

Bearing Selection

The following procedure gives the steps to be followed when bearings are selected from the information contained in this catalogue. It will be found satisfactory for most applications.

1. a. Determine the speed of the bearing
b. Calculate the loads on the bearings.
2. Establish if accurate alignment can be obtained between the bearing seatings. If it cannot, then bearings that accommodate misalignment should be selected.
3. If the bearings rotate under load decide the life required, calculate the required dynamic load rating 'Cr' values, and then select suitable bearings that have comparable 'Cr' values.
If the bearings do not rotate under load selected then use the static load rating 'Co'.
4. Check if the bearings are suitable for the speed and decide if grease or oil is to be the lubricant.
5. Refer to shaft and housing tolerance in technical section to ensure required bearing seating fits.
6. Finally:
 - Decide if 'Standard' or 'Extra Precision' limits of accuracy are required.
 - Select the most suitable range of radial clearance.
 - Choose the abutment diameters.
 - Choose suitable closures.
 - Issue mounting and handling instructions for the bearings if necessary.

SELECTING BEARING TYPE

Each type of bearing has different properties making it suitable for certain applications. The factors to be considered when choosing a bearing are numerous so guidance is given to the main points when selecting a bearing. Special consideration must be given to aspects relating to the running and operating. Consideration must also be given to the aspects relating to at least one of the principal dimensions of the bearing as determined by the machine design or shaft size.

LOAD AND DIRECTION OF LOAD

The magnitude and direction of the external loads along with built in factors of safety are two of the main points which determine the bearing size, and in some instances the bearing type to be used. The important factors are the speed of rotation, temperature, the amount of precision required, mounting conditions and running noise.

The following illustrations indicate the magnitude and direction of the external loads the bearings can provide.

RADIAL LOADS

For light and medium radial loads ball bearings are generally used. Whereas for heavy loads and large shaft diameters, roller bearings are often the only choice.

Cylindrical roller bearings are available in several types. Types NU (with outer ring ribs) and N (with inner ring ribs) are only suitable for radial loads, whereas the NUP, NJ and NJ with angle ring HJ can be used to take certain combined loads.

THRUST LOADS

Thrust ball bearings are only suitable for light or medium purely axial loads. Double-acting thrust ball bearings can carry thrust loads in either direction. Spherical roller thrust bearings are used where heavy thrust loads are to be absorbed and in addition can carry a certain amount of radial load acting simultaneously.

COMBINED LOADS

If a radial and thrust load act on a bearing simultaneously this is termed as a 'Combined Load'. The most important feature affecting the ability of carry axial loads is the angle of contact in relation to the shaft axis. The greater the angle, the more suitable the bearing is to accommodate axial loading. Combined loads are carried by deep groove ball bearings, self-aligning ball bearings single and double row angular contact ball bearings, spherical roller bearings, cylindrical roller bearing of the locating types and taper roller bearings.

LIMITING SPEED

The speeds at which bearings can rotate are limited by the bearing type, the operating load and the permissible operating temperature of the lubricant.

Bearings with low frictional resistance and correspondingly low internal heat generation are most suitable for high speeds with proper attention being given to the correct bearing clearance after mounting.

For radial loads, the bearings most suitable are deep groove ball bearings or cylindrical roller bearings. For combined loads angular contact bearings should be selected.

MISALIGNMENT

Self-aligning ball bearings, spherical roller bearings and spherical roller thrust bearings allow, at assembly, for the correction of misalignment where the shaft can be misaligned relative to the housing. Values for permissible angular misalignment are listed in the tables which precede the bearing sizes of those particular types.

LOW NOISE LEVEL

Even though the running noise of roller bearings can blend in with the background noise of other moving parts. It is sometimes of prime importance to reduce this to a minimum level for electric motors used for example in lifts for hospitals and hotels and other domestic appliances. Such applications usually demand the fitting of a deep-groove ball bearing selected for low noise level.

RIGIDITY

This is sometimes a very important requirement, especially on machine tool spindles, where rigidity controls the bearing selection. In applications of this nature single or double row cylindrical roller bearing, or taper roller bearings are best suited, compared with the point contact of ball bearings. The stiffness can be further enhanced by pre-loading.



Bearing Selection

AXIAL MOVEMENT

In a normal bearing arrangement supporting a shaft it is usual to locate one bearing (fixed) and allow the non-locating bearing (free) to float in the housing thus preventing axial pre-load as a result of thermal expansion of the shaft. Axial movement produced by thermal expansion can be accommodated by the use of a cylindrical roller bearing of the N or NU pattern. This allows axial movement to occur by displacement of the rollers over the track.

TAPERED BORE AND SLEEVE MOUNTING

Tapered bore bearings are used for easier mounting and adjustments of the radial clearance. It is usual to fit sleeve bearings on a bright drawn steel bar thus cutting machining costs and easing assembly. Withdrawal sleeves are used to ease the removal of the bearing. The residual clearance should be checked with the tables relating to the axial drive-up for spherical roller bearings, for that particular bearing size.

PRECISION

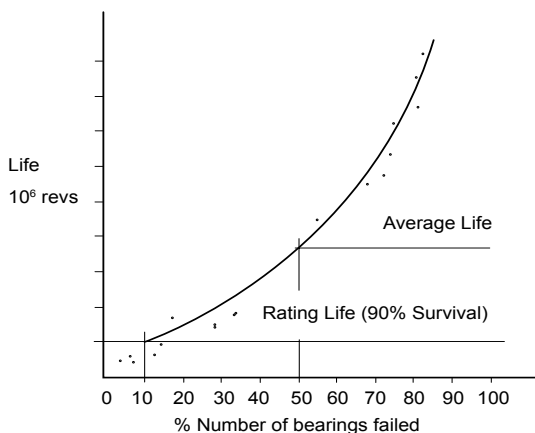
Rolling bearings with a higher degree of precision than normal are required for shafts where running accuracy is of prime importance eg, machine tools spindles and shafts rotating at very high speeds (see section relating to bearing tolerances).

DETERMINATION OF ROLLING BEARING SIZE

To determine the size of the bearing, static and dynamic load conditions and design life requirements must be considered. The load ratings for the size and type are given in the bearing tables on the appropriate pages.

DYNAMIC LOADING

When a batch of apparently identical bearings is tested under identical load, speed and operating conditions, a wide difference is obtained in the lives of the bearings. Typical results are plotted on the graph; this graph shows the 'rating life' – sometimes called the '90 percent survival life', this is the calculated life obtained by following the procedure set out in this catalogue. Also shown is the average life, which is appreciably greater than the 'rating life'.



The reason for this difference is that even with the best steel minute imperfections exist in the material and, as the area of contact between the rolling elements and rings under load is very small, these imperfections upset the distribution and intensity of stress in the material. Variations in contact area resulting from the manufacturing tolerances on the rings and rolling elements also contribute towards this difference.

In addition to the load conditions on a bearing, failure can also result from other factors, notably lack of attention to lubrication, protection or accuracy of mounting, these cannot be included in the basic load/life formulae.

The required basic static load rating C_0 of a bearing can be determined using the equation

$$C_0 = S_0 P_0$$

where:

C_0 = basic load rating [KN]

P_0 = equivalent static load [KN]

S_0 = static safety factor

For bearings operating in elevated temperatures the hardness of the bearing material will be reduced.

Values of S_0 for a few typical non rotating bearing applications are shown below and may be used as a guide.

Application	S_0 Factor
Variable pitch propeller blades on aircraft	0.5
Dams on aircraft	1.0
Swing bridges	1.5
Crane hooks for large cranes without additional dynamic forces	1.5
Small cranes for bulk goods with large additional dynamic forces	1.6

On rotating bearings where the load fluctuates dramatically or, where heavy shock loads occur during a fraction of a revolution, it is necessary to check that the basic static load rating is adequate. Heavy shock loads could cause permanent deformation in the form of indentation being unevenly distributed over the raceway. Shock loads are also generally such that they cannot be calculated exactly. In some cases they may also cause deformation of the housing and therefore producing unfavorable load distribution. Depending on the operating conditions the maximum load should not exceed a value determined by the static safety factor S_0 .

Values for S_0 for certain operating conditions can be used.

Application	S_0 Factor (min ^m)
Operation is smooth and vibration free	0.5
Operation is normal and vibration conditions normal	1.0
Pronounced shock loads	1.5 - 2
Demand on smooth running is of prime importance	2.0

Bearing Life and Load Ratings

BASIC DYNAMIC LOAD RATING C_r

Basic dynamic load rating (C_r) is defined as that constant radial load which a group of apparently identical radial ball bearings, angular contact ball bearings and radial roller bearings can endure for a rating life of one million revolutions.

For thrust ball bearings, the basic dynamic load rating is that constant, central, axial load which a group of apparently identical thrust bearings can endure for a rating life of one million revolutions.

STATIC LOAD RATING C_o

The static load C_o is defined as a load acting on a non-rotating bearing. Permanent deformations appear in rolling elements and raceways under static load of moderate magnitude and increase gradually with increasing load. The permissible static load is, therefore, dependent upon the permissible magnitude of permanent deformation.

Experience shows that a total permanent deformation of 0.0001 of the rolling element diameter, occurring at the most heavily loaded rolling element and raceway contact, can be tolerated in most bearing applications without impairment of bearing operation.

RATING LIFE

Rating life (L) is defined as the number of revolutions (or hours at some constant speed) that 90% of a group of apparently identical bearings will exceed before the first evidence of fatigue develops. This may be referred to as B10 life.

LIFE EQUATION

The expression $L_u = (C/P)^3 [10^6 \text{ revs}]$ is used to establish a mathematical relationship for the rating life as a function of the load.

L_u = rating life in millions of revolutions of the inner ring with constant direction of loading

C = basic dynamic load rating in [KN]

P = equivalent dynamic load rating in [KN]

p = exponent for life equation

$p = 3$ for ball bearings

$p = 10/3$ for roller bearings

In most cases it is common practice to employ the rating life L_h (hours). The relationship between L_u and L_h with constant rotational speed n (rpm) is:

$$L_u = \frac{L_h \cdot n \cdot 60}{10^6} \cdot [10^6 \text{ revs}]$$

If the rating life of 1×10^6 revs, to which the basic load rating C refers, is resolved into a reference life $L_h = 500$ hours, and a reference rotation speed of $n = 33.1/3$ rpm it follows that

for ball bearings: $\left(\frac{C}{P}\right)^3$

for roller bearings: $\left(\frac{C}{P}\right)^{10/3}$

$$L_u = \frac{Lh.n.60}{500.33.1/3.60} = \left(\frac{C}{P} \right)^3 [10^6 \text{ revs}]$$

$$\text{or: } \sqrt[3]{\frac{LH}{500}} = \sqrt[3]{\frac{33.1/3}{n} \cdot \frac{C}{P}}$$

$$\text{letting: } \sqrt[3]{\frac{33.1/3}{n}} = \text{speed factor } f_n \text{ (equation 1)}$$

$$\text{and: } \sqrt[3]{\frac{LH}{500}} = \text{life factor } f_L \text{ (equation 2)}$$

The rating life equation may be obtained in the form life factor: $f_L = \frac{C.f_n}{P}$

$$\text{basic load rating required: } C = \frac{P.f_L}{f_n} [KG]$$

The relationship of equation 1 and 2 are graphically represented in nomograms below. Also on page 14 are charts showing the L10 life in relation to C/P for ball and roller bearings. To determine the size of a rolling bearing for a particular field of operation it is necessary to establish the nominal life corresponding to the field of application.

EXAMPLE

A deep groove ball bearing is required to run at speed $n=850$ RPM under constant radial load of $f_r = 5$ KN and is to achieve a basic rating life L_{10h} of 20.000 hours.

From the nomogram using the right hand column (L_{10}) a line drawn from 20.000 to the left hand column (n RPM) this passes through the centre column (C/P L_{10} 10.6) at 10:1000 therefore a bearing is required with a basic load rating C of at least $C = 10 \times 5$ KN. Reading from the tables relating to deep groove ball bearings it can be seen that a bearing ref 6309 has a C value of 52.7 KN. Of course the choice of bearing is also governed by the shaft and housing parameters.

