{or}$

$$\Delta_{dF} = 0.02\frac{F_r}{B} \quad \cdots\cdots 4.2$$

Where,

Δ_{dF} : Required effective interference (for load) μm
d : Nominal bore diameter mm
B : Inner ring width mm
F_r : Radial load N
C_{or} : Basic static rated load N

Temperature rise and interference

To prevent loosening of the inner ring on steel shafts due to temperature increases (difference between bearing temperature and ambient temperature) caused by bearing roatation, and interference fit must be given. The required amount of interference can be found by formula (4.3).

$$\Delta_{dT} = 0.0015 \cdot d \cdot \Delta T \quad \cdots\cdots 4.3$$

Where,

Δ_{dT}: Required effective interference (for temperature) μm
ΔT : Difference between bearing temperature and ambient temperature ℃
d : Bearing bore diameter mm

Effective interference and apparent interference

The effective interference (the actual interference after fitting) is different from the apparent interference derived from the dimensions measured value. This differenct is due to the roughness or slight variations of the mating surfaces, and this slight flattening of the uneven surfaces at the time of fitting is taken into consideration.
The relation between the effective and apparent interference, which varies according to the finish given to the mating surfaces, is expressed by formula (4.4).

$$\Delta d_{eff} = \Delta d_f - G \quad \cdots\cdots 4.4$$

Where,

Δd_{eff} : Effective interference μm
Δd_f : Apparent interference μm
G = 1.0 ~ 2.5 μm for ground shaft
= 5.0 ~ 7.0 μm for turned shaft

Maximum interference

When bearing rings are installed with an interference fit on shafts or in housings, tension or compression stree may occur. If the interference is too large, it may cause damage to the bearing rings and reduce the fatigut life of the bearing. For these reasons, the maximum amount of interference should be less than 1/1 000 of the shaft diameter, or

4.3 Selection

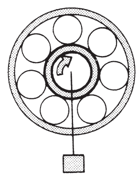
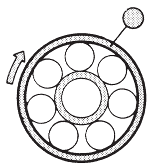
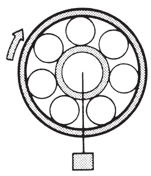
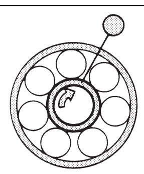
Selection of the proper fit is generally based on the following factors: 1) the direction and nature of the bearing load 2) whether the inner ring or outer ring rotates 3) whether the load on teh inner or outer ring rotates or not 4) whether there is static load or direction indeterminate load or not.
For bearings under rotating loads or direction indeterminate loads, a tight fit is recommended; but for static loads, a transition fit or loose fit should be sufficient.

The interference should be tighter for heavy bearing loads or vibration and shock load conditions. Also, a tighter than normal fit should be given when the bearing is installed on hollow shafts or in housings with thin walls, or housingsa made of light alloys or plastic.

In applications where high rotational accuracy must be maintained, high precision bearings and high tolerance shafts and housing should be employed instead of a tighter interference fit to ensure bearing stability. High interference fits should be avoided if possible as they cause shaft or housing deformities to be induced into the bearing rings, and thus reduce bearing rotational accuracy.

Because mounting and dismounting become very difficult when both the inner ring and outer ring of a non-separable bearing (for example a deep groove ball bearing) are given tight interference fits, one or the other ring should be given a loose fit.

Table 4.1 Radial Load and bearing fit

Bearing rotation and load	Illustration	Ring load	Fit
Inner ring : Rotating Inner ring : Rotating Load direction : Constant	 Static Load	Rotating inner ring load	Inner ring : Tight Fit
Inner ring : Stationery Outer ring : Rotating Load direction : Rotates with outer ring	 Unbalanced Load	Static outer	Outer ring : Loose fit ring load
Inner ring : Stationery Outer ring : Rotating Load direction : Constant	 Static Load	Static inner ring load	Inner ring : Loose fit
Inner ring : Rotating Outer ring : Stationery Loan direction : Rotates with inner ring	 Unbalanced Load	Rotating outer ring load	Outer ring : Tight fit

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4.4 Recommended fits

Metric size standard dimension tolerances for bearing shaft diameters and housing bore diameters are governed by ISO 286.

Accordingly, bearing fits are determined by the precision (dimensional tolerance0 of the shaft diameter and housing bore diameter. Widely used fits for various shaft and housing bore diameter tolerances, and bearing bore and outside diameters are shon in Fig. 4.1.

Generally, recommended fits relating to the primary factors of bearing shape, dimensions, and load conditions are listed in Tables 4.2 and 4.3.

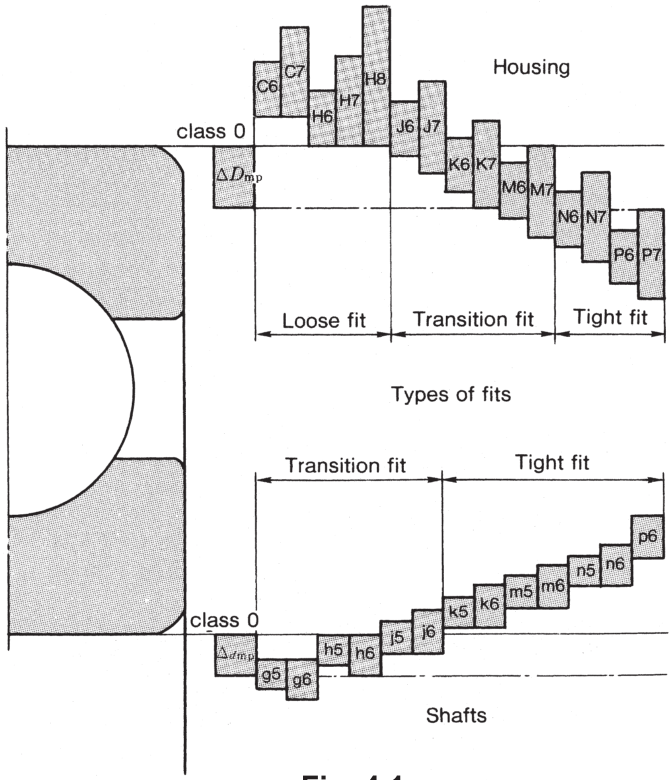

Fig. 4.1

Table 4.2 General standards for radial bearing fits
Housing fit

Housing type	Load condition		Housing fits
Solid or split housing	Outer ring static load	all load conditions	H7
		Heat conducted throuh shaft	G7
	Direction indeterminate load	Light to normal	JS7
Solid housing		Normal to heavy	K7
		Heavy shock	M7
	Outer ring rotating load	Light or variable	M7
		Normal to heavy	N7
		Heavy (thin wall housing)	P7
		Heavy shock	P7

Note : Fits apply to cast iron or steel housings. For light alloy housings, a tighter fit than listed is required.

Table 4.2 Cylindrical bore radial bearings, Shaft fit

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

Table 4.3 for electric motor bearings, Shaft / Housing fit

Shaft or housing	Deep groove ball bearings			Cylindrical roller bearings		
	Shaft or housing bore diameter mm		Fits	Shaft or housing bore diameter mm		Fits
	over	incl.		over	incl.	
Shaft	-	18	j5	-	40	k5
	18	100	k5	40	160	m5
	100	160	m5	160	200	n5
Housing	All sizes		H6 or J6	All sizes		H6 or J6

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5 Clearance

5.1 Internal clearance

Internal clearance of a bearing is the amount of internal clearance a bearing has before being installed on a shaft or in a housing.

As in Fig.5.1, when either the inner ring or the outer ring is fixed and the other ring is free to move, displacement can take place in either an axial or radial direction. This amount of displacement (radially or axially) is termed the internal clearance and, depending on the direction, is called the radial internal clearance or the axial internal clearance.

When the internal clearance of a bearing is measured, a slight measurement load is applied to the raceway so the internal clearance may be measured accurately. However, at this time, a slight amount of elastic deformation of the bearing occurs under the measurement load, and the clearance measurement value is slightly larger than the true clearance. This discrepancy between the true bearing clearance and the increased amount due to the elastic deformation must be compensated for. These compensation values are given in Table 5.1. For roller bearings the amount of elastic deformation can be ignored.

