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:

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

$L_{10h} = \frac{1000000}{60n} \left(\frac{C_r}{P}\right)^p$, 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 o fthe following form:

$L_{na} = a_1 a_2 a_3 \left(\frac{C_r}{P}\right)^p$ 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 execessive lubricant contamination wtih hard particles or poor oil circulation, a_{23} shall be taken to be 0.1.

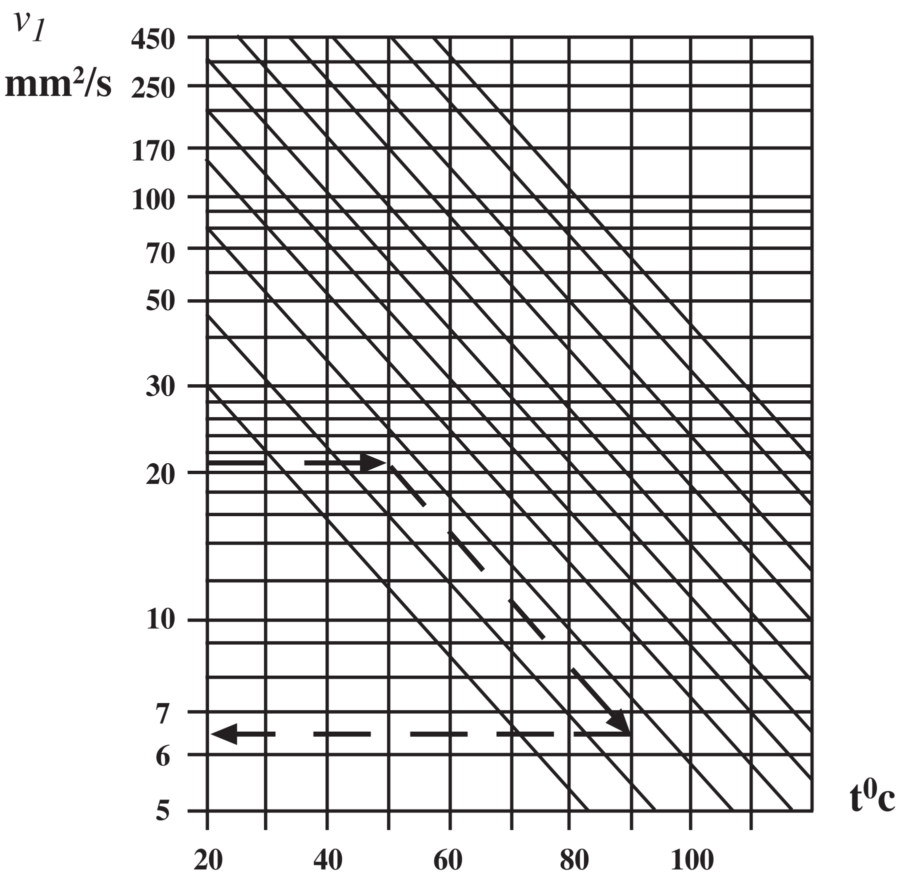

Fig. 1. A nomograph chart to determine lubricant viscosity at operating temperatures when its viscosity at basic temperature is known

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, $mm^2.s^{-1}$;

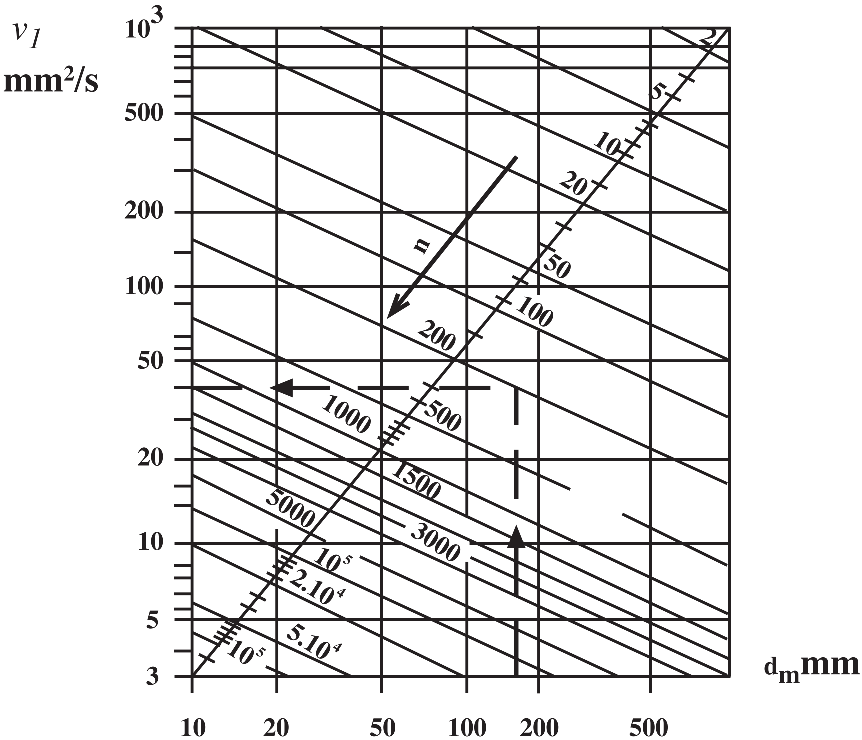

Fig. 2. A nomograph chart to determine normatice lubricant viscosity, v_1

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

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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:

$$P_r = XF_r + YF_a \quad \text{....... 2.5}$$

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,

$$P_r = F_r \quad \text{....... 2.6}$$

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:

$$P_a = XF_r + YF_a \quad \text{....... 2.7}$$

while for a thrust bearing it has the following form:

$$P_a = F_a \quad \text{....... 2.8}$$

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:

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

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

$p = \frac{10}{3}$ 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:

$F=mrw^2, H,$ 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.

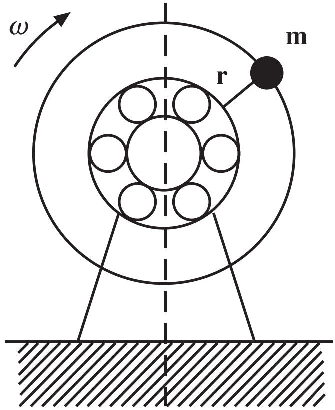

Fig. 3. Diagram of loading a bearing with rotational force.

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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 mechanisms 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-power ventilators and exhausters.

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:

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

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_o). Deviations from this rule are based on experimental data. Thus, if the notion of the static

safety coefficient S_o $\left(S_o = \frac{C_0}{P_0}\right)$ 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 $s = 1{,}5\text{-}2{,}5$.

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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)