For motor vehicles and rolling stock the service life is expressed as a function of the wheel diameter and kilometers travelled as per formulae below:

$$L_{10} = \frac{1000}{\pi D} . L_{10s} \quad \text{or} \quad L_{10s} = \frac{\pi D}{1000} L_{10}$$

Where:

L_{10} = nominal life in 10^6 RPM

L_{10s} = life in 10^6 kilometers traveled

D = diameter of wheel in meters. Values for selecting service life in kilometers covered are in table below.



Bearing Life and Load Ratings

Vehicle type	L10s/10 ⁶ km
Wheel bearings for motor vehicles:	
Cars	0.2
Trucks, buses	0.4
Axle boxes for rolling stock – freight cars	0.8
Suburban traffic	1.5
Long distance coaches	3
Rail cars	3-1
Diesel and electric locomotives	3-4

Depending on the working temperature of the bearings, their service life is reduced at elevated temperatures. This is to be taken into consideration when the service life is established by the application of temperature factor f_t specified in table below:

Working temperature °C	150	200	250	300
Symbol	S0	S1	S2	S3
f_t	1	0.73	0.42	0.22

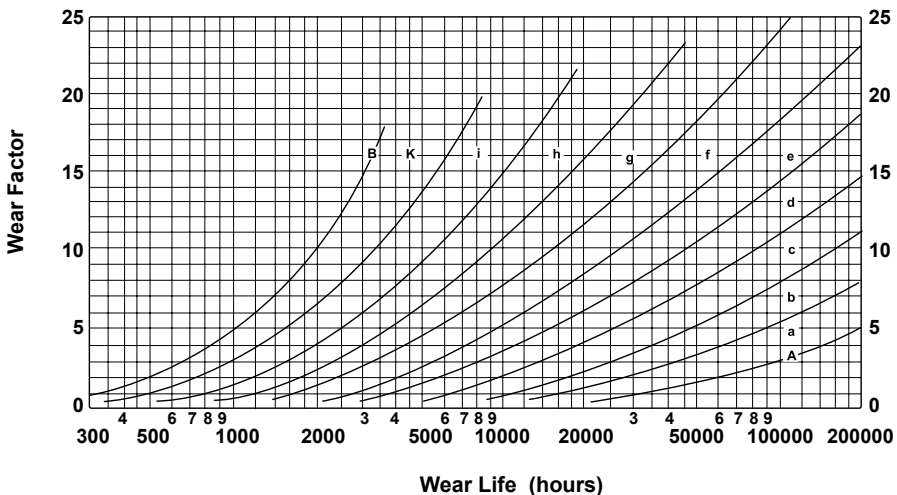
The following table provides recommendations for factor f_v along with typical applications and life factor f_L .

Application	Fields of operating conditions	Factor f_v	Factor f_L
Motor vehicles - gear boxes - axle drives - water pumps - wheel bearings	g – k h – k k h – l	3 – 8 3 – 6 5 – 7 4 – 6	1.7 – 2.2 2 – 3 1.5 – 2 1.6 – 2.5
Railbound vehicles - haulage trolleys - trams - passenger coaches and freight cars - motor coaches and locomotives - gears	f – h e – f c – d d – e c – d	12 – 15 8 – 12 8 – 12 6 – 10 3 – 6	2.5 – 3 3.5 – 4 3 – 3.5 3.5 – 4 3 – 4.5
Motors - electric motors for household appliances - traction motors and standard motors - large motors	i – k c – d b – d	3 – 5 3 – 5 3 – 5	1.5 – 2 3.5 – 4.5 4 – 4.5

Bearing Life and Load Ratings

Application	Fields of operating conditions	Factor f_v	Factor f_L
Machine-tools - lathe spindles and milling spindles - boring and grinding machine spindles - machine tool gears - electric and pneumatic tools	a – b c – d c – d g – h	0.5 – 1.5 0.5 – 1 3 – 8 3 – 6	3 – 4.5 2.5 – 3.5 3 – 4 1.8 – 2.7
Woodworking machines - milling cutter and cutter shaft - main bearing - rod bearing	e – f e – g c – d	0.5 – 3 3 – 4 2 – 3	3 – 4 3 – 4 2 – 3
Gears general engineering - universal gears - large-sized gears, stationary	d – c c – d	3 – 1 2 – 3	2 – 3 2 – 3
Materials handling - belt drives opencast mining - medium-sized and large fans - centrifugal pumps and compressors	c – d c – l d – f	5 – 12 3 – 5 3 – 5	4 – 6 3 – 4.5 3 – 4.5
Crushers, mills, screens etc. - jaw crushers, roll crushers	f – g	8 – 12	3 – 3.5
Hammer mills - hammer mills and impact mills - tube mills - vibrating mills - vibrating screens	d – c f – g f – g e – f	5 – 8 12 – 18 3 – 5 4 – 6	3.5 – 4.5 3 – 5 2 – 3 2.5 – 3

The wear life diagram indicates the operating conditions with the least wear factor at curve A and the heaviest wear occurring at curve B. The area between A and B being subdivided into individual fields from a to k shows how the operating conditions deteriorate progressively.



Bearing Life and Load Ratings

ADJUSTED RATING LIFE

Adjustments to life equations

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

The above formula is adequate for conventional applications but in exceptional cases other factors must be considered which influence the life of the bearing. To accommodate these factors the ISO life equation is

$$L_{na} = a_1.a_2.a_{iso} \cdot \left(\frac{C}{P} \right)^p \quad \text{or} \quad L_{na} = a_1.a_2.a_3.L_{10}$$

Where:

L_{na} = adjusted rating life in 10^6 revolutions the index being the difference between the specified probability life and 100%

a_1 = life adjustment factor for reliability

a_2 = life adjustment for material

a_{iso} = life adjustment for operating conditions including tribology

Calculations for the adjusted rating life are based on the pre-conditions mentioned in the above formulae ie. That bearing loads can be calculated with accuracy considering all aspects of the loads involved along with shaft deflection etc. Also, that reliability of the bearing materials are in accordance with the corresponding C values and that normal operating conditions $a_1=a_2=a_3=1$ and that two life equations become identical.

Life adjustment factor a_1 for reliability

The a_1 factor is used to determine lives which are obtained or exceeded with a greater probability than 90% (L10). The table below lists the factors for failure probability values between 10% and 1% L10 being the normal rating life.

Probability %	Failure probability %	Life before fatigue appears	Factor a_1
90	10	L10	1
95	5	L5	0.62
95	4	L4	0.53
97	3	L3	0.44
98	2	L2	0.33
99	1	L1	0.21

Life adjustment factor a_2 for material

The factor a_2 accounts for the properties of the material and its heat treatment. $a_2=1$ is applicable to the high quality steels used in the production of normal bearing series. For elevated temperatures see reduction in dynamic load rating in table below.

Working temperature °C	150	200	250	300
Symbol	S0	S1	S2	S3
ft	1	0.73	0.42	0.22

Life adjustment factor a_{iso} for operating conditions

The efficiency of lubrication is determined primarily by the degree of separation between the rolling elements and raceways. The highest life values are reached when there is a hydrodynamic state of lubrication where metal to metal contact does not exist between rolling elements and raceway and under the cleanest conditions which would normally prevail in an adequately sealed bearing arrangement.

The a_{iso} factor can be calculated or estimated graphically. We cover the graphical method on the next page.



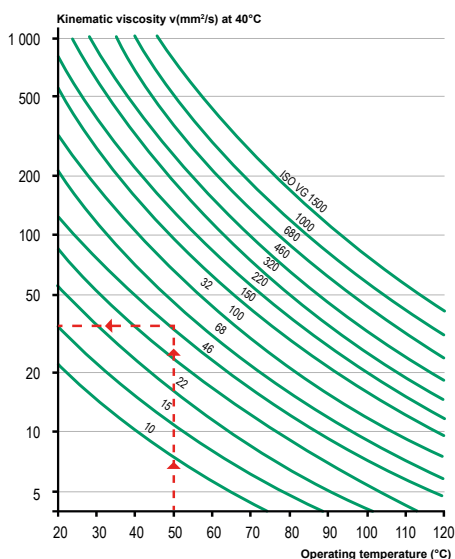
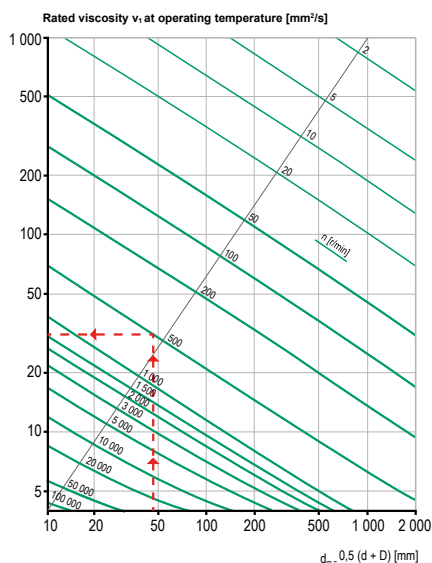
Bearing Life and Load Ratings

Define the rated viscosity V1

Define the rated viscosity based on bearing mean diameter (Dm) and speed (n). In the left graphic, find the intersection between n and Dm. Report it on the V1 axis to obtain the rated viscosity.

DEFINE THE REQUIRED ISO VG VISCOSITY

Define the required ISO VG viscosity suitable for the application (Right graphic). Mark the V_1 value on the Y axis and the operating temperature (t) on the X axis. Select a ISO VG hyperbolic above the intersection of V_1 and t. select 1 or 2 above.



FACTOR KAPPA (K)

The a_{iso} factor is based on the viscosity ratio Kappa (K). It is defined as the ratio of the actual lubricant viscosity V for the viscosity v_1 required for adequate lubrication. K should be higher than 1. With thinner lubricating films, there is an increase in metal to metal contact and life expectancy decreases.

$$kappa = V / V_1 \gg 1 \gg 4$$

Cleanliness Factor (ec)

Level of contamination	ec	
	$D_{pw} < 100 \text{ mm}$	$D_{pw} \geq 100 \text{ mm}$
Extreme cleanliness Particle size of the order of lubricant film thickness; laboratory conditions	1	1
High cleanliness Oil filtered through extremely fine filter; conditions typical of bearing greased for life and sealed	0,8 to 0,6	0,9 to 0,8
Normal cleanliness Oil filtered through fine filter; conditions typical of bearings greased for life and shielded	0,6 to 0,5	0,8 to 0,6
Slight contamination Slight contamination in lubricant	0,5 to 0,3	0,6 to 0,4
Typical contamination Conditions typical of bearings without integral seals; coarse filtering; wear particles and ingress from surroundings	0,3 to 0,1	0,4 to 0,2
Severe contamination Bearing environment heavily contaminated and bearing arrangement with inadequate sealing	0,1 to 0	0,1 to 0
Very severe contamination	0	0

Select the appropriate ec for the application.