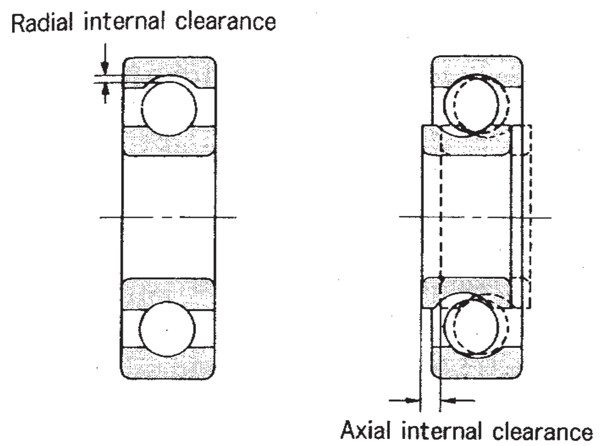

Fig. 5.1 Internal clearance

Table 5.1 Adjustment of radial internal clearance based on measured load Unit μm

Nominal bore diameter of bearing d (mm)		Measuring Load (N)	Radial Clearance Increase				
over	incl		C2	Normal	C3	C4	C5
10	18	24.5	3~4	4	4	4	4
18	50	49	4~5	5	6	6	6
50	200	147	6~8	8	9	9	9

5.2 Internal clearance selection

The internal clearance of a bearing under operating conditions (effective clearance) is usually smaller than the same bearing's initial clearance before being installed and operated. This is due to serveral factors including bearing fit, the difference in temperature between the inner and outer rings, etc. As a bearing's operating clearance has an effect on bearing life, heat generation, vibration, noise, etc.; care must be taken in selectng the most suitable operating clearance.

Effective internal clearance:

The internal clearance differential between the initial clearance and the operating (effective) clearance (the amount of clearance reduction caused by interference fits, or clearance variation due to the temperature difference between the inner and outer rings) can be calculated by the following formula:

When $\delta_{eff} = \delta_o - (\delta_f + \delta_t)$ 5.1

Where,

δ_{eff} : Effective internal clearance mm
δ_o : Bearing internal clearance mm
δ_f : Reduced amount of clearance due to interference mm
δ_t : Reduced amount of clearance due to temperature differential of inner and outer rings mm

Reduced clearance due to interference:

When bearings are installed with interference fits on shafts and in housings, the inner ring will expand and the outer ring will contract; thus reducing the bearings' internal clearance. The amount of expansion or contraction varies depending on the shape of the bearing, the shap of the shaft or housing, dimensions of the respective parts, and the type of material used. The differential can range from approximately 70% to 90% of the effective interference.

$\delta_f = (0.70 \sim 0.90) \cdot \Delta_{deff}$ 5.2

Where,

δ_f : Reduced amount of clearance due to interference mm
Δ_{deff} : Effective interference mm

Reduced internal clearance due to inner/outer ring temperature difference:

During operation, normally the outer ring will be from 5x to 10xC cooler than the inner ring or rotating parts. However, if the colling effect of the housing is large, the shaft is Connected to a heat source, or a heated substance is conducted through the hollow shaft; the temperature difference between the two rings can be even greater. The amount of internal clearance is thus further reduced by the differential expansion of the two rings.

$$\delta_t = \alpha \cdot \Delta_r \cdot D_o \qquad 5.3$$

Where,

δ_t : Amount of reduced clearance due to heat differential mm

α : Bearing steel linear expansion coefficietn 12.5x10-6/XC

Δ_r : Inner/outer ring temperature differential XC

D_o : Outer ring raceway diameter mm

Outer ring raceway diameter, Do, values can be approximated by using formula (8.4) or (8.5).

For ball bearings and self-aligning roller bearings,

$$D_o = 0.20\,(d + 4.0D) \qquad 5.4$$

For roller bearings (except self-aligning)

$$D_o = 0.25\,(d + 3.0D) \qquad 5.5$$

Where, d : Bearing bore diameter mm

D : Bearing outside diameter mm

5.3 Bearing internal clearance seletion standards

Theoretically, as regards bearing life, the optimum operating internal clearance for any bearing would be a slight negative clearance after the bearing had reached normal operating temperature.

Under actual operating conditions, maintaining such optimum tolerances is often difficult at best. Due to various fluctuating operating conditions this slight minus clearance can quickly become a large minus, greatly lowering the life of teh bearing and causing excessive heat to be generated. Therefore, an initial internal clearance which will result in a slightly greater than minus internal operating clearance should be selected.

Under normal operating conditions (e.g. normal load, fit, speed, temperature, etc.), a standard internal cleracne will give a very staisfactory operating clearance.

Table 5.2. lists non-standard clearance recommendations for various applications and operating conditions.

Table 5.2 Examples of applications where bearing clearances other than normal clearance are used

Operating conditions	Applications	Selected clearance
With heavy or shock load, clearance is great.	Railway vehicle axles	C3
	Vibration screens	C3, C4
With direction indeterminate load both inner and outer rings are tight-fitted.	Railway vehicle traction motors	C4
	Tractors and final speed regulators	C4
Shaft or inner ring is heated	Paper making machines and driers	C3, C4
	Rolling mill table rollers	C3
Clearance fit for both inner and outer rings	Rolling mill roll rollers	C2
to reduce noise and vibration when rotating.	Micromotors	C2

Table 5.3 Radial internal clearance of bearing for electric motor

Unit μm

Nominal bore diameter d mm		Radical internal clearance, E			
		Deep groove ball bearings		Cylindrical[2] roller bearings	
over	incl.	min	max	min	max
10	18	4	11	-	-
18	24	5	12	-	-
24	30	5	12	15	30
30	40	9	17	15	30
40	50	9	17	20	35
50	65	12	22	25	40
65	80	12	22	30	45
80	100	18	30	35	55
100	120	18	30	35	60
120	140	24	38	40	65
140	160	24	38	50	80
160	180	-	-	60	90
180	200	-	-	65	100

1) Suffix E is added to bearing numbers.
2) Non-interchargeable clearance.

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Table 5.4. Radial internal clearance of deep groove ball bearings.

Unit μm

Nominal bore diameter *d* mm		C2		Normal		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
-	2.5	0	6	4	11	10	20	-	-	-	-
2.5	6	0	7	2	13	8	23	-	-	-	-
6	10	0	7	2	13	8	23	14	29	20	37
10	18	0	9	3	18	11	25	18	33	25	45
18	24	0	10	5	20	13	28	20	36	28	48
24	30	1	11	5	20	13	28	23	41	30	53
30	40	1	11	6	20	15	33	28	46	40	64
40	50	1	11	6	23	18	36	30	51	45	73
50	65	1	15	8	28	23	43	38	61	55	90
65	80	1	15	10	30	25	51	46	71	65	105
80	100	1	18	12	36	30	58	53	84	75	120
100	120	2	20	15	41	36	66	61	97	90	140
120	140	2	23	18	48	41	81	71	114	105	160
140	160	2	23	18	53	46	91	81	130	120	180
160	180	2	25	20	61	53	102	91	147	135	200
180	200	2	30	25	71	63	117	107	163	150	230
200	225	4	32	32	82	82	132	132	187	197	255
225	250	4	36	36	92	92	152	152	217	217	290
250	280	4	39	39	97	97	162	162	237	237	320
280	315	8	50	50	110	110	180	180	260	260	350
315	355	8	50	50	120	120	200	200	290	290	380
355	400	8	60	60	140	140	230	230	330	330	430
400	450	10	70	70	160	160	260	260	370	-	-
450	500	10	80	80	180	180	290	290	410	-	-
500	560	20	90	90	200	200	320	320	460	-	-
560	630	20	100	100	220	220	350	350	510	-	-

Table 5.5 Radial internal clearance of self-aligning ball bearings, cylindrical bore

Unit μm

Nominal bore diameter *d* mm		C2		Normal		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
2.5	6	1	8	5	15	10	20	15	25	21	33
6	10	2	9	6	17	12	25	19	33	27	42
10	14	2	10	6	19	13	26	21	35	30	48
14	18	3	12	8	21	15	28	23	37	32	50
18	24	4	14	10	23	17	30	25	39	34	52
24	30	5	16	11	24	19	35	29	46	40	58
30	40	6	18	13	29	23	40	34	53	46	66
40	50	6	19	14	31	25	44	37	57	50	71
50	65	7	21	16	36	30	50	45	69	62	88
65	80	8	24	18	40	35	60	54	83	76	108
80	100	9	27	22	48	42	70	64	96	89	124
100	120	10	31	25	56	50	83	75	114	105	145
120	140	10	38	30	68	60	100	90	135	125	175
140	160	15	44	35	80	70	120	110	161	150	210

Table 5.5 Radial clearance of self-aligning ball bearings, tapered bore

Unit μm

C2		Normal		C3		C4		C5		Nominal bore diameter *d* mm	
min	max	min	max	min	max	min	max	min	max	over	incl.
-	-	-	-	-	-	-	-	-	-	2.5	6
-	-	-	-	-	-	-	-	-	-	6	10
-	-	-	-	-	-	-	-	-	-	10	14
-	-	-	-	-	-	-	-	-	-	14	18
7	17	13	26	20	33	28	42	37	55	18	24
9	20	15	28	23	39	33	50	44	62	24	30
12	24	19	35	29	46	40	59	52	72	30	40
14	27	22	39	33	52	45	65	58	79	40	50
18	32	27	47	41	61	56	80	73	99	50	65
23	39	35	57	50	75	69	98	91	123	65	80
29	47	42	68	62	90	84	116	109	144	80	100
35	56	50	81	75	108	100	139	130	170	100	120
40	68	60	98	90	130	120	165	155	205	120	140
45	74	65	110	100	150	140	191	180	240	140	160

Table 5.6.1 Radial internal clearance of spherical roller bearings, cylindrical bore Unit μm

Nominal bore diameter *d* mm		C2		Normal		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
14	18	10	20	20	35	35	45	45	60	60	75
18	24	10	20	20	35	35	45	45	60	60	75
24	30	15	25	25	40	40	55	55	75	75	95
30	40	15	30	30	45	45	60	60	80	80	100
40	50	20	35	35	55	55	75	75	100	100	125
50	65	20	40	40	65	65	90	90	120	120	150
65	80	30	50	50	80	80	110	110	145	145	180
80	100	35	60	60	100	100	135	135	180	180	225
100	120	40	75	75	120	120	160	160	210	210	260
120	140	50	95	95	145	145	190	190	240	240	300
140	160	60	110	110	170	170	220	220	280	280	350
160	180	65	120	120	180	180	240	240	310	310	390
180	200	70	130	130	200	200	260	260	340	340	430
200	225	80	140	140	220	220	290	290	380	380	470
225	250	90	150	150	240	240	320	320	420	420	520
250	280	100	170	170	260	260	350	350	460	460	570
280	315	110	190	190	280	280	370	370	500	500	630
315	355	120	200	200	310	310	410	410	550	550	690
355	400	130	220	220	340	340	450	450	600	600	750
400	450	140	240	240	370	370	500	500	660	660	820
450	500	140	260	260	410	410	550	550	720	720	900
500	560	150	280	280	440	440	600	600	780	780	1000
560	630	170	310	310	480	480	650	650	850	850	1100
630	710	190	350	350	530	530	700	700	920	920	1190
710	800	210	390	390	580	580	770	770	1010	1010	1300
800	900	230	430	430	650	650	860	860	1120	1120	1440
900	1000	260	480	480	710	710	930	930	1220	1220	1570
1000	1120	290	530	530	530	780	780	1020	1330	1330	1720
1120	1250	320	580	580	860	860	1120	1120	1460	1460	1870
1250	1400	350	640	640	950	950	1240	1240	1620	1620	2080

Table 5.6.2 Radial internal clearance of spherical roller bearigns, tapered bore

Unit μm

C2		Normal		C3		C4		C5		Nominal bore diameter d mm	
min	max	min	max	min	max	min	max	min	max	over	incl.
-	-	-	-	-	-	-	-	-	-	14	18
15	25	25	35	35	45	45	60	60	75	18	24
20	30	30	40	40	55	55	75	75	95	24	30
25	35	35	50	50	65	65	85	85	105	30	40
30	45	45	60	60	80	80	100	100	130	40	50
40	55	55	75	75	95	95	120	120	160	50	65
50	70	70	95	95	120	120	150	150	200	65	80
55	80	80	110	110	140	140	180	180	230	80	100
65	100	100	135	135	170	170	220	220	280	100	120
80	120	120	160	160	200	200	260	260	330	120	140
90	130	130	180	180	230	230	300	300	380	140	160
100	140	140	200	200	260	260	340	340	430	160	180
110	160	160	220	220	290	290	370	370	470	180	200
120	180	180	250	250	320	320	410	410	520	200	225
140	200	200	270	270	350	350	450	450	570	225	250
150	220	220	300	300	390	390	490	490	620	250	280
170	240	240	330	330	430	430	540	540	680	280	315
190	270	270	360	360	470	470	590	590	740	315	355
210	300	300	400	400	520	520	650	650	820	355	400
230	330	330	440	440	570	570	720	720	910	400	450
260	370	370	490	490	630	630	790	790	1000	450	500
290	410	410	540	540	680	680	870	870	1100	500	560
320	460	460	600	600	760	760	980	980	1230	560	630
350	510	510	670	670	850	850	1090	1090	1360	630	710
390	570	570	750	750	960	960	1220	1220	1500	710	800
440	640	640	840	840	1070	1070	1370	1370	1690	800	900
490	710	710	930	930	1190	1190	1520	1520	1860	900	1000
530	770	770	1030	1030	1300	1300	1670	1670	2050	1000	1120
570	830	830	1120	1120	1420	1420	1830	1830	2250	1120	1250
620	910	910	1230	1230	1560	1560	2000	2000	2470	1250	1400

*In some critical applications, it is necessary to use bearings with controlled vibration and frequency. In such case, please contact your FBJ sales or engineering department.

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6. Speed and High Temperature Suitability

6.1 Maximum Rotational speed

The permissible rotational speed is given in the Catalogue for two kinds of lubricates: grease and liquid oil. However, it does not mean that the maximum rotational speed is acceptable at any load. The ultimate factor limiting the speed is temperature which depends on friction in the bearing and on the heat removal possibility. The limiting values of rotational speed given in the Catalogue are based on the following assumptions: the operating radial clearance is sufficient to enable compensation of the difference in the linear expansion between the outer and inner rings caused by their being heated to different temperatures; the assembly uses rigid shafts and housings; the lubricant is properly selected. The amount of the maximum permissible load is determined by the temperature factor.

The maximum rotational speed found in the Catalogue can, in certain cases, be exceeded by changing the loading conditions and the lubricant. However, in this case, care must be taken to apply a strictly specified dose of properly selected lubricant and to ensure removal of heat arising from friction. A further significant increase of the above-mentioned maximum rotational speed can be made possible by improvement of the bearings design, primarily that of bearing cages, by developing better lubricants, etc. Whenever problems arise in connection with the operation of bearings at higher rotational speed, please consult our personnel.

6.2 Temperature Suitability

FBJ bearings are heat treated in such a way that they can be used at operating temperatures of up to 120˚C. Bearings with polyamide cages can be used operating temperatures of up to 100˚C only.

If bearings are designed for operation under high temperature conditions, their life expectancy gets somewhat lower because of reduced hardness and fluctuation of the impact viscosity level. In order to prevent the parts dimensions form being changed, they are additionally tempered at high temperatures exceeding the maximum operating temperatures of bearings. Such bearings carry additional marking symbols placed to the right of the location of the bearing designation. Table contains the values of the selected dynamic load-carrying capacity should be multiplied depending on the bearing operating temperature.

Table 6.1 Speed and High Temperature Suitability

Bearing Operating Temperature,˚C	Temperature coefficient
160	0.90
180	0.85
200	0.80
250	0.71
300	0.60

7. Bearing Material

7 Bearing Material

The quality of bearings is influenced, to a large measure, by the properties of the material they are made of.

Rings and rolling elements are fabricated predominantly of through-hardening carbon chromium bearing steel of high cleaniness. On delivery to the plant, all the purchased steel is tested for compliance with the basic technical specifications such as the chemical composition, contamination with non-metallic inclusions, metal structure. The examination is conducted at the plant's specialized laboratories equipped with up-to date apparatus and instruments and manned with highly qualified specialists.

The entire work on heat treatment and machining of the rings and rolling elements is carried out with the use of non-destructive testing facilities, which permits to assure high stability of the technological process.

The plant performs systematic tests of the principal types of the manufactured bearings for fatigue life and, in so doing, checks the basic dynamic load-carrying ratings found in this Catalogue and the quality of the steel used.

7.1 Rings and Balls

Standard material for Rings and Balls is vacuum degassed high carbon chromium bearing steel (SUJ2*) allowing for high efficiency, low torque, low noise level and long bearing life. Bearings require anti-corrosion properties, used stainless steel.

7.2 Cage

Cold rolled steel sheets or strips used for pressed cages and High tensile brass castings or machined steel is used for Machined cages. Polyamide material is used in moulded cages. Bearings require anti-corrosion properties used stainless steel cages.

7.3 Shield

Cold rolled steel sheets or strips is used for standard shields and bearings and require anti-corrosion properties used stainless steel.

7.4 Seal

All FBJ seals are made of molded synthetic nitrile rubber which can withstand the temperatures from -45˚C to 125˚C.