Bearing Life and Load Ratings

CALCULATE THE LOAD RATIO

$$e_c Cu / Pe$$

Where:

- e_c = Cleanliness factor
- Cu = Fatigue limit load rating
- Pe = Equivalent load

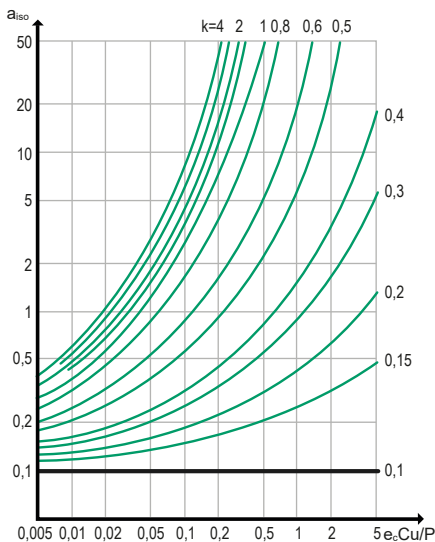
a_{iso} factor is

$$a_{iso} = f(k, e_c Cu / Pe)$$

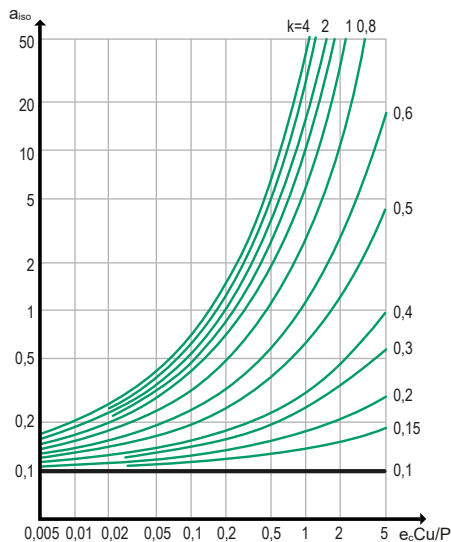
Where:

- k = Kappa factor

In the below graphics, report the load factor on the X axis and select the kappa hyperbolic. Report the intersection point on the a_{iso} axis. The a_{iso} adjustment factor, depending the bearing type, use the left graphic for ball bearing or right graphic for roller bearing



BALL



ROLLER

SERVICE LIFE

Since the fatigue life modified by the adjustment factors a , a_2 and a_{iso} only considers material fatigue as the cause of failure, the calculated life corresponds to the service life only if the following points are met:

- (a) Lubrication conditions are constant throughout.
- (b) Loads and speeds used for analysis are a true reflection of the actual operating conditions.
- (c) Operating viscosity is based on actual operating temperature.
- (d) Lubricant contamination is limited during the whole running time.
- (e) The service life limited by wear and break down of lubrication is not shorter than the fatigue life.

Wear of the acting surfaces is primarily caused by contamination which over a period of time may penetrate the bearing. The situation is made worse by inadequate lubrication and corrosion due to condensation. The amount of wear experienced in a bearing is dependent on the operating conditions, lubrication and effective sealing arrangement.

WEAR FACTOR

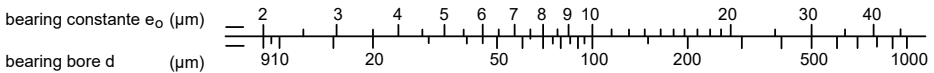
The permissible amount of wear is expressed by the wear factor f_v .

$$f_v = \frac{v}{e_o}$$

where:

v = permissible increase in radial clearance (mm)

e_o = bearing constant depending on the bore diameter – see below for e_o values in relationship with bore diameter mm.



Limiting Speed

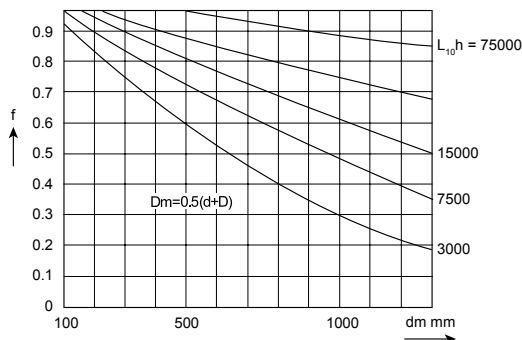
LIMITING SPEED

The maximum rotational speed of ball and roller bearings depends upon various factors. The size and design of the bearing, type of lubrication whether grease or oil, type of cage fitted and the internal clearance of the bearing when mounted.

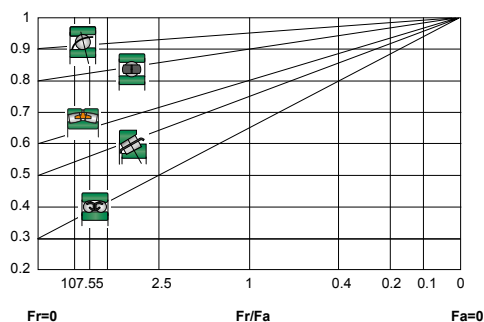
If the radial run-out, which produces out of balance forces, is reduced, then higher speeds can be obtained for such as machine-tool applications, hence the use of high precision bearings. Reduction of cage weight will also reduce out of balance forces as when made from light alloy or plastic. Cages that are centred on the inner or outer races rather than the rolling elements are used for high speed applications.

The surface of the riding lips being specially ground and lubrication between the sliding surface must be maintained. Heavier loads influence the speed and also effect the basic rating life of L10h75000 hours.

In such cases the speeds listed in the tables should be multiplied by a factor f which you can obtain from the fig. 1.



For combined loads the speeds indicated in the bearing tables are to be multiplied by the reduction factor f_1 given in diagram fig. 2.



For ball thrust bearings there must be a minimum load applied to counteract the centrifugal forces of the balls on rotation. Factor M is indicated in the bearing tables against the appropriate bearing size.

Bearing rings and rolling elements are subjected to high stresses on a very small contact area and must have a high resistance to wear as well as high elastic and fatigue limits. Primarily these are manufactured from high-carbon chromium bearing steel with a chemical composition as indicated in the table 1 below and are in accordance with SAE 52100 - 100 Cr6.

HIGH CARBON CHROMIUM BEARING STEEL Steel Grade	Chemical Composition %					
	C	Mn	Si	Cr	S	P
105Cr4	1.00..1.10	0.25..0.40	0.15..0.35	0.90...1.15	<0.025	<0.030
100Cr6 / ROL 2	0.90..1.05	0.25..0.40	0.15..0.35	1.40..1.65	<0.025	<0.025
100CrMn6	0.90..1.05	1.00..1.20	0.50..0.70	1.40...1.60	0.020	<0.025

CAGE MATERIALS

Types of cages for bearings vary in accordance with the operating conditions. The most common are those made from pressed steel. Machined cages are made from high strength copper alloys or carbon steel and for high speeds manufactured from plastic or phenolic resins.

HEAT TREATMENT

Bearings are generally used up to a temperature of maximum + 150°C. In case of higher temperatures, bearings with special heat treatments should be used. Sealed bearings, 2RS type, should be used at operating temperatures up to +80°C. If this temperature is exceeded, the efficacy of lubricants is considerably reduced.

In order to use bearings at a higher operating temperature, the bearings have to be subjected to a special heat treatment. This will ensure the dimensional stability, but will reduce the lifetime by a factor (ft) as per the table.

Operating temp °C	150	200	250	300
Symbol	S0	S1	S2	S3
ft	1	0.73	0.42	0.22

Handling of Bearings

CARE & FITTING OF BEARINGS

STORAGE

1. Store ball and roller bearings in a clean, dry place in their original wrappings. This will preserve them from deterioration.
2. Use older stocks first.
3. Do not stack too many large bearings on top of each other otherwise the protective oil could be squeezed out from between the bearing and its wrapping, thus leading to corrosion problems. Also, never store large bearings upright, but lay them flat.

FITTING

4. Absolute cleanliness is essential when handling bearings. They should not be removed from their wrappings until required for fitting. A smooth metal topped bench that can be wiped clean is a great advantage. All tools, shafts, housings and other components must be perfectly clean. If fitting operations are delayed or interrupted, the assembly should be wrapped with grease proof paper to exclude dirt and dust.
5. All bearings are usually coated with a rust preventative oil, unless pre-lubricated and/or packed to suit individual requirements. There is no need to remove this oil unless:
 - It is sufficient to cause serious dilution of the oil or grease used in the bearing. This normally applies to smaller bearings where the rust preventative oil represents a large proportion of the required amount of lubricant.
 - Low torque is required.
 - A synthetic lubricant is used that may not be compatible with the protecting oil.
6. To remove the rust preventative oil, wash the bearing in a good-quality washing fluid; white spirit or good-quality paraffin are suitable. Allow the bearings to drain thoroughly. Finally dry them, the following being satisfactory methods.

Place the bearing in an oven or on a hot plate, temperature of 65-80°C should be adequate.

Direct dry, clean, compressed air on to the bearings. The cage and rings of smaller bearings must be held firmly otherwise a sudden blast of air would rapidly accelerate the free bearing parts; this could cause the balls to skid, thus damaging the highly finished internal surfaces of the bearing.

The fits of the rings on their seating's are very important. Therefore, ensure that the shaft and housing seatings are of correct size and of good shape.

7. All shoulders must be smooth and square with the axis of rotation.
8. Never drive one ring on to its seating by blows on the other. Such blows would irretrievably damage the balls or rollers and raceways.
9. Apply pressure evenly around the rings. A press is better than a hammer.
10. Should a hammer be used, mild steel or brass tube of suitable size, faced up square, should be interposed between it and the bearing. This will distribute the force of the blows (or rather taps), which should be given progressively around the ring.

11. When the parts of a separable roller bearing are brought together, the inner ring, the outer ring and the rollers must all be square one with the other. If not square, then the rollers would not slide freely, and force would have to be used to bring the parts together. Such force would result in the rollers and raceways becoming scored and this, in addition to causing noisy running, could cause early failure of the bearing.
12. Where the ring of a bearing is to be against an abutment, make sure it is properly seated.
13. For heavy interference fits, inner rings may be shrunk onto their seatings after heating in clean mineral oil at a temperature of approximately 100%. Be sure that the bearing is in contact with the abutment shoulder after it has cooled.
14. In the case of taper clamping sleeve and nut bearings, the clamping nut must not be overtightened, this could expand the inner ring and eliminate all clearance within the bearing, or even fracture the inner ring.
15. It is recommended that when using pin spanners they should have a length of approximately five times the shaft diameter, one or two light hammer blows should be given to the handle of the spanner after the nut has been tightened as far as possible by hand pressure; this should tighten the nut sufficiently. It is good practice, if possible, to check that the sleeve is still clamped firmly to the shaft after a few days running.

As an additional precaution it is recommended that, whenever possible, the bearings are fitted so that the rotation of the shaft tends to tighten the nut on the sleeve.

When using torque spanners it is recommended that the following torques be applied to the clamping nut.

For LIGHT series bearings

Shaft Diameter	Torque on Nut
1" and 25m/m	7.6 Kgm/M (55 lbs ft)
1 1/2" and 40 m/m	12.4 Kgm/M (90 lbs ft)
2" and 50 m/m	17.25 Kgm/M (125 lbs ft)
3" and 75 m/m	30.3 Kgm/m (220 lbs ft)

For MEDIUM series bearings increase the above figures by approximately 50 percent.

DISMANTLING AND REPLACEMENT

16. Unnecessary removal of a bearing should be avoided, particularly where interference fits have been used. Removal can damage a bearing and, in some instances, cause deterioration of the interference fit. Very often it is sufficient to clean and relubricate the bearing in its fitted position.

Only remove a bearing if you need to inspect it closely. Symptoms that guide are the condition of the lubricant, the bearing temperature and noise level.

With roller journal bearings there is sometimes a ball locating bearing. this may be only push fit on the shaft, and therefore facilities easy dismantling.

Handling of Bearings

17. In certain applications some form of extractor may be necessary. This must act directly on the ring to be removed. Never try to remove the inner ring by applying force to the outer ring, or vice versa.
18. Thrust bearings offer no difficulty as push fits should be used; but, take care to keep the rings square or they will bind.
19. Carefully protect bearings from dirt moisture while they are out of their housings.

It is advisable to wash them thoroughly immediately after removal, by the following procedure.

- Immerse in a washing fluid such as clean white spirit or good-quality paraffin. The washing fluid must not attack the bearing components. After Soaking, move each separate bearing around in the fluid, using a basket or other container if convenient. Occasional slow oscillations of the bearing rings will help to dislodge dried out grease and other matter.
- When clean, thoroughly drain and dry.



The Radial clearance is the total internal clearance between the balls or rollers in a bearing and their raceways measured normal to the axis of the bearing. This clearance compensates for (a) expansion of the inner ring and/or contraction of the outer ring when interference fits are used, (b) for differential expansion of the two ring when the inner ring of a bearing operates at a higher temperature than the outer ring; (c) accommodates the minute inaccuracies unavoidable with even the most modern methods of machining; (d) affects the end play and also affects their capacity for carrying axial loads – the greater the radial clearance, the greater the capacity for supporting axial load.

When bearings with small radial clearances are used, special attention must be given to the selection of seating dimensions.

Once ball and roller bearings are mounted and running, a small amount of radial or running clearance is normally desirable. In the case of bearings under radial load, quieter running is generally obtained when this clearance is a minimum.

Radial clearance figures for Ball and Roller bearings mentioned in our Tables are in accordance with I.S.O. 5753 recommendations.