7.5 Stainless Steel

For bearings requiring anti-corrosion or heat resistance properties, rings and balls are made of martensitic stainless steel (SUS440C)** and this martensitic stainless steel is magnetic type. SUS 304 is used In FBJ Stainless steel Cage and Stainless steel Shield.

Table 7 Chemical Composition of Bearing Materials

MATERIAL	SYMBOL	CHEMICAL COMPOSITION %							Hardness HRC
		C	**Si**	**Mn**	**P**	**S**	**Cr**	**Mo**	
HIGH CARBON CHROMIUM STEEL	SUJ2* or SAE52100 or 100Cr6	0.9~1.10	0.15~0.35	≤ 0.50	≤ 0.025	≤ 0.025	1.30~1.60	0.08	58~65
COLD ROLLED STEEL	SPCC	≤ 0.12	--	≤ 0.50	≤ 0.040	≤ 0.045	--	--	--
STAINLESS STEEL	SUS440C** or AISI440C or X102CrMo17	0.9~1.20	--	--	--	--	16.0~18.0	0.75	58~65

Information

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8. Lubrication and storage

8.1 Lubrication

Bearing lubrication reduces friction and wear, acts as a coolant, minimizes contamination, prevents corrosion, and generally extends bearing life. Selecting the best lubricant for your specific application becomes a very important decision; however, choosing from the hundreds available can be an overwhelming task. FBJ's engineering staff is available to help make the right decision for your application.

8.2 Oil Lubrication

Oil is the basic lubricant for ball and roller bearings. The main adventage if an oil lubricant is that there is less bearing torque. The use of synthetic oils such as diesters, silicone polymer, and fluorinated compounds has improved volatility and viscosity characteristics and increased temperature properties.

Table 8.1 Reccomended Oils in Industrial Use

Manufacturer	Manufacturer Code	FBJ Suffix	Lubricant Base	Flash Point˚C	Visocsity (cSt)	Operating Temperature˚C
Anderson Oil Co.	Windsor Lube L-245X	OA01	Diester	215	14(38˚C)	-55+175
Dow Corning Co.	SH550R	OD01	Methylphenly	316	125(25˚C)	-40+230
Nihon Oil Co.	Antirust P21 00	ON-1	Mineral	166	13(40˚C)	-20-+115
Shell Oil Co.	Aero Shell Fluid 12	OS01	Diester	235	14(38˚C)	-50-+120
Shell Oil Co.	Aero Shell Fluid 3	OS02	Petroleum	145	10.2(40˚C)	-55+115
Thenneco Chemicals	Anderol L-40 I D	OT01	Diester	220	12.7(38˚C)	-60+125

Table 8.2 Greases used in FBJ Bearings

Manufacturer	Manufacturer Code	FBJ Suffix	Thickening agent	Lubricant Base	Drop Point˚C	Consistency	Operating Temperature˚C Range (˚C)
Caltex	Chevron SRI-2	GC01	Urea	Mineral	240	270	-30~+175
Dow Corning	Molykote 33M	GD01	Lithium	Silicone	210	260	-70~+180
	Molykote 44M	GD02	Lithium	Silicone	204	260	-40~+200
	Molykote FS 1292	GD03	Fluorotelomer	Phlorosilicone	232	310	-40~+200
	Molykote FS 3451	GD04	Fluorotelomer	Phlorosilicone	260	285	-40~+230
Esso	Andok B	GB01	Sodium	Mineral	260	285	-40~+120
	Andok C	GB02	Sodium	Mineral	260	205	-20~+120
	Andok 260	GB03	Sodium	Mineral	200	260	-30~+150
	Beacon 325	GB04	Lithium	Diester	193	280	-60~+120
Kyodo Yushi	Multemp PS2	GK01	Lithium	Diester.Mineral	190	275	-55~+130
	Multemp SRL	GK02*	Lithium	Ester	191	245	-40~+150
Nihon Oil	Multinocurea	GM01	Urea	Mineral	260	290	-20~+175
Shell Oil	Alvania No.2	GS01*	Lithium	Mineral	182	272	-25~+120
	Alvania No.3	GS02	Lithium	Mineral	1S3	233	-20~+135
	Alvania RA	GS03	Lithium	Mineral	183	252	-40~+130
	Aero Shell Grease No.7	GS01*	Microgel	Diester	260	288	-73~+149
	Aero Shell Grease No.15A	GS05	Fluorotelomer	Silicone	260	280	-73~+260
Shinetsu Silicone	Silicolube G40M	GS31	Lithium	Silicone	210	260	-30~+200

*These suffixes may not indicated in the bearing or bearing box when numbering.

8.3 Grease Lubrication

For lubrication of rolling bearings, use is mainly made of grease, because the techniques of their employment are more simple, they do not require complicated sealing devices and demand less expenditures for the maintenance of mechanisms. When a machine or a mechanism is stopped, grease does not run off from the bearing but remains there and even seals the assembly isolating it from the surroundings. These and other advantages of greases are so decisive that allow the wear of bearings to be ignored. The use of grease brings about a more rapid wear than when operating with oils due to the accumulation of abrasive particles in the former.

Greases are obtained by solidifying lubricating oils with the aid of various thickening agents. Such silidification agent creates a structural framework of interwoven fibers which imparts plasticity to the lubricating material and retains lubricating oil in its cells.

Grease is well held in place in a bearing, does not flow out under the effect of the force of gravity and resists the action of centrifugal forces attempting to throw lubricant away from the bearing during its rotation. The properties of grease are determined by the composition of the thickening agent.

For rolling bearing lubrication, use is normally made of grease in which mineral oil is solidified with the aid of sodium, calcium or lithium soaps.

Rolling bearings should be filled with grease just immediately before the unit is to be assembled. The decisive reason for this is very stringent requirements to the lubricant purity. The later the lubricant is put in, the lesser the danger of its getting contaminated.

The bearing type or design features of a unit may demand it to be filled with grease at a later stage.

Thus, for instance, if it is necessary to adjust the amount of clearance in bearings with a tapered inner bore, the required measurements can be only performed before the unit is filled with grease. It is also impracticable to put in grease before the bearing is heated for mounting. Preliminary packing a bearing with grease is only recommended when it is impossible to distribute grease over rolling elements and raceways after assembly.

Normally, a bearing as a whole and the free space in the unit housing is only partially filled with grease-from 30 to 50%. However, when using lithium-base lubricants for supports that are not subjected to strong vibration, the free space of the housings can be filled up to 90% disregarding the danger of overheating. When a support is filled with a larger than normal amount of grease, this improves the reliability of protection against contamination and prolongs the support's service life.

High-speed rolling bearings, for example, spindle units of metal-cutting machine tools mush be lubricated with a small amount of grease in order to limit the temperature of unit heating. In supports subjected to strong vibration, for example, in the hubs of motor car wheels and in the boxes of railroad car wheels, as well as in vibration machines, grease should fill not more than 60% of the free space.

The technique of packing a bearing unit with grease is selected depending on the bearing type.

Separable bearings (cylindrical, tapered, thrust-type) are filled with grease following the sequence of assembly, applying a thin layer on the raceway of the installed ring and then filling the space between the rolling elements.

In inseparable bearings, for example, in radial and angular contact bearings grease should be stuffed in form both ends. Self-aligning ball bearings and spherical roller bearings can be filled with grease by turning the ring and stuffing the lubricant in between the rolling elements.

8.4 Solid Lubrication

A solid film lubrication can range from simple sacrifical retainers, graphite, or molybdenum disulphide (MoS_2) powders, to complexion sputtering or plating. Each type must be ehgineered for the specific application. They are very useful in areas of temperature extremes, vacuum, radiation, pressure, or harsh environments where coventional lubricants would fail. Solid film lubricants do not deteriorate in storage.

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8.5 Storage Of Bearings

Rolling bearings have high-quality working surfaces. Any deterioration of the surface quality results in a premature wear and reduction in the service life of bearings.

Bearings are made predominantly of ferrous metals, therefore, the main danger for them is corrosion which is absolutely intolerable on the working surfaces of bearings. To prevent in-storage corrosion, bearings are delivered to the customer in a preserved state, i.e., washed to remove dirt, contamination, slushed with corrosion-protective lubricant - mineral oil with an inhibitor, and packed in special packing.

The time this lubricant will be capable of protecting the bearing against corrosion depends on storage conditions. The customer's task is to store bearings in conformity to the Manufacturer's instructions.

The occurrence of corrosion of bearings during storage depends on two main factors:

1) relative air humidity in the storage place: the lower the humidity, the weaker the process of corrosion. No in-storage corrosion is practically observedc when relative humidity is below 40%

2) temperature gradient in the storage premises during the day. The smaller the temperature difference, the more favorable the storage conditions. Great temperature fluctuations are particularly dangerous when there is a high relative humidity. In this case moisture can condense on the surfaces of bearings, increasing sharply the probability of corrosion. These factors need to be considered when establishing requirements to bearing storage premises.

A room used for storage of bearings must be dry, heated, well ventilated, located far from places where the air contains trings of substances that cause metal corrosion-away from chemical, pickling, galvanic shops. The storage room air temperature must be kept, as far as possible, within 10˚C to 30˚C. The daily temperature variation should not exceed 5˚C.

Relative air humidity in the storage room should not be in excess of 60%. It is desirable that it should be as low as possible. The bearing storage conditions in the room (humidity and temperature) should be continuously monitored.

It is recommended to store large-size bearings with an inner diameter over 200mm placed on their end-faces to avoid possible deformation of the thin-walled rings.

9. Mounting And Dismounting

9.1 Mounting

Ball and roller bearings should be mounted by qualified personnel paying special attention to keeping them clean: this is very important for ensuring satisfactory operation of bearings and preventing their premature breakdown.

9.2 Preparation of Mounting

The mounting should be preferably done in a room with dry clean air located far from the sources of dust, emulsion, dirt. The shaft and housing surfaces mating with the bearings should be thoroughly washed with gasoline or kerosene, wiped, dried and coated with a thin layer of lubricant. Care must be taken to check the accuracy of dimensions and shapes of all the parts mating with the bearing; they should not exceed the dimensions.

The manufacturer's packing is to be removed from the bearings immediately before mounting to prevent penetration of dirt. Preservative coating is removed from the mounting surfaces only. The mounting surfaces are to be washed with gasoline or kerosene and wiped dry with clean nap-free cloth. If a bearing is dirty or its packing is damaged, it should be thoroughly washed prior to mounting. The mounting surfaces in this case are to lubricated with medium-viscosity oil.

Prior to mounting, check the bearing appearance, marking, ease of rotation, clearances for compliance with the requirements of the technical documents and this Catalogue.

Radial clearance in spherical roller bearings are measured with the aid of a set of feeler gauges, or by other methods. Feeler gauges are used to measure clearances between the outer ring and the unloaded roller. Prior to mounting, bearings, especially those with the ratio of the length and the largest shaft diameter exceeding 8, must be tested for straightness (absence of bending).

9.3 Mounting of bearings

The method of bearings mounting (mechanical, hydraulic, thermal) depends on their type and size. In all the cases, it is very important to protect rings, bearing cages, rolling elements against direct knocks, because they can damage the bearings. The principal rule to be observed when mounting a bearing is never to allow the compressing force to be transferred through rolling elements.

When mounting a bearing, it is necessary to ensure the required precision of the bearing rings location with respect to the rotation axis which mainly depends on the absence of misalignment. Misalignments of rings is one of the factors causing initial damages of bearings and concentration of contact stresses. The operating misalignment of the rings should not exceed 0.7 maximum design-permissible angle of alignement of the bearing rings under normal operating conditions (this parameter is to be taken from the description of bearing groups).

It is should be borne in mind that the outer ring of a spherical radial bearing has the property of readily swivel out. To replace the ring in its original position, it is necessary to set the dislocated (braking) rollers, with the aid of fingers, back into the outer ring and restore the latter to its original position. NEVER knock on the ring or rollers with a hammer.

9.4 Bearings with Cylindrical Bore

When mounting inseparable bearings, usually the ring with a tighter fit is to be mounted the first. If pre-loading in the tight fit is not too high, small-size bearings can be mounted by knocking lightly with a hammer on

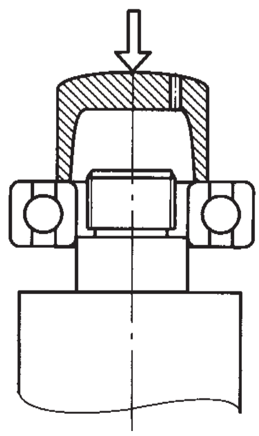

Figure 9.1

the sleeve installed on the front end or the bearings ring.

Knocks should be uniformly distributed over the circumference to prevent the bearing from misalignment. When an aligning bar is used instead of the sleeve, the force must be applied at the center

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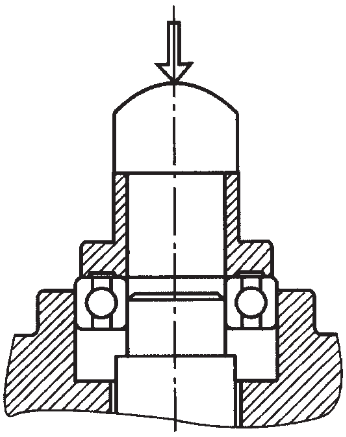


Figure 9.2

(see Figure 9.1). If the bearing is inseparable, it should be simultaneously pressure-fitted onto the shaft and into the housing seat (see Figure 9.2) with the aid of the mounting tool shown in the sketch, a mounting ring is inserted between the bearing and aligning bar, resting on the front ends of the inner and outer rings. The supporting surfaces of the mounting ring should lie in the same plane to ensure even distribution of the forces applied to both Rings during the mounting procedure. When mounting self-

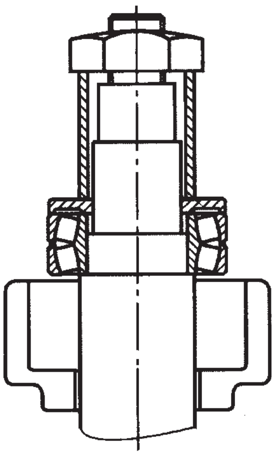


Figure 9.3

aligning bearings, for example, spherical roller bearings, the use of an intermediate mounting rings permits to prevent misalignment and turning of the outer ring after the bearing with the shaft has been installed in the housing seat (see Figure 9.3).

The intermediate mounting ring must have a groove to keep it form touching the rolling elements or bearing cages. Bearings having a diameter of up to 100 mm can be pressure-fitted onto the shaft in cold state with the use of mechanical or hydraulic presses.

The inner ring of a separable bearing can be mounted independently of the outer ring. When

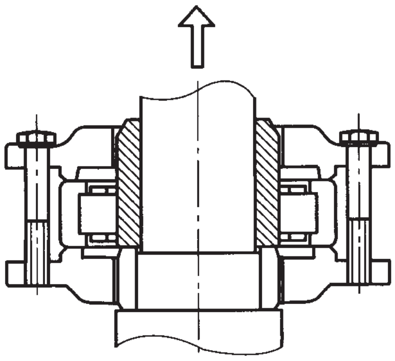
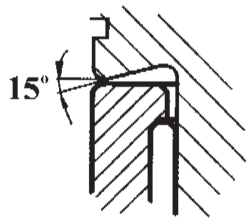


Figure 9.4

mounting a shaft already carrying the inner ring into the housing with the outer ring care should be taken to make them properly centered, otherwise the raceways, balls or rollers can get scored. That is why, when mounting bearings with needle and cylindrical



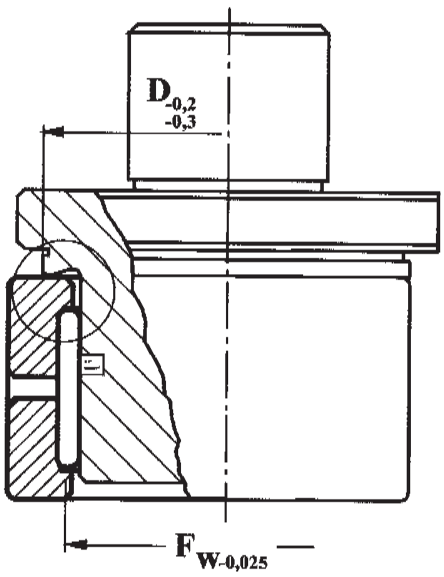


Figure 9.5

rollers, it is recommended to use a mounting sleeve (see Figure 9.4). The outer diameter of the sleeve must be equal to the raceway diameter F of the inner ring machined with a d10 accuracy. The values of F are given in the bearings table. Needle roller bearings with a die-stamped outer ring best mounted with the aid of a special aligning bar (see Figure 9.5).

Large-size bearings or those with tight fit should be heated before mounting. NEVER pre-heat bearings in excess of 120˚C, for this may cause changes of both the bearing material, as well as possible burning or deformation of polya-mide bearing cages. DO NOT pre-heat bearings having protective shields or seals, because they are filled with grease.

When pre-heating bearings, care should to be taken to avoid local overheating. Uniform safe Heating can be achieved with the aid of electric heaters, heating furnaces and oil bath.