For normal applications the general guide given in Table 1 below may be used. Excessive radial tightness in the bearing should be avoided under all conditions.

STORAGE

TABLE 1 – SUMMARIZING THE CORRECT RADIAL CLEARANCE

Radial clearance of bearing	Fit of races on seating	Possibility of temperature changes reducing radial clearance
C2	No appreciable interference either race	absent
CN	One race only interference fit	absent
C3	One race only interference fit	present
C3	Both races interference fit	absent
C4	Both races interference fit	present

C2 fit: These bearings have the smallest amounts of radial clearance. They should only be used where freedom from all play is required in the assembled bearing and there is no possibility of the initial radial clearances being eliminated by external causes. Therefore, special attention must be given to the seating dimensions, as the expansion of the inner ring or contraction of the outer ring may cause tight bearings.

CN fit: This grade of radial clearance is intended for use where only one ring is made an interference fit and there is no appreciable loss of clearance due to temperature differences. Ball and Roller bearings for general engineering applications are usually of this clearance.

C3 fit: This grade of radial clearance should be used when both rings of a bearing are made an interference fit, or when only one ring is an interference fit but there is likely to be some loss of clearance due to temperature differences. It is the grade normally employed for roller bearings on general engineering applications, especially where there is a tendency for “creep” to take place due to out-of-balance loading.

Fits and Clearance

It is also the grade normally used for ball bearings that take axial loading, but for some purposes even “C4” fit bearings may be required.

C4 fit: Where there will be some loss of clearance due to temperature differences and both races must be an interference fit, this is the grade of radial clearance to adopt. One example of its use is in bearings for vibration screen.

Where seating limits give an interference fit tighter than the recommended figures, or where temperature differences could cause radial tightness, the correct clearance can be established by calculating the maximum loss of clearance at both extremes of the following basis. A suitable clearance grade from the tables can then be selected.

Total loss of clearance = RI + RO + RT + RM

RI = Expansion of inner ring raceway due to shaft interference. (See table below).

RO = Contraction of outer ring raceway due to housing interference. (See table below).

RT = Loss of clearance due to the inner ring being at a higher temperature than the outer ring.

RM = Loss of clearance due to increase in seating interference resulting from nonferrous seating expanding or contracting at different rates from bearing steel.

Table 2 below gives approximate values for RO and RI assuming a solid shaft and substantial housing.

TABLE 2

Bearing series	Inner ring raceway expansion RI	Outer ring raceway Contraction RO
Extra light	100% interference	80% interference
Light	80% interference	60% interference
Medium and heavy	70% interference	50% interference

For the clearance tables, refer to the bearing type chapter

SHAFT TOLERANCES FOR CYLINDRICAL BORE BEARINGS

Type of Load	Bearing type	Diameter	Axial Movement Magnitude of Load	Tolerance field	
Rotating Outer Ring Load	Ball& Roller Bearing	All sizes	Angular contact ball bearing and tapered roller bearing adjustment via inner ring	g6 h6 h6	(g5) (h5) (j6)
Rotating Inner Ring or indeterminate Load	Ball Bearing	Up to 40mm	normal load	J6	(j5)
		Up to 100mm	low load normal & high load	J6 k6	(j5) (k5)
		Up to 200mm	low load normal & high load	K6 m6	(k5) (m5)
		Over 200mm	normal load high load, shock load	M6 n6	(m5) (n5)
-	Roller Bearing	Up to 60mm	low load normal & high load	J6 k6	(j5) (k5)
		Up to 200mm	low load normal load high load	K6 m6 n6	(k5) (m5) (n5)
		Up to 500mm	normal load high load, shock load	m6 p6	(m5)
		Over 500mm	normal load high load	N6 p6	(n5)

Shaft and Housing Fits

SHAFT TOLERANCES FOR THRUST BEARINGS

Type of Load	Bearing type	Diameter	Operating conditions	Tolerance field	
Thrust load	thrust ball bearings	all sizes		J6	
	thrust ball bearings double acting	all sizes		J6	(k6)
	cylindrical roller thrust bearing	all sizes		h6	(j6)
	thrust cylindrical roller & cage assembly	all sizes		h10	
	thrust cylindrical roller & cage assembly or thrust needle roller & cage assembly	all sizes		h8	
Combined Load	spherical roller thrust bearing	all sizes	point load on shaft washer	j6	
		up to 200mm	circumferential	j6	(k6)
		over 200 mm	load on shaft washer	k6	(m6)

ADAPTER SLEEVES, WITHDRAWAL SLEEVES

	Permissible geometrical inaccuracy (out-of-roundness taper)	Tolerance field
Adapter sleeves and withdrawal sleeves	IT 5/2	H7
	IT 5/2	h8
	IT 6/2	h9

SHAFT TOLERANCES FOR ADAPTER SLEEVES AND WITHDRAWAL SLEEVES

d mm		H7		IT 5/2	h8		IT 5/2	h9		IT 6/2
μm										
over	up to	upper	lower	max.	upper	lower	max.	upper	lower	max.
0	0	0	0	0	0	0	0	0	0	0
6	10	0	-15	3	0	-22	3	0	-36	4.5
10	18	0	-18	4	0	-27	4	0	-43	5.5
18	30	0	-21	4.5	0	-39	4.5	0	-52	6.5
30	50	0	-25	5.5	0	-39	5.5	0	-62	8
50	80	0	-30	6.5	0	-46	6.5	0	-74	9.5
80	120	0	-35	7.5	0	-54	7.5	0	-87	11
120	180	0	-40	9	0	-63	9	0	-100	12.5
180	250	0	-46	10	0	-72	10	0	-115	14.5
250	315	0	-52	11.5	0	-81	11.5	0	-130	16
315	400	0	-57	12.5	0	-89	12.5	0	-140	18
400	500	0	-63	13.5	0	-97	13.5	0	-155	20

Note: IT basic tolerances indicate accepted from circularity and cylindricity

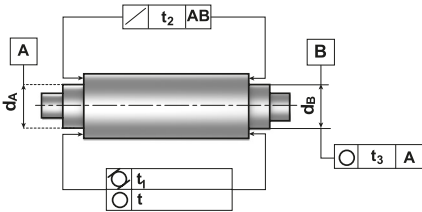
HOUSING TOLERANCES FOR RADIAL BEARING

Type of Load	Axial Movement Magnitude of Load	Operation Conditions	Tolerance field	
Rotating inner Ring Load	Outer Ring slides in Housing	closeness of tolerance function of running accuracy	H7	(H6)
		high running accuracy	H7	(J6)
		standard running accuracy	H7	(J6)
		temperature increase through shaft	G7	
Rotating Outer Ring Load or indeterminate load	low load	with high running accuracy requirements K6,M6,N6 and P6	K7	(KJ6)
	normal load, shock load		M7	(M6)
	high load, shock load		N7	(N6)
	high load, heavy shock load thin-walled housings		P7	(P6)

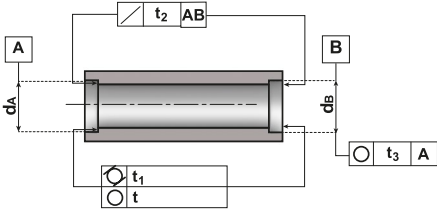
SHAFT TOLERANCES FOR THRUST BEARINGS

Type of load	Bearing type	Operating conditions	Tolerance field	
Thrust load	thrust ball bearings	standard running accuracy	E8	(K6)
		high running accuracy	E6	
	cylindrical roller thrust bearing		H7	
	thrust cylindrical roller & cage assembly		H11	
	thrust cylindrical roller & cage assembly		H10	
	spherical roller thrust bearing	normal load	E8	
		high load	G7	
Radial & axial loads on spherical roller thrust bearings	stationary load on housing washer		H7	
	rotating load on housing washer		M7	

Accuracy of shaft



Accuracy of housing



Description	Fit	Symbol of tolerance		Allowable tolerances depending on precision classes				
				P0	P6	P5	P4(SP)	P2(UP)
Dimensional accuracy	shaft	--		IT 6	IT 5	IT 4	IT 4	IT 3
	housing			IT 7	IT 6	IT 5	IT 5	IT 4
Roundness	shaft		t_1	IT 4	IT 3	IT 2	IT 1	IT 0
	housing			IT 5	IT 4	IT 3	IT 2	IT 1
Cylindricity	shaft		t_2	IT 3	IT 3	IT 2	IT 1	IT 0
	housing			IT 4	IT 3	IT 3	IT 2	IT 1
Runout	shaft		t_3	IT 3	IT 3	IT 2	IT 1	IT 0
	housing			IT 4	IT 3	IT 2	IT 2	IT 1
Eccentricity	shaft		t_e	IT 5	IT 4	IT 4	IT 3	IT 3
	housing			IT 6	IT 5	IT 5	IT 4	IT 3

Shaft and Housing Fits

BASIC TOLERANCE RANGE ISO-IT

d; D		IT 0	IT 1	IT 2	IT 3	IT 4	IT 5	IT 6	IT 7	IT 8
from	to									
mm		μm								
1	3	0.5	0.8	1.2	2	2	4	6	10	14
3	6	0.6	1	1.5	2.5	4	5	8	12	18
6	10	0.6	1	1.5	2.5	4	6	9	15	22
10	18	0.8	1.2	2	3	5	8	11	18	27
18	30	1	1.5	2.5	4	6	9	13	21	33
30	50	1	1.5	2.5	4	7	11	16	25	39
50	80	1.2	2	3	5	8	13	19	30	46
80	120	1.5	2.5	4	6	10	15	22	35	54
120	180	2	3.5	5	8	12	18	25	40	63
180	250	3	4.5	7	10	14	20	28	46	72
250	315	4	6	8	12	16	23	32	52	81
315	400	5	7	9	13	18	25	36	57	89
400	500	6	8	10	15	20	27	40	63	97

DEVIATION OF SHAFT DIAMETERS 0,001 mm

Nominal diameter		g5		g6		h5		h6		h8		h10		j5	
over	inc	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min
3	6	-4	-9	-4	-12	0	-5	0	-8	0	-18	0	-48	+3	-2
6	10	-5	-11	-5	-14	0	-6	0	-9	0	-22	0	-58	+4	-2
10	18	-6	-4	-6	-17	0	-8	0	-11	0	-27	0	-70	+5	-3
18	30	-7	-16	-7	-20	0	-9	0	-13	0	-33	0	-84	+5	-4
30	50	-9	-20	-9	-25	0	-11	0	-16	0	-39	0	-100	+6	-5
50	80	-10	-23	-10	-29	0	-13	0	-19	0	-46	0	-120	+6	-7
80	120	-12	-27	-12	-34	0	-15	0	-22	0	-54	0	-140	+6	-9
120	180	-14	-32	-14	-39	0	-18	0	-25	0	-63	0	-160	+7	-11
180	250	-15	-35	-15	-44	0	-20	0	-29	0	-72	0	-185	+7	-13
250	315	-17	-40	-17	-49	0	-23	0	-32	0	-81	0	-210	+7	-16
316	400	-18	-43	-18	-54	0	-25	0	-36	0	-89	0	-230	+7	-18
400	500	-20	-47	-20	-60	0	-27	0	-40	0	-97	0	-250	+7	-20

DEVIATION OF HOUSING DIAMETERS 0,001 mm

Nominal diameter		E8		G6		G7		H6		H7		H10		H11		J6	
over	inc	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min
6	10	+47	+25	+14	+5	+20	+5	+9	0	+15	0	+58	0	+90	0	+5	-4
10	18	+59	+32	+17	+6	+24	+6	+11	0	+18	0	+70	0	+110	0	+6	-5
18	30	+73	+40	+20	+7	+28	+7	+13	0	+21	0	+84	0	+130	0	+8	-5
30	50	+89	+50	+25	+9	+34	+9	+16	0	+25	0	+100	0	+160	0	+10	-6
50	80	+106	+60	+29	+10	+40	+10	+19	0	+30	0	+120	0	+190	0	+13	-6
80	120	+126	+72	+34	+12	+47	+12	+22	0	+35	0	+140	0	+220	0	+16	-6
120	180	+148	+85	+39	+14	+54	+14	+25	0	+40	0	+160	0	+250	0	+18	-7
180	250	+172	+100	+44	+15	+61	+15	+29	0	+46	0	+185	0	+290	0	+22	-7
250	315	+191	+110	+49	+17	+69	+17	+32	0	+52	0	+210	0	+320	0	+25	-7
315	400	+214	+125	+54	+18	+75	+18	+36	0	+57	0	+230	0	+360	0	+29	-7
400	500	+232	+135	+60	+20	+83	+20	+40	0	+63	0	+250	0	+400	0	+33	-7

DEVIATION OF HOUSING DIAMETERS 0,001 mm

Nominal diameter		j6		k5		k6		m5		m6		n5		n6		p6	
over	inc	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min
3	6	+6	-2	+6	+1	+9	+1	+9	+4	+12	+4	+13	+8	+16	+8	+20	+12
6	10	+7	-2	+7	+1	+10	+1	+12	+6	+15	+6	+16	+10	+19	+10	+24	+15
10	18	+8	-3	+9	+	+12	+1	+15	+7	+18	+7	+20	+12	+23	+12	+29	+18
18	30	+9	-4	+11	+2	+15	+2	+17	+8	+21	+8	+24	+15	+28	+15	+35	+22
30	50	+11	-5	+13	+2	+18	+2	+20	+9	+25	+9	+28	+17	+33	+17	+42	+26
50	80	+12	-7	+15	+2	+21	+2	+24	+11	+30	+11	+33	+20	+39	+20	+51	+32
80	120	+13	-9	+18	+3	+25	+3	+28	+13	+35	+13	+38	+23	+45	+23	+59	+37
120	180	+14	-11	+21	+3	+28	+3	+33	+15	+40	+15	+45	+27	+52	+27	+6	+43
180	250	+16	-13	+24	+4	+33	+4	+37	+17	+46	+17	+51	+31	+60	+31	+79	+50
250	315	+16	-16	+27	+4	+36	+4	+43	+20	+52	+20	+57	+34	+66	+34	+88	+56
316	400	+18	-18	+29	+4	+40	+4	+46	+21	+57	+21	+62	+37	+73	+37	+98	+62
400	500	+20	-20	+32	+5	+45	+5	+50	+23	+63	+23	+67	+40	+80	+40	+108	+68

Shaft and Housing Fits

DEVIATION OF SHAFT DIAMETERS 0,001 mm

Nominal diameter		j6		k5		k6		m5		m6		n5		n6		p6	
over	inc	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min
3	6	+6	-2	+6	+1	+9	+1	+9	+4	+12	+4	+13	+8	+16	+8	+20	+12
6	10	+7	-2	+7	+1	+10	+1	+12	+6	+15	+6	+16	+10	+19	+10	+24	+15
10	18	+8	-3	+9	+	+12	+1	+15	+7	+18	+7	+20	+12	+23	+12	+29	+18
18	30	+9	-4	+11	+2	+15	+2	+17	+8	+21	+8	+24	+15	+28	+15	+35	+22
30	50	+11	-5	+13	+2	+18	+2	+20	+9	+25	+9	+28	+17	+33	+17	+42	+26
50	80	+12	-7	+15	+2	+21	+2	+24	+11	+30	+11	+33	+20	+39	+20	+51	+32
80	120	+13	-9	+18	+3	+25	+3	+28	+13	+35	+13	+38	+23	+45	+23	+59	+37
120	180	+14	-11	+21	+3	+28	+3	+33	+15	+40	+15	+45	+27	+52	+27	+66	+43
180	250	+16	-13	+24	+4	+33	+4	+37	+17	+46	+17	+51	+31	+60	+31	+79	+50
250	315	+16	-16	+27	+4	+36	+4	+43	+20	+52	+20	+57	+34	+66	+34	+88	+56
316	400	+18	-18	+29	+4	+40	+4	+46	+21	+57	+21	+62	+37	+73	+37	+98	+62
400	500	+20	-20	+32	+5	+45	+5	+50	+23	+63	+23	+67	+40	+80	+40	+108	+68

DEVIATION OF HOUSING DIAMETERS 0,001 mm

Nominal diameter		J7		K6		K7		M6		M7		N6		N7		P6		P5	
over	inc	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min
6	10	+8	-7	+2	-7	+5	-10	-3	-12	0	-15	-7	-16	-4	-19	-12	-21	-9	-24
10	18	+10	-8	+2	-9	+6	-12	-4	-15	0	-18	-9	-20	-5	-23	-15	-26	-11	-29
18	30	+12	-9	+2	-11	+6	-15	-4	-17	0	-21	-11	-24	-7	-28	-18	-31	-14	-35
30	50	+14	-11	+3	-13	+7	-18	-4	-20	0	-25	-12	-28	-8	-33	-21	-37	-17	-42
50	80	+18	-12	+4	-15	+9	-21	-5	-24	0	-30	-14	-33	-9	-39	-26	-45	-21	-51
80	120	+22	-13	+4	-18	+10	-25	-6	-28	0	-35	-16	-38	-10	-45	-30	-52	-24	-59
120	180	+26	-14	+4	-21	+12	-28	-8	-33	0	-40	-20	-45	-12	-52	-36	-61	-28	-68
180	250	+30	-16	+5	-24	+13	-33	-8	-37	0	-46	-22	-51	-14	-60	-41	-70	-33	-79
250	315	+36	-16	+5	-27	+16	-36	-9	-41	0	-52	-25	-57	-14	-66	-47	-79	-36	-88
315	400	+39	-18	+7	-29	+17	-40	-10	-46	0	-57	-26	-62	-16	-73	-51	-87	-41	-98
400	500	+43	-20	+8	-32	+18	-45	-10	-50	0	-63	-27	-67	-17	-80	-55	-95	-45	-108

FITS

Tolerances for the boundary dimensions of bearings are to ISO standards. To ensure satisfactory performance of the bearing under variable operating conditions, it is necessary to select suitable fits between the inner ring and the shaft and the outer ring and the housing.

When selecting the correct fits from the ISO range of shaft and housing tolerances, it is necessary to consider adequate radial support of the bearing, ease of mounting and dismounting, and allowance for axial movement of the free bearing.

Selection of the fit also depends on the loading on the bearing and on the operating temperature. It should be noted that tight fits reduce the internal clearance of the bearing, and allowance should be made when selecting the bearing clearance.

TOLERANCES

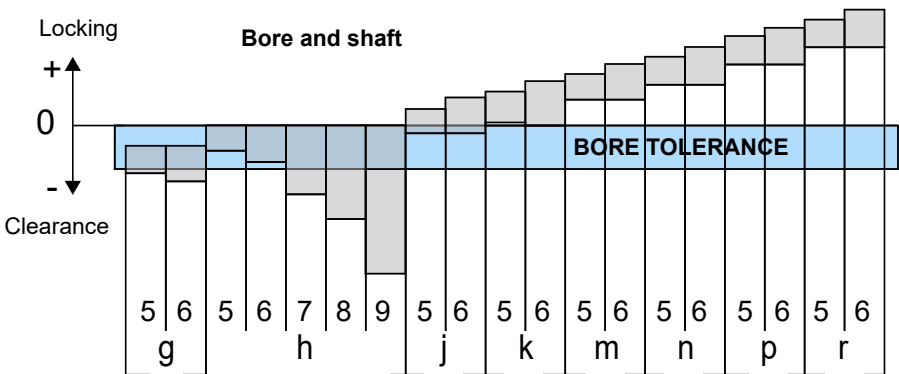
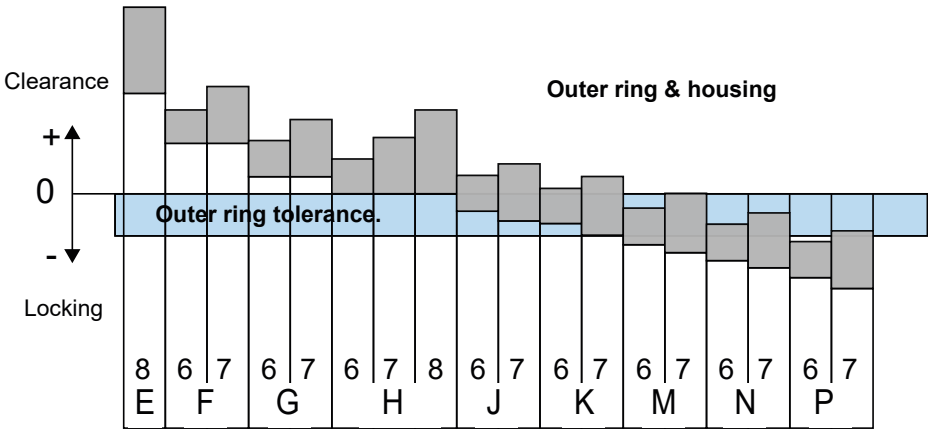
The boundary dimensions and tolerances of rolling bearings have been standardized by ISO. Bearings are manufactured to normal class 0 tolerances, unless otherwise stated. Tolerances are also listed for the closer-than-normal limits required, for example, in machine tool and high speed applications.

The more common ISO norms referred to are as follows:

ISO 15-2011	Rolling bearings – Radial bearings – Boundary dimensions
ISO 104-2002	Thrust bearings with flat housing washers – Boundary dimensions
ISO 199-2005	Rolling bearings – Thrust ball bearings – Tolerances
ISO 355-2007	Metric tapered roller bearing - Boundary dimensions
ISO 464-2002	Rolling bearings with locating snap ring – Dimensions
ISO 492-2002	Radial bearings – Tolerances
ISO 1132-2001	Rolling bearings – Tolerances – Definitions
ISO 5753-2009	Rolling bearings – Radial internal clearances



Shaft and housing fits



INTRODUCTION

The dimensional and running accuracies of our bearings are standardized according to ISO.

BORE DIAMETER

d	Nominal bore diameter
d_1	Nominal large diameter of tapered bore
d_s	Single bore diameter
Δd_s	Deviation of a single bore diameter
v_{ds}	Bore diameter variation
D_m	Mean bore diameter
Δd_m	Mean bore diameter deviation
D_{mp}	Single plane mean bore diameter
Δd_{mp}	Single plane mean bore diameter deviation
Δd_{1mp}	Deviation of mean large diameter from nominal-tapered bore
V_{DP}	Bore diameter variation in a single radial plane
V_{DMP}	Mean bore diameter variation
α	Taper angle

OUTSIDE DIAMETER

D	Nominal outside diameter
D_s	Single outside diameter
v_{Ds}	Deviation of a single outside diameter
D_m	Mean outside diameter
ΔD_m	Mean outside diameter deviation
D_{mp}	Single plane mean outside diameter
ΔD_{mp}	Single plane outside diameter deviation
V_{DP}	Outside diameter variation in a single radial plane
V_{DMP}	Mean outside diameter variation



Tolerances

ROLLING BEARING TOLERANCE SYMBOL

OUTSIDE DIAMETER

B	Nominal inner ring width
V	Nominal outer ring width
Bs	Single inner ring width
Cs	Single outer ring width
ΔB_s	Deviation of a single inner ring width
ΔC_s	Deviation of a single outer ring width
$\sqrt{B_s}$	inner ring width variation
$\sqrt{C_s}$	Outer ring width variation
Bm	Mean inner ring width

RADIAL RUN OUT

Kla	Radial run out of assembled bearing inner ring
Kea	Radial run out of assembled bearing outer ring
Sd	Face run out with bore
SD	Variation of outside surface inclination with face
Sl _a	Assembled bearing inner ring face run out with raceway
Se _a	Assembled bearing outer ring face run out with raceway
D1	Nominal diameter of outer ring flange
T	Nominal width of tapered roller bearing
ΔT_s	Deviation in width of tapered roller bearing at single position
T1	Nominal width of tapered roller bearing-cone
ΔT_{1s}	Deviation of width of tapered roller bearing-cone
T2	Nominal width of tapered roller bearing-cup
ΔT_{2s}	Deviation of width of tapered roller bearing-cup
d2	Nominal shaft washer diameter – double acting thrust bearing
Δd_{2p}	Deviation of shaft washer mean bore diameter single plane
Dw	Nominal diameter of roller
Dwm	Mean diameter of roller
Lw	Nominal length of roller