It is also recommended to use special electric induction heaters. Here the bearing (ring) is heated by an alternating magnetic field which gives rise to eddy currents. After induction heating the bearings (rings) need to be demagnetized.

9.5 Bearings with Tapered Bore

Inner rings of tapered-bore bearings are always mounted with a tight fit. The amount of interference in this case is determined not by the shaft size tolerances as for cylindrical-bore bearings, but by shifting the bearings along the conical surface of the shaft mounting journal, of adapter or withdrawal sleeve.

Double-row spherical roller bearings with tapered bore are mounted onto cylindrical shaft with the aid of adapter or withdrawal sleeves, while on tapered-journal shafts they are installed directly on the shaft. Before mounting, the washed bearing bore and the sleeve may be covered with a thin coat of lubricant. A thicker lubricant layer will reduce friction and, in so doing, facilitate mounting, but in the course of operation the lubricant will be pressed out from the mounting joints. As a result, the fit will lose tightness and the ring or the sleeve will run wearing out the mounting surfaces.

It is a good practice to mount bearings with the bore of up to 70 mm and normal tightness using a hammer and a mounting sleeve screwed onto the threaded shaft end. The pressure part acts on the adapter sleeve end or directly on the inner ring end-face (when mounting is carried out without adapter and withdrawal sleeves). Bearings with a diameter exceeding 100 mm should be mounted using a hydraulic formed (expanded) along with the axial shift of the adapter sleeve.

When any previously dismounted bearing is to be mounted again, it is not sufficient to restore the lock nut to its original position, because after a prolonged operation the radial clearance fit loose due to the wear of the thread and smoothing of the mounting seats, the shift gets longer. For self-aligning spherical roller bearings the values of teh decrease in teh original radial clearance which are neccessary to ensure a tight static fit. The radial clearance of spherical roller bearings is measured with the use of feelre gauges in both roller rows simultaneously. It is necessary to observe that the rollers are pressed against the meddle flange (a guiding lip). Yhe outer and inner rigns should be located so as to ensure equal radial calearance for both rows.

The method of mounting bearings is selected based on teh mounting conditions.

Small-and medium-size bearings can be fitted onto the mounting seats with the aid of the lock nut. The nut is tighteded using a box wrench (see Figure 9.6)

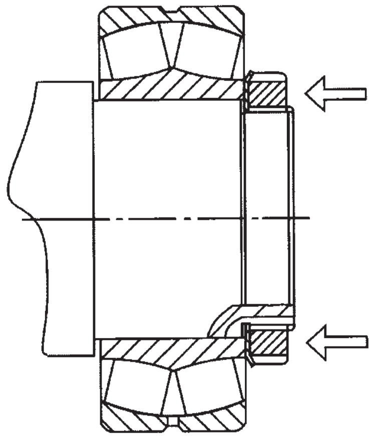


Figure9.6

Small-size bearings with an adapter sleeve are mounted onto the tapered surface of the adapter sleeve with the aid of teh lock nut (see Figure 9.7).

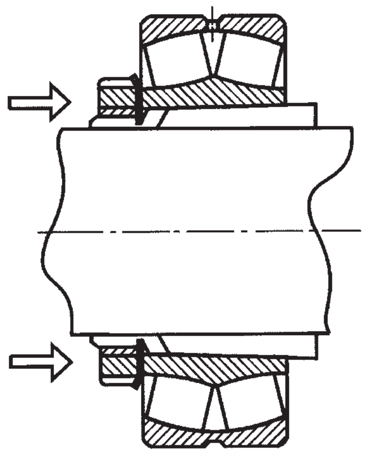


Figure 9.7

Small-size withdrawal sleeves are pressure-fitted with the aid of the lock nut into the gap between the shaft and the inner ring (see Figure 9.8).

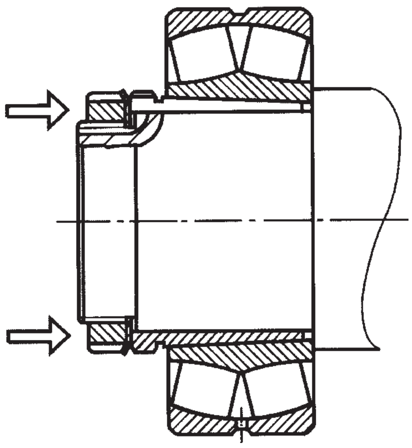


Figure 9.8

The nuts of larger-size bearings require a greater tightening force. In these cases mounting can be made easier with the use of a mut with thrust bolts shown in Figure 9.9. To prevent the bearing or sleeve from being wedged, it is necessary to screw up the

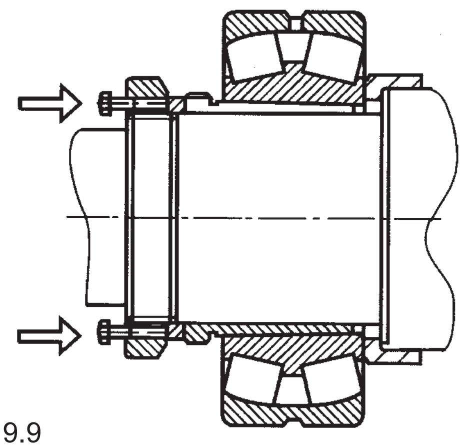


Figure 9.9

nut preliminarily unitl it comes fully against the mounting sleeve. The thrust bolts made of improved steel and located evenly along the circumference (the number of bolts depends on the force required) are screwed in uniformly in a cross manner until the necessary decrease of radial clearance is obtained. Since a tapered mounting surface provides self-braking, the accessory may be, then, removed and the bearing can be fastened tight with it own fastening nut. This principle is applicable for bearings mounted on a sleeve or directly on a tapered journal.

When mounting large-size bearings, a hydraulic accessory, for example, a circular piston pump, is normally usrd, to mount a bearing or to press-fit a sleeve (see Figure 9.10). The ring can be shifted axially with the aid of a screw - or hydraulically-operated nut (for large-sizebearings). A hydraulically driven nut has

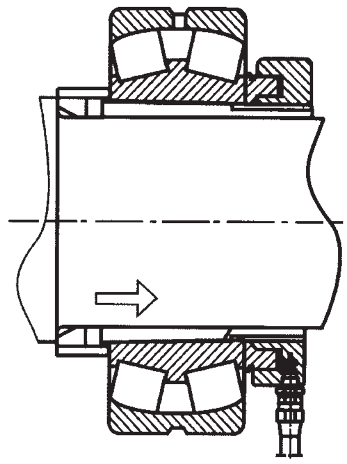


Figure 9.10

a cylindrical groove on one of the end which serves for insertion of a round piston provided with an O-ring seal. The nut is connected, by means of a hose, with a pump feeding oil to the nut. the pump is a jet-type oil pump with a flexible high pressure hose. The nut piston is moved by oil pressure, then it is extended and pressure-fits the bearing onto the mounting seat.

The most expedient method of mounting large-size bearings (with the bore diameter of over 300-mm) is the use of a hydraulic outward thrust which affords high - quality mounting of a bearing. For this purpose, special channels and grooves are made on the shaft to enadle oil to be fed to under the bearing inner ring. When employing hydraulically-aided mounting, pump-driven oil is supplied via the oil-conducting channels and grooves to the contact zone of the bearing inner ring and the shaft. The pressurized oil fed to the contact zone of the rings and the shaft thrusts the ring outwards, thus permitting axial displacement of the ring along the shaft (see Figure 9.11).

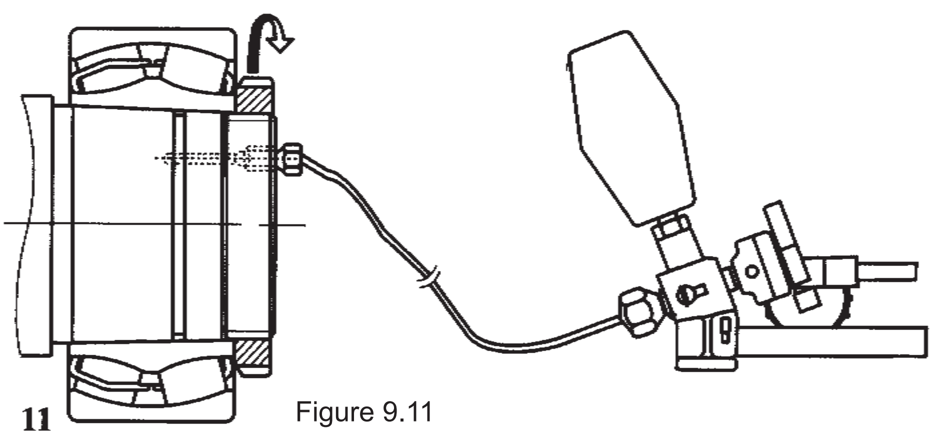


Figure 9.11

Table 9.1 Reduction in Radial Clearance (Gap) Depending on Axial Displacement a Tapered Shaft or Sleeve (Reference)

Bearing bore nominal size, d, mm		Reduction in radial clearance*, mm		Minimum permissible residual clearance ** after mounting of bearing with initial clearance, mm		
Over	Up to	min	max	Normal	Group 3	Group 4
24	30	0.015	0.02	0.015	0.020	0.035
30	40	0.020	0.025	0.015	0.025	0.040
40	50	0.025	0.03	0.020	0.030	0.050
50	65	0.030	0.04	0.025	0.035	0.055
65	80	0.040	0.05	0.025	0.040	0.070
80	100	0.045	0.06	0.035	0.050	0.080
100	120	0.050	0.07	0.050	0.065	0.100
120	140	0.065	0.09	0.055	0.08	0.11
140	160	0.075	0.10	0.055	0.09	0.13
160	180	0.080	0.11	0.06	0.10	0.15
180	200	0.090	0.13	0.07	0.10	0.16
200	225	0.100	0.14	0.08	0.12	0.18
225	250	0.110	0.15	0.09	0.13	0.20
250	280	0.120	0.17	0.10	0.14	0.22
280	315	0.130	0.19	0.11	0.15	0.24
315	355	0.150	0.21	0.12	0.17	0.26
355	400	0.170	0.23	0.13	0.19	0.29
400	450	0.200	0.26	0.13	0.20	0.31
450	500	0.210	0.28	0.16	0.23	0.35
500	560	0.240	0.32	0.17	0.25	0.36
560	630	2.260	0.35	0.20	0.29	0.41
630	710	0.300	0.40	0.21	0.31	0.45
710	800	0.340	0.45	0.23	0.35	0.51
800	900	0.370	0.50	0.27	0.39	0.57
900	1000	0.410	0.55	0.30	0.43	0.64
1000	1120	0.450	0.60	0.32	0.48	0.70
1120	1250	0.490	0.65	0.34	0.54	0.77
1250	1400	0.550	0.72	0.36	0.59	0.84

*Valid for solid steel shafts and hollow shafts with bore diameter of up to half diameter of the shaft, only.

** Bearings with the radial clearance in the upper-half o ftolerance limit, prior to mounting, shall be mounted with provision of reduced radial clearance or axial shift at an upper limit; bearings kwith the radial clearance in the lower half of tolerance limite-with reduced radial clearance or axial shift at lower value.

Information

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Axial displacement*. mm							
1:12 Taper				1:30 Taper			
Shaft		Sleeve		Shaft		Sleeve	
min	max	min	max	min	max	min	max
0.30	0.35	0.30	0.40	-	-	-	-
0.35	0.40	0.35	0.45	-	-	-	-
0.40	0.45	0.45	0.50	-	-	-	-
0.45	0.60	0.50	0.70	-	-	-	-
0.6	0.75	0.70	0.85	-	-	-	-
0.7	0.9	0.75	1.0	1.7	2.2	1.8	2.4
0.7	1.1	0.8	1.2	1.9	2.7	2.0	2.8
1.1	1.4	1.2	1.5	2.7	3.5	2.8	3.6
1.2	1.5	1.3	1.7	3.0	4.0	3.1	4.2
1.3	1.7	1.4	1.9	3.2	4.2	3.3	4.6
1.4	2.0	1.5	2.2	3.5	4.5	3.6	5.0
1.6	2.2	1.7	2.4	4.0	5.5	4.2	5.7
1.7	2.4	1.8	2.6	4.2	6.0	4.6	6.2
1.9	2.6	2.0	2.9	4.7	6.7	4.8	6.9
2.0	3.0	2.2	3.2	5.0	7.5	5.2	7.7
2.4	3.4	2.6	3.6	6.0	8.2	6.2	8.4
2.6	3.6	2.9	3.9	6.5	9.0	6.8	9.2
3.1	4.1	3.4	4.4	7.7	10.0	8.0	10.4
3.3	4.4	3.6	4.8	8.2	11.0	8.4	11.2
3.7	5.0	4.1	5.4	9.2	12.5	9.6	12.8
4.0	5.4	4.4	5.9	10.0	13.5	10.4	14
4.6	6.2	5.1	6.8	11.5	15.5	12.0	16
5.3	7.0	5.8	7.6	13.3	17.5	13.6	18
5.7	7.8	6.3	8.5	14.3	19.5	14.8	20
6.3	8.5	7.0	9.4	15.8	21	16.4	22
6.8	9.0	7.6	10.2	17.0	23	18.0	24
7.4	9.8	8.3	11.0	18.5	25	19.6	26
8.3	10.8	9.3	12.1	21.0	27	22.2	28.3

When mounting a bearing on a tapered sleeve, hydraulic fluid can be supplied through the channels located in the sleeve itself.

When mounting a bearing into the housing with a tight fit, it is recommended, before mounting, either to pre-cool the bearing (with liquid nitrogen or dry ice) or to preheat the housing.

When mounting bearings, especia those that are subjected to axial loads, it is advisable whenever possible to make sure, with the use of a feelre gauge or a light slit, that the bearing ring end-faces abut properly and tightly (without misalignment) to the shoulder ends. A similar check should be made on the opposite bearing ends and the ends of the parts pressing them in the axial direction.

It is necessary to check the correctness of the mutual loaciton of bearings in the supports of one shaft. When the supports of one shaft are installed in different split housings, they should be checked, after installation of the housing, for correctness of their mutual position, i.e.; they must be accurately in line with each other. After mounting, the shaft must be easily started by hand and rotate freely and evenly.

9.6 Running Tests

After the bearing has been mounted and checked for ease of rotation, the unit is filled with a prescribed type of lubricant and subjected to running tests aimed at checking the noise level created by the runnung bearing and the working temperature.

The running test should be performed under partial loading at low and medium rotational speeds. NEVER can bearings, especially thrust-type and angular contact thrust bearings, be tested under no-load conditions, nor be accelerated immediately to high speeds, because in this case balls and rollers will slip over raceway and damage it, or excessive stresses may arise in the bearing cage, Noise credted by the bearing retation should be checked with the use a stethoscope, tube or hollow rod. Properly mounted and well lubricated bearings produce a soft, slightly buzzing noise in their operation.

The ocurrence of a shrill noise may be the evidence of improper mounting, misalignment, damage form the use of hammer; non-uniform noise or knocking reveals teh presence of foreign particles in teh bearing; a metallic sound is indicative of an insufficient clearance in the bearing; a whishtling or gritting sound points to insufficient lubrication.

A rise of bearing temperature immediately after starting is a normal event, with time temperature gets stabilized. Abnormally high temperatures or persisten temperature variations point to an excessive amount of lubricant in the unit, an unduly tight fit of the bearing in teh radial or axial direction, an improper workmanship of teh mating parts which causes catching of the bearing cage or rolling elements, s stronger friction of seals, or mutual tiltness of the rings. Make sure to check the quality of seales and operation of the lubricating equipment during the running tests. The running test process can be considered completed only after stabilization of the bearing temperature conditions.

9.7 Dismounting

Bearing dismounting should be made without damage of bearings of bearings and mating parts. If bearings are to be used again after the machine has been disassembled, the dismounting effort shall not be transmitted through rolling elements. With separable bearings, one ring, together with the rolling elements and the bearing cage, can be removed independently of the other ring. Dismantling of non-separable bearings should begin with the removal of a more loosely fitted ring.

9.8 Bearings with Cylindrical Bore

Small-size bearings can be removed from the shaft by lightly knocking with a hammer on the aligning bar made from light metals, shifting the bar over teh bearing ring circumference. Larger-size bearings are normally dismantled with the use of various extractor: mechanical screw-type and hydraulically-driven

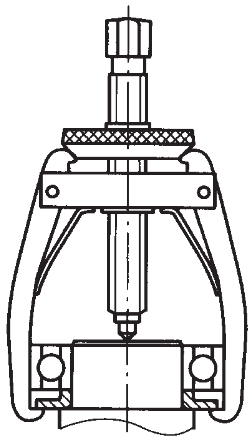

Figure 9.12

removers (see Figure 9.12). the remover rods are pressed directly to the face of the ring to be removed or to the adjacent part. Use may be made of removers carrying stripping rings or half-rings, as well as of three-rod screw removers.