PRECISION CLASS P0 - INNER RING

d mm		Δdmp		V _{dp} Diameter ranges			V _{dmp}	K _{ia}	ΔBs				V _{Bs}
				7,8,9	0,1	2,3,4					Modified ²⁾		
over	up to	upper	lower	max.			max.	max.	upper	lower	upper	lower	max.
0,6 ¹⁾	2,5	0	-8	10	8	6	6	10	0	-40	-	-	12
2,5	10	0	-8	10	8	6	6	10	0	-120	0	-250	15
10	18	0	-8	10	8	6	6	10	0	-120	0	-250	20
18	30	0	-10	13	10	8	8	13	0	-120	0	-250	20
30	50	0	-12	15	12	9	9	15	0	-120	0	-250	20
50	80	0	-15	19	19	11	11	20	0	-150	0	-380	25
80	120	0	-20	25	25	15	15	25	0	-200	0	-380	25
120	180	0	-25	31	31	19	19	30	0	-250	0	-500	30
180	250	0	-30	38	38	23	23	40	0	-300	0	-500	30
250	315	0	-35	44	44	26	26	50	0	-350	0	-500	35
315	400	0	-40	50	50	30	30	60	0	-400	0	-500	40
400	500	0	-45	56	56	34	34	65	0	-450	0	-630	50
500	630	0	-50	63	63	38	38	70	0	-500	-	-	60
630	800	0	-75	-	-	-	-	80	0	-750	-	-	70
800	1000	0	-100	-	-	-	-	90	0	-1000	-	-	80
1000	1250	0	-125	-	-	-	-	100	0	-1250	-	-	100
1250	1600	0	-160	-	-	-	-	120	0	-1600	-	-	120
1600	2000	0	-200	-	-	-	-	140	0	-2000	-	-	140
-	-	-	-	-	-	-	-	-	-	-	-	-	-

Tolerances in μm

1) Including this dimension

2) Only for bearings mounted in sets

Tolerances

PRECISION CLASS RADIAL BEARING - METRIC SIZE - PRECISION CLASS PO

OUTER RING

D mm		ΔDmp		V _{dp2)} Open Bearings Diameter ranges			Sealed bearings	V _{dmp2)}	K _{ea}	ΔCs	VCs
				7,8,9	0,1	2,3,4	2,3,4				
over	up to	upper	lower	max.			max.	max.	max.		
2,5 ¹⁾	6	0	-8	10	8	6	10	6	10	Identical with ΔBs and VBs of the inner ring of the same bearing	
6	18	0	-8	10	8	6	10	6	10		
18	30	0	-9	12	9	7	12	6	10		
30	50	0	-11	14	11	8	16	8	13		
50	80	0	-13	16	13	10	20	9	15		
80	120	0	-15	19	19	11	26	11	20		
120	150	0	-18	23	23	14	30	15	25		
150	180	0	-25	31	31	19	38	19	30		
180	250	0	-30	38	38	23	-	23	40		
250	315	0	-35	44	44	26	-	26	50		
315	400	0	-40	50	50	30	-	30	60		
400	500	0	-45	56	56	34	-	34	65		
500	630	0	-50	63	63	38	-	38	70		
630	800	0	-75	94	94	55	-	-	80		
800	1000	0	-100	125	125	75	-	-	90		
1000	1250	0	-125	-	-	-	-	-	100		
1250	1600	0	-160	-	-	-	-	-	120		
1600	2000	0	-200	-	-	-	-	-			
2000	2500	0	-250	-	-	-	-	-	140		

Tolerances in μm

- 1) Including this dimension
- 2) Main diameter variation before fitting snap rings

PRECISION CLASS RADIAL BEARINGS WITH TAPERED BORE

Precision class P0

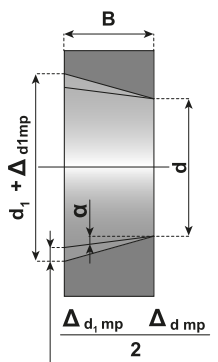
Tolerances in μm

d mm		$\Delta d_{mp}^{1)}$		$\Delta d_{1mp} - \Delta d_{mp}^{2)}$		$\sqrt{dp}^{3)}$
over	to	upper	lower	upper	lower	max.
-	10	+15	0	+15	0	10
10	18	+18	0	+18	0	10
18	30	+21	0	+21	0	13
30	50	+25	0	+25	0	15
50	80	+30	0	+30	0	19
80	120	+35	0	+35	0	25
120	180	+40	0	+40	0	31
180	250	+46	0	+46	0	38
250	315	+52	0	+52	0	44
315	400	+57	0	+57	0	50
400	500	+63	0	+63	0	56

- 1) Single plane main bore diameter deviation at smallest theoretical opening
- 2) Main diameter deviation of large diameter less main diameter deviation from smaller diameter
- 3) Bore diameter variation in a single radial plane

Tolerances For Tapered Bores – Taper 1:12

Nominal dimensions



$$\alpha = 2^\circ 23' 9.4'' = 2.38594^\circ = 0.041643 \text{ rad}$$

half the angle of taper

Tolerances

PRECISION CLASS RADIAL BEARINGS - METRIC SIZE - PRECISION CLASS P6

PRECISION CLASS P6 - INNER RING

d mm		Δdmp		V _{dp} Diameter ranges			V _{dmp}	K _{ia}	ΔBs				V _{Bs}
				7,8,9	0,1	2,3,4					Modified ²⁾		
over	up to	upper	lower	max.			max.	max.	upper	lower	upper	lower	max.
0,6 ¹⁾	2,5	0	-7	9	7	5	5	5	0	-40	-	-	12
2,5	10	0	-7	9	7	5	5	6	0	-120	0	-250	15
10	18	0	-7	9	7	5	5	7	0	-120	0	-250	20
18	30	0	-8	10	8	6	6	8	0	-120	0	-250	20
30	50	0	-10	13	10	8	8	10	0	-120	0	-250	20
50	80	0	-12	15	15	9	9	10	0	-150	0	-380	25
80	120	0	-15	19	19	11	11	13	0	-200	0	-380	25
120	180	0	-18	23	23	14	14	18	0	-250	0	-500	30
180	250	0	-22	28	28	17	17	20	0	-300	0	-500	30
250	315	0	-25	31	31	19	19	25	0	-350	0	-500	35
315	400	0	-30	38	38	23	23	30	0	-400	0	-630	40
400	500	0	-35	44	44	26	26	35	0	-450	-	-	50
500	630	0	-40	50	50	30	30	40	0	-500	-	-	60

1) Including this dimension

2) Only for bearings mounted in sets
Tolerances in μm

OUTER RING

D mm		ΔD_m p		VDp ²⁾ Open Bearings Diameter ranges			Sealed bearings	VDmp ²⁾	K _{oa}	ΔC_s	VCs
				7,8,9,	0,1	2,3,4	2,3,4				
over	up to	upper	lower	max.			max.	max.	max.	Identical with ΔB_s and $V B_s$ of the inner ring of the same bearing	
2,5 ¹⁾	6	0	-7	9	7	5	9	5	8		
6	18	0	-7	9	7	5	9	5	8		
18	30	0	-8	10	8	6	10	6	9		
30	50	0	-9	11	9	7	13	7	10		
50	80	0	-11	14	11	8	16	8	13		
80	120	0	-13	16	16	10	20	10	18		
120	150	0	-15	19	19	11	25	11	20		
150	180	0	-18	23	23	14	30	14	23		
180	250	0	-20	25	25	15	-	15	25		
250	315	0	-25	31	31	19	-	19	30		
315	400	0	-28	35	35	21	-	21	35		
400	500	0	-33	41	41	25	-	25	40		
500	630	0	-38	48	48	29	-	29	50		
630	800	0	-45	56	56	34	-	34	60		
800	1000	0	-60	75	75	45	-	45	75		

Tolerances in μm

Tolerances

PRECISION CLASS RADIAL BEARINGS - METRIC SIZE - PRECISION CLASS P5

PRECISION CLASS P5 - INNER RING

d mm		cylindrical bore			Δds		tapered bore				ΔBs	√Bs	K _{ia}	Sd	Sia
		Δdmp, Δds √dp	V _{dp} Δd1mp - Δdmp												
from	to	lower	upper	max.	lower	upper	max.	lower	upper	lower	upper	max.	max.	max.	max.
18	30	-6	0	3	0	+10	3	0	+4	-100	0	5	3	8	8
30	50	-8	0	4	0	+12	4	0	+6	-120	0	5	4	8	8
50	80	-9	0	5	0	+15	5	0	+6	-150	0	6	4	8	8
80	120	-10	0	5	0	+20	5	0	+8	-200	0	7	5	9	9
120	180	-13	0	7	0	+25	7	0	+8	-250	0	8	6	10	10
180	250	-15	0	8	0	+30	8	0	+10	-300	0	10	8	11	13
250	315	-15	0	9	0	+35	9	0	+12	-350	0	13	8	13	15
315	400	-23	0	12	0	+40	12	0	+12	-400	0	15	10	15	20
400	500	-27	0	14	0	+45	14	0	+14	-400	0	17	10	17	23

Tolerances in μm

PRECISION CLASS P5 - OUTER RING

D mm		$\Delta D_{mp}, \Delta D_s$		V_{dp}	K_{ea}	SD	Sea
from	to	lower	upper	max.	max.	max.	max.
30	50	-7	0	4	5	8	8
50	80	-9	0	5	5	8	10
80	120	-10	0	5	6	9	11
120	150	-11	0	6	7	10	13
150	180	-13	0	7	8	10	14
180	250	-15	0	8	10	11	15
250	315	-18	0	9	11	13	18
315	400	-20	0	10	13	13	20
400	500	-23	0	12	15	15	23
500	630	-28	0	14	17	18	25
630	800	-35	0	18	20	20	30

Tolerances in μm

PRECISION CLASS P4 - INNER RING

d mm		cylindrical bore			Δds		tapered bore				ΔBs	√Bs	K _{ia}	Sd	Sia
		Δdmp, Δds V _{dp}					V _{dp} Δd1mp - Δdmp								
from	to	lower	upper	max.	lower	upper	max.	lower	upper	lower	upper	max.	max.	max.	max.
18	30	-5	0	2.5	0	+6	2.5	0	+2	-25	0	1.5	1.5	3	3
30	50	-6	0	3	0	+7	3	0	+3	-30	0	2	2	3	3
50	80	-7	0	3.5	0	+8	3.5	0	+3	-40	0	3	2	4	3
80	120	-8	0	4	0	+10	4	0	+4	-50	0	3	3	4	4
120	180	-10	0	5	0	+12	5	0	+4	-60	0	4	3	5	6
180	250	-12	0	6	0	+14	6	0	+5	-75	0	5	4	6	7
250	315	-15	0	8	0	+15	8	0	+6	-100	0	5	4	6	8
315	400	-19	0	10	0	+17	10	0	+6	-100	0	6	5	7	9
400	500	-23	0	12	0	+19	12	0	+7	-100	0	7	5	8	10

Tolerances in μm

PRECISION CLASS UP - OUTER RING

D mm		$\Delta Dmp, \Delta Ds$		V_{dp}	K_{ea}	SD	Sea
from	to	lower	upper	max.	max.	max.	max.
30	50	-5	0	3	3	2	4
50	80	-6	0	3	3	2	4
80	120	-7	0	4	3	3	5
120	150	-8	0	4	4	3	6
150	180	-9	0	5	4	3	7
180	250	-10	0	5	5	4	9
250	315	-12	0	6	6	4	9
315	400	-14	0	7	7	5	12
400	500	-17	0	9	8	5	12
500	630	-20	0	10	9	6	14
630	800	-25	0	13	11	7	17

Tolerances in μm

Tolerances

PRECISION CLASS TAPERED ROLLER BEARING - METRIC SIZE

PRECISION CLASS P0

PRECISION CLASS P0 - INNER RING

d mm		ΔD_{mp}		$\sqrt{dp}$	$\sqrt{dmp}$	K_{ia}
over	up to	upper	lower	max.	max.	max.
10	18	0	-12	12	9	15
18	30	0	-12	12	9	18
30	50	0	-12	12	9	20
50	80	0	-15	15	11	25
80	120	0	-20	20	15	30
120	180	0	-25	25	19	35
180	250	0	-30	30	23	50
250	315	0	-35	35	26	60
315	400	0	-40	40	30	70

Tolerances in μm

OUTER RING

D mm		ΔD_{mp}		$\sqrt{Dp}$	$\sqrt{Dmp}$	K_{ea}
over	up to	upper	lower	max.	max.	max.
18	30	0	-12	12	9	18
30	50	0	-14	14	11	20
50	80	0	-16	16	12	25
80	120	0	-18	18	14	35
120	150	0	-20	20	15	40
150	180	0	-25	25	19	45
180	250	0	-30	30	23	50
250	315	0	-35	35	26	60
315	400	0	-40	40	30	70
400	500	0	-45	45	34	80
500	630	0	-50	50	38	100

Tolerances in μm

Note: the limit tolerance of the outer diameter D1 of a flanged bearing is h9

PRECISION CLASS P0 - WITH OF INNER, OUTER RING AND HEIGHT

d mm		$\Delta BS, \Delta CS$		ΔTS		$\Delta T1S$		$\Delta T2S$	
over	up to	upper	lower	upper	lower	upper	lower	upper	lower
10	18	0	-120	+120	0	+100	0	+100	0
18	30	0	-120	+200	0	+100	0	+100	0
30	50	0	-120	+200	0	+100	0	+100	0
50	80	0	-150	+200	0	+100	0	+100	0
80	120	0	-200	+200	-200	+100	-100	+100	-100
120	180	0	-250	+350	-250	+150	-150	+200	-100
180	250	0	-300	+350	-250	+150	-150	+200	-100
250	315	0	-350	+350	-250	+150	-150	+200	-100
315	400	0	-400	+400	-400	+200	-200	+200	-200

Tolerances in μm

PRECISION CLASS P6X

D mm		ΔBs		ΔCs		ΔTs		$\Delta T1s$		$\Delta T2s$	
over	up to	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
10	18	0	-50	0	-100	+100	0	+50	0	+50	0
18	30	0	-50	0	-100	+100	0	+50	0	+50	0
30	50	0	-50	0	-100	+100	0	+50	0	+50	0
50	80	0	-50	0	-100	+100	0	+50	0	+50	0
80	120	0	-50	0	-100	+100	0	+50	0	+50	0
120	180	0	-50	0	-100	+150	0	+50	0	+100	0
180	250	0	-50	0	-100	+150	0	+50	0	+100	0
250	315	0	-50	0	-100	+200	0	+100	0	+100	0
315	400	0	-50	0	-100	+200	0	+100	0	+100	0

Tolerances in μm

The limit tolerances for the diameter and the radial run-out of the outer ring and the inner ring in this precision class are the same with precision class P0.

The limit tolerances for the width and mounting height for the outer and inner ring are those indicated below.

Tolerances

PRECISION CLASS TAPERED ROLLER BEARINGS - METRIC SIZE

PRECISION CLASS P4

PRECISION CLASS P5 - INNER RING

d mm		Δd_{mp}		V_{dp}	VD_{MP}	K_{ia}	S_d	ΔB_s		ΔT_s	
over	up to	upper	lower	max.	max.	max.	max.	upper	lower	upper	lower
10	18	0	-7	5	5	5	7	0	-200	+200	-200
18	30	0	-8	6	5	5	8	0	-200	+200	-200
30	50	0	-10	8	5	6	8	0	-240	+200	-200
50	80	0	-12	9	6	7	8	0	-300	+200	-200
80	120	0	-15	11	8	8	9	0	-400	+200	-200
120	180	0	-18	14	9	11	10	0	-500	+350	-250
180	250	0	-22	17	11	13	11	0	-600	+350	-250

Tolerances in μm

OUTER RING

D mm		ΔD_{mp}		V_{dp}	VD_{MP}	K_{ea}	SD	ΔC_s
over	up to	upper	lower	max.	max.	max.	max.	
18	30	0	-8	6	5	6	8	Identical with ΔB_s of the inner ring of the same bearing
30	50	0	-9	7	5	7	8	
50	80	0	-11	8	6	8	8	
80	120	0	-13	10	7	10	9	
120	150	0	-15	11	8	11	10	
150	180	0	-18	14	9	13	10	
180	250	0	-20	15	10	15	11	
250	315	0	-25	19	13	18	13	
315	400	0	-28	22	14	20	13	

Tolerances in μm

The limit tolerance of the outer diameter D_1 of a flanged bearing is h9.

PRECISION CLASS P4 - INNER RING

d mm		$\Delta d_{mp}, ds$		V_{dp}	VD_{MP}	K_{ia}	Sd	Sia	ΔBs		ΔTs	
over	up to	upper	lower	max.	max.	max.	max.	max.	upper	lower	upper	lower
10	18	0	-5	4	4	3	3	3	0	-200	+200	-200
18	30	0	-6	5	4	3	4	4	0	-200	+200	-200
30	50	0	-8	6	5	4	4	4	0	-240	+200	-200
50	80	0	-9	7	5	4	4	4	0	-300	+200	-200
80	120	0	-10	8	5	5	5	5	0	-400	+200	-200
120	180	0	-13	10	7	6	7	7	0	-500	+350	-250
180	250	0	-15	11	8	8	8	8	0	-600	+350	-250

Tolerances in μm

OUTER RING

D mm		$\Delta D_{mp}, \Delta Ds$		V_{dp}	VD_{MP}	K_{ea}	SD	Sea	ΔCs
over	up to	upper	lower	max.	max.	max.	mas.	max.	Upper / lower
18	30	0	-6	5	4	4	4	5	Identical with ΔBs of the inner ring of the same bearing
30	50	0	-7	5	5	5	4	5	
50	80	0	-9	7	5	5	4	5	
80	120	0	-10	8	5	6	5	6	
120	150	0	-11	8	6	7	5	7	
150	180	0	-13	10	7	8	5	8	
180	250	0	-15	11	8	10	7	10	
250	315	0	-18	14	9	11	8	10	
315	400	0	-20	15	10	13	10	13	

Tolerances in μm

Tolerances

PRECISION CLASS TAPERED ROLLER BEARINGS - INCH SIZE – PRECISION

ACCORDING AFBMA STANDARD

INNER RING

d mm		Precision classes									
		4		2		3		0		00	
over	up to	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
-	76.2	+13	0	+13	0	+13	0	+13	0	+8	0
76.2	304.2	+25	0	+25	0	+13	0	+13	0	+8	0
304.2	609.6	+51	0	+51	0	+25	0	-	-	-	-
609.6	914.4	+76	0	-	-	+38	0	-	-	-	-
914.4	1219.2	+102	0	-	-	+51	0	-	-	-	-
1219.2	-	+127	0	-	-	+76	0	-	-	-	-

Tolerances in μm

OUTER RING

D mm		Precision classes									
		4		2		3		0		00	
over	up to	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
-	304.8	+25	0	+25	0	+13	0	+13	0	+8	0
304.8	609.6	+51	0	+51	0	+25	0	-	-	-	-
609.6	912.4	+76	0	+76	0	+38	0	-	-	-	-
914.4	1219.2	+102	0	-	-	+51	0				
1219.2	-	+127	0	-	-	+76	0	-	-	-	-
-	-	-	-	-	-	-	-	-	-	-	-

Tolerances in μm

TOLERANCES FOR THE MOUNTING HEIGHT ON THE INNER RING ROLLER SUB-ASSEMBLY WITH OUTER RING ($\Delta T1S$)

d mm		D mm		Precision classes									
				4		2		3		0		00	
over	up to	over	up to	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
-	101.6			+102	0	+102	0	+102	-102	+102	-102	+102	-102
101.6	304.8			+152	-152	+102	0	+102	-102	+102	-102	+102	-102
304.8	609.6		508.0	+178	-178	+178	-178	+102	-102	-	-	-	-
304.8	609.6	508.0	-	+178	-178	+178	-178	+178	-178	-	-	-	-
609.6	-			+178	-178	-	-	+178	-178	-	-	-	-

Tolerances in μm

RADIAL RUN-OUT OF AN ASSEMBLED BEARINGS

d mm		Precision classes				
		4	2	3	0	00
over	up to	max.	max.	max.	max.	max.
-	304.8	51	38	8	4	2
304.8	609.6	51	38	18	-	-
609.6	914.4	76	51	51	-	-
914.4	-	76	-	76	-	-

Tolerances in μm

TOLERANCES FOR MOUNTING HEIGHT ΔT_s

d mm		D mm		Precision classes									
				4		2		3		0		00	
over	up to	over	up to	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
-	101.6			+203	0	+203	0	+203	-203	+203	-203	+203	-203
101.6	304.8			+356	-254	+203	0	+203	-203	+203	-203	+203	-203
304.8	609.6	-	508.0	+318	-381	+381	-381	+203	-203	-	-	-	-
304.8	609.6	508.0		+318	-381	+318	-381	+381	-381	-	-	-	-
609.6	-			+318	-381	-	-	+381	-381	-	-	-	-

Tolerances in μm

TOLERANCES FOR THE MOUNTING HEIGHT ON THE INNER RING ROLLER SUB-ASSEMBLY WITH OUTER RING (ΔT_{2S})

d mm		D mm		Precision classes									
				4		2		3		0		00	
over	up to	over	up to	upper	lower	upper	lower	upper	lower	upper	lower	upper	lower
-	101.6			+102	0	+102	0	+102	-102	+102	-102	+102	-102
101.6	304.8			+203	-102	+102	0	+102	-102	+102	-102	+102	-102
304.8	609.6		508.0	+203	-203	+203	-203	+102	-102	-	-	-	-
304.8	609.6	508.0	-	+203	-203	+203	-203	+203	-203	-	-	-	-
609.6	-			+203	-203	-	-	+203	-203	-	-	-	-

Tolerances in μm

Tolerances

PRECISION CLASS THRUST BEARING

SINGLE ACTING THRUST BEARING (SINGLE ROW)

SHAFT WASHER

d mm		P0; P6 ; P5			P4; P2		
		Δd_{mp} Δd_{2mp}		V_{dp} VD_{2P}	Δd_{mp} Δd_{2mp}		V_{dp} VD_{2P}
over	up to	upper	lower	max.	upper	lower	max.
-	18	0	-8	6	0	-7	5
18	30	0	-10	8	0	-8	6
30	50	0	-12	9	0	-10	8
50	80	0	-15	11	0	-12	9
80	120	0	-20	15	0	-15	11
120	180	0	-25	19	0	-18	14
180	250	0	-30	23	0	-22	17
250	315	0	-35	26	0	-25	19
315	400	0	-40	30	0	-30	23
400	500	0	-45	34	0	-35	26
500	630	0	-50	38	0	-40	30
630	800	0	-75	-	0	-50	-
800	1000	0	-100	-	-	-	-
1000	1250	0	-125	-	-	-	-

Tolerances in μm

HOUSING WASHER

d mm		P0; P6 ; P5			P4; P2		
		VD_{MP}		V_{dp}	Δd_{mp}		V_{dp}
over	up to	upper	lower	max.	upper	lower	max.
10	18	0	-11	8	0	-7	5
18	30	0	-13	10	0	-8	6
30	50	0	-16	12	0	-9	7
50	80	0	-19	14	0	-11	8
80	120	0	-22	17	0	-13	10
120	180	0	-25	19	0	-15	11
180	250	0	-30	23	0	-20	15
250	315	0	-35	26	0	-25	19
315	400	0	-40	30	0	-28	21
400	500	0	-45	34	0	-33	25
500	630	0	-50	38	0	-38	29
630	800	0	-75	55	0	-45	34
800	1000	0	-100	75	0	-	-
1000	1250	0	-125	-	0	-	-
1250	1600	0	-160	-	0	-	-

Tolerances in μm

DOUBLE ACTING THRUST BEARING (DOUBLE ROW)