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To facilitate future dismantling, designers should make provision for slots in the shaft or housing shoulders permitting insertion of the extraction tools, or for insertion of withdrawal ringshoulders.

The outer ring will be more readily removed from housings if teh latter have threaded holes for driving in thrust screws.

The force applied to remove a bearing is generally much greater than that necessary for pressure-fitting, as teh ring sets down with time or fretting can occur, i.e., corrosion (rust from friction) and microseizure of the ring and shaft metal.

Large-size bearings mounted with a tight fit usually require great effort for removal. The use of an oil-pressure fitting method (Supply of oil under pressure to teh mounting surface) will substantially facilitate the dismantling procedure. Of course, oil channels and distribution grooves necessary for this purpose should be provided for at the stage of the bearing assembly design.

9.9 Bearings with Tapered Bore

Dismantling of bearings located on an adapter sleeve starts with loosing the lock nut and screwing it out a few turns. Then a special intermediate part-a knock-out bar and a hammer are used to loosen the fit between the sleeve and the bearing (see Figure 9.13). When a press is used. the adapter sleeve or

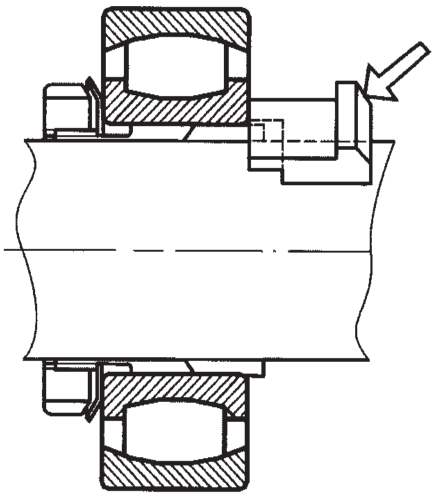

Figure 9.13

the loosended nut should be supported and the bearing should be pressed off from the adapter sleeve.

Dismounting of withdrawal sleeve of mounted bearings beings with the removeal of teh axial locking elements (the shaft nut, thrust washer, end cover, and

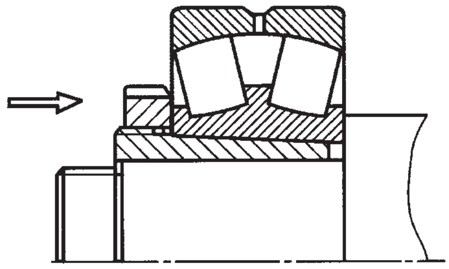

Figure 9.14

the like), then a withdrawal nut is screwed onto the sleeve thread until the sleeve fit in the bearing ring gets loose (see Figure 9.14). If the threaded portion of the sleeve goes beyong the shaft journal, a supporting ring should be inserted into the sleeve bore to protect the thread from damage when the nut is being screwed on. Ind difficult cases, especially when dismantling large-size bearings, use can be made of extraction

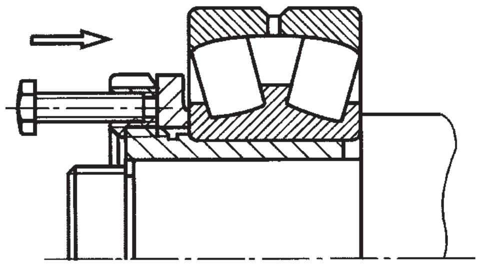

Figure 9.15

nuts with additonal thrust bolts (see Figure 9.15).

A washer is inserted between the inner ring and the thrust bolts.

If a bearing is abutted on teh lock ring, the simplest way of dismantling withdrawal sleeves is to remove them with the aid of a circular poston pump (see Figure 9.16).

The most simple and reliable technique of dismantling bearings fitted on a tapered shaft journal or those installed with teh aid of tapered sleeve, is to remove them using dydraulically-driven nuts or by means of an oil-pressure fillting method, i.e., by supplying oil to the contact zone of the inner ring and the shaft (see Figure 9.17, 9.18). When oil is fed under high pressure, the tight fit repidly gets looser and teh bearing is readily removed from the shaft journal.

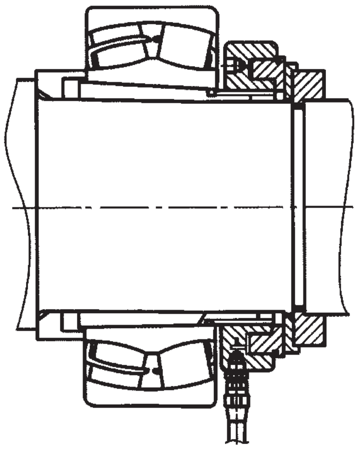

Figure 9.16

The hydraulic method is acceptable for both cylindrical and tapered fits, In both cases the shaft must be provided with oil grooves, supply channels and connecting threads. Large-size adapter and withdrawal sleeves must possess corresponding grooves and holes.

it should be borne in mind that when oil is forced in between tapered mounting surfaces, teh pressure joint is immediately released. To prevent accidents while dismounting, it is necessary to limit the axial motion (shift) of teh bearing or withdrawal sleeve with teh aid of teh lock nut, fastening sleeve nut or with a stop.

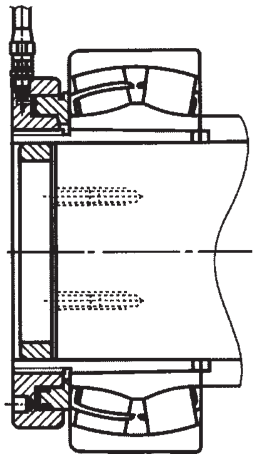

Figure 9.17

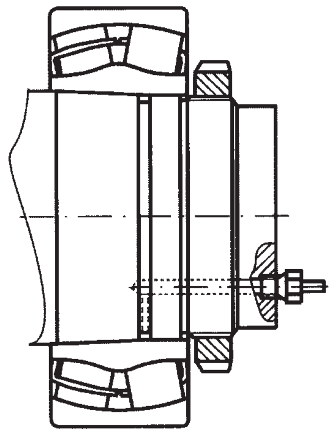

Figure 9.18

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10. Prefixes and Suffixes

Prefixes and sufixes are used to identify components, designs or variants of bearings. Moreover this will help to avoid confusion with other bearing designations. Most commonly used prefixes and suffixes are listed below.

PREFIX	SUFFIX	EXPLANATION
AN		Lock Nut
ANL		Lock Nut
AW		Lock Washer
AWL		Lock Washer
	B	Contact angle 40º for Angular contact ball bearing
	CA	Brass cage with symmetrical rollers
	CC	Flangless inner ring, symmetrical rollers, loose guide ring, window type steel cage with enhanced roller guidance
	CN	Radial Clearance Standard
	C3	Radial Clearance Larger than CN
	DU	Bush type plain bearing
	E	Controlled clearance specially for electrical motors
F		Metric series miniature ball bearing with Flange
G		Groove and Lubrication holes in the outer ring
	FDU	Bush type plain bearing with Flange
H		Adapter sleeve for mm size shaft
HA		Adapter sleeve for inch size shaft
HE		Adapter sleeve for inch size shaft
HS		Adapter sleeve for inch size shaft
K		Needle roller cage
	K	Tapered bearing Bore, taper 1:12
	K30	Tapered bearing Bore, taper 1:30
	M	Machined Brass Cage
MB		Bush type plain bearing of mm dimension
	MB	Two piece brass cage with symmetrical rollers
	/++MM	Special mm size Bore Diameter (Eg 6202ZZ/17MM, ID=17mm)
MR		Metric series miniature ball bearing Special Dimensions
MF		Metric series miniature ball bearing Special Dimensions with Flange
	N	Groove on outer ring
	NR	Snap ring on outer ring groove
	PP	Both side Rubber seal
	P0	Standard Tolerance Class
	P4	Tolerance is higher than P5
	P5	Tolerance is higher than P6
	P6	Tolerance is higher than P0
	2RS	Both side Rubber seal
R		Bearing ring with rolling element and cage assembly of separable roller bearing
R		Inch series miniature ball bearing
77R		Inch series ball bearing with both side metal shield
99R		Inch series ball bearing with both side rubber seal
SS		Stainless steel bearing
	V2Z2	Controlled Clearance, Vibration and Frequency: Specially for electrical motors
	V3Z3	Higher controlled Clearance, Vibration and Frequency than V2Z2: Specially for electrical motors
	W33	Groove and lubrication holes on outer ring
	X	Inner Diameter is different from the standard dimension for Miniature Bearings
	Y++	Bearing thickness is different from standard (Eg. 6205Y14, Special thickness of 14mm)
	ZZ	Both side Metal shield
		Grease Suffixes, please look for "Lubrication and Storage" section, Page 33