PRECISION CLASS P5 AND P6 - INNER RING

d mm		P5					P6			P5 and P6			
		Δd_{mp}		Δd_s		S_{ia}	Δd_s		S_{ia}	H_s		C_s	
over	up to	upper	lower	upper	lower	max.	upper	lower	max.	upper	lower	upper	lower
16	18	0	-7	+1	-8	3	0	-5	1.5	+50	-80	0	-30
18	30	0	-8	+1	-9	3	0	-6	1.5	+50	-80	0	-30
30	50	0	-10	+1	-11	3	0	-8	1.5	+60	-100	0	-30
50	80	0	-12	+2	-14	4	0	-9	2	+70	-120	0	-30
80	120	0	-15	+3	-18	4	0	-10	2	+85	-140	0	-30
120	180	0	-18	+3	-21	5	0	-13	3	+95	-160	0	-30
180	250	0	-22	+4	-26	5	0	-15	3	+120	-200	0	-30

PRECISION CLASS P5 AND P6 - OUTER RING

D mm		SP and UP		
		ΔD_S	S_{ea}	
over	up to	upper	lower	max.
30	50	-20	-27	Identical with the inner ring of the same bearing
50	80	-24	-33	
80	120	-28	-38	
120	150	-33	-44	
150	180	-33	-46	
180	250	-37	-52	
250	315	-41	-59	

SINGLE & DOUBLE ACTING THRUST BEARING

d mm		Si					Se
		P0;	P6;	P5;	P4;	P2	P0; P6; P5; P4; P2
over	up to	max.	max.	max.	max.	max.	max.
-	18	10	5	3	2	1	Identical with S1 of the shaft washer
18	30	10	5	3	2	1.2	
30	50	10	6	3	2	1.5	
50	80	10	7	4	3	2	
80	120	15	8	4	3	2	
120	180	15	9	5	4	3	
180	250	20	10	5	4	3	
250	315	25	13	7	5	4	
315	400	30	15	7	5	4	
400	500	30	18	9	6	-	
500	630	35	21	11	7	-	
630	800	40	25	13	8	-	
800	1000	45	30	15	-	-	
1000	1250	50	35	18	-	-	

Tolerances in μm

GENERAL GUIDELINES, ROLLING BEARING LUBRICATION

The main duties of introducing lubricants into ball and roller bearings, apart from protecting the finely finished surfaces when rotating at high speeds, is to reduce friction between the rolling elements, the separator or cage, and the races at any point where true rolling is absent. Lubrication also assists in dissipating heat, and sealing the bearing against the entry of contaminants such as dust and moisture.

Rolling bearings may be lubricated with oil or grease. The choice of lubricant is usually decided by temperature, speed, load, and operating conditions along with bearing design. We summarize as follows:

1. SIZE OF BEARING

The size of bearing governs the viscosity of the lubricant – the larger the bearing the higher should be the viscosity. Regarding size, rolling bearings can be divided into four sizes depending on the outside diameter:

- Very small bearings $D \leq 22 \text{ mm}$
- Small bearings $D \leq 62 \text{ mm}$
- Medium bearings $62 \leq D \leq 240 \text{ mm}$
- Large bearings $D \geq 240 \text{ mm}$

2. SPEED

Speed has an influence upon the viscosity of the lubricant because the resisting force, opposed to the moving parts by the lubricant, depends on its viscosity. The higher the revolution speed, the lower the viscosity of the lubricant should be.

The revolution speed may be:

- Normal $n \leq 75\%$ of the limit speed specified in the tables
- High $75\% \leq 100\%$ of the limit speed specified in the tables
- Very high $n \geq 100\%$ of the limit speed specified in the tables. For very high revolution speeds, oil lubrication is required to transfer frictional heat or other sources of heat away from the bearing.

3. LOAD

Equivalent loading capacity, $P = XFr + YFa$, conditions the viscosity grade of the lubricant, due to specific pressure which appears between the contact surfaces. The higher this is, the greater the resistance of the lubricant film should be, and the respective viscosity. Loads may vary as follows:

- normal loads where
- $P/Cr \leq 0.1$ for bearings within diameter ranges 1, 2 and 3.
- $P/Cr \leq 0.15$ for bearings within diameter range 4.
- high loads where
- $P/Cr \geq 0.1$ for bearings within diameter ranges 1, 2 and 3.
- $P/Cr \geq 0.15$ for bearings within diameter range 4.
- P = equivalent dynamic load [kN]
- Cr = basic dynamic load [kN]

4. TEMPERATURE

The operating temperature affects selection of lubricants, as it is an influence upon viscosity. Therefore, each lubricant is used only within the limits of certain clearly defined temperature ranges.



Lubrication

GREASE & OIL LUBRICATION

Although oil is the better lubricant, grease is often preferred because of the following natural advantages:

- Grease helps to form an effective closure between the shaft and housing, thus preventing the ingress of dirt, moisture and other corrosive agents.
- Grease protects the finely finished working surfaces of a bearing by clinging to them, particularly when the bearing is not in motion. Oil tends to drain away, leaving the surfaces open to attack.
- Grease is easier to retain within the housing than oil. This is of great help in the food, printing, textile, chemical and other industries where contamination or staining can ruin the product.
- Grease is convenient to handle, and re-lubrication of bearings is quick and clean. Planned lubrication cycles are often possible, resulting in smaller labour costs.
- Whatever type of grease is used, it should have no tendency to separate under operating conditions. When separation occurs, the oil runs out of the bearing and leaves behind dry soap, which hardens and cakes. This interferes with the movement of the rolling element, which may result in overheating and mechanical failure. Excessive softening is also undesirable, because the grease might then leak out of the bearing and leave working surfaces unprotected.

The quantity of grease used for the lubrication of a rolling bearing should not be too great, as a tightly packed bearing is liable to overheat if operated at high speed. Relubrication intervals depend on the bearing type, inner diameter and revolution speed for filling with fresh grease. The quantity required is given in the following equation:

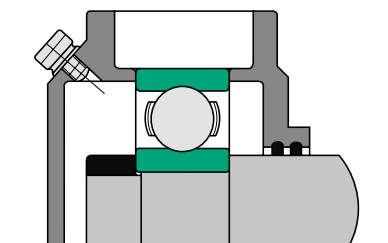
$$G = 0.005 \text{ DB grams}$$

where:

D = outer diameter (m/m)

B = width (m/m)

After a certain number of refills, it is necessary to remove old grease completely using a suitable solvent. Never mix two grades of grease.



Typical Grease Lubrication

OIL LUBRICATION

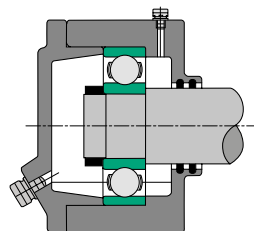
Oil is sometimes more convenient to use than grease, and there are circumstances when it is definitely preferred. These are as follows:

- When frictional resistance in light machinery and instruments must be kept low.
- Where either the speed or the temperature is too high for grease lubrication.
- Where high temperature and heavy load occur together, with or without high speed.
- Where the bearings are enclosed in a casing that contains other components lubricated by oil, e.g. a gearbox.

A good quality mineral oil should be used of a viscosity to suit the operating conditions involved. Vegetable or animal oils are not recommended as these can become rancid under certain conditions and cause corrosion problems. A small supply of oil is required to lubricate the bearings; a more copious supply should be used if the bearing must be kept cool, when it is often advantageous to use a synthetic oil to cope with the temperature conditions. Limiting temperature for mineral oil is about 150°C (302°F) and for synthetic oils about 220°C (428°F).

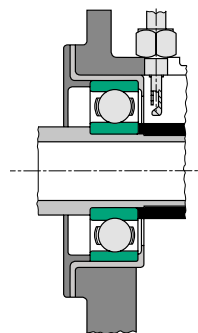
OIL BATH

This method is suitable for horizontal shaft applications. The oil should reach the center of the bottom ball or roller in the bearing; a greater depth than this could cause overheating due to churning of the oil. The surface area and volume of the oil in the bath should be sufficiently large to maintain an adequate depth of oil for the cage to dip into when running. Sight oil level indicators can be used. Alternatively, a tapped and plugged hole can be provided at the correct level; when replenishing the oil, the plug is removed and oil added until it starts to escape through the hole. The plug should, of course, be replaced before the machine is started!



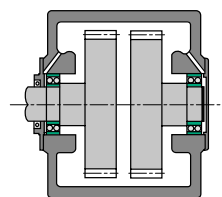
PUMP-FEED LUBRICATION

This is especially suitable for heavily loaded, high-speed bearings, since such conditions can result in bearing temperatures well in excess of 100°C (212°F). Oil is pump-fed to each bearing, being directed by jets on to the outside diameter of the inner ring so that some of it gains access to the internal parts of the bearing. Each bearing may require from 45 to 140 litres of oil per hour, although most of this only flushes the face of the bearing to keep it cool. A reservoir is often provided to lubricate the bearing during starting; alternatively, the pump can be started before the machine is set in motion so that the bearing never runs unlubricated.



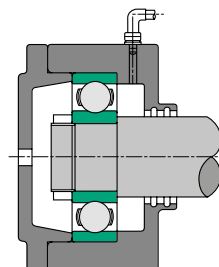
SPLASH LUBRICATION

This is suitable when the bearings are enclosed in a casing, such as a gearbox, and the oil used to lubricate the gears is distributed sufficiently to lubricate the bearings. The oil is either splashed directly on to the bearings or collected in galleries and directed to the bearings. The drawing below illustrates how the oil is made to pass through the bearing before it returns to the gearbox casing.



OIL MIST SYSTEM

One important advantage of this method of lubrication is that only a small quantity of oil is required, carried in a stream of compressed air. The oil mist equipment should be turned on before the machine is set in motion so as to ensure that the bearings are constantly covered by a thin film of oil when rotating. Oil mist is advantageous for applications such as machine tools (where it can also be used to lubricate slideways, gears, chains and other components), since the air escaping from the bearing housing prevents the ingress of foreign matter. The flow of air also keeps the bearing cool. It is important that the compressed air used is absolutely clean and dry.

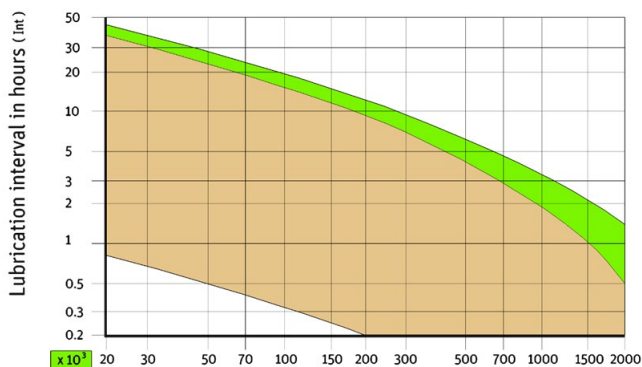


Lubrication

LUBRICATION INTERVAL

The below table gives an estimation of the lubrication interval. Calculate the Kr value using the below formula and place it on the x axis. The crossing point with the upper green zone will indicate the maximum interval in hours. Then based on operating condition K1, K2, K3 & K4 adjust the interval.

$$\text{Lub interval} = \text{Int} \times K1 \times K2 \times K3 \times K4 \quad \text{operating hours}$$



k _r × n × d _m (rpm × mm)											
Type	DGB	ACB1	ACB2	SAB	TB	CRB1	CRB2	CRB FC	TRB	SRB	SRTB
Kr	1	1.5	1.9	1.3	5	1.75	1.9	23	3.5	6	12
Cleanliness & Moisture					LOW	HIGH	VERY HIGH				
K1					0.9	0.65	0.4				
Shocks & Vibrations					LOW	HIGH	VERY HIGH				
K2					0.9	0.65	0.4				
Operating temperature					< 70°	< >	> 90°				
K3					0.9	0.65	0.4				
Loading		P/C			0.10	0.22	0.35				
K4					0.9	0.65	0.4				

Typical: For Crushers: 1 x day – 1 x week For Oscillating movement: reduced interval

Example: DGB: Kr = 1, dm = 120 mm, n = 1000 rpm => Int = 1 x 120 x 1000 = 120000 => 19000 h

Best condition: $0.9 \times 0.9 \times 0.9 \times 0.9 = 0.650 \times \text{Int} = 12350 \text{ h}$ /Worse condition: $0.4 \times 0.4 \times 0.4 \times 0.4 = 0.025 \times \text{Int} = 475 \text{ h}$