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 dependent on the structural features of the machine in question.

The bearing 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 characteristic of a bearing - the basic dynamic load-carrying rating, denoted with symbol \( C_r \) - 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 \( n > 20 \text{ min}^{-1} \) is calculated with the formula:

\[
L_{10} = \left( \frac{C_r^p}{P} \right) \text{ million rotations} \quad \ldots \quad 2.1
\]

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: \( P = 3 \);
\( P = \frac{10}{3} \) for roller bearings.

The basic rated resource is mainly expressed in operating hours:

\[
L_{10h} = \frac{1000000}{60n} \left( \frac{C_r^p}{P} \right), \text{ hour} \quad \ldots \quad 2.2
\]

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:

\[
L_{10} = \frac{\pi D_1}{1000} L_{10}
\]

Where \( L_{10} \) – 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 service life \( (L_{50}) \) is, as a rule, five times as that of the basic rated resource \( (L_{10}) \).

To improve the compactness of bearing units and to reduce their weight, it is not recommended to overestimate the basic rated resource. However, in a number of technical fields another level of reliability is required. Besides, due to the extensive research and development activity, it has been found that the conditions of lubrication greatly affect the bearing service life. Hence, ISO has introduced a notation of basic rated resource, the formula of which is of the following form:

\[
L_{n} = a_1 a_2 a_3 \left( \frac{C_r^p}{P} \right) \text{ or} \quad \ldots \quad 2.3
\]

\[
L_{n} = a_1 a_2 a_3 L_{10}
\]

Where \( L_n \) – is the adjusted rated resource/million revolutions, \( C_r \) is the basic dynamic load-carrying capacity,
\( P \) – is the exponent of a power, for ball bearings: \( P = 3 \);
\( P = \frac{10}{3} \) for roller bearings.

\( a_1 \) – 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 lubrication 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

<table>
<thead>
<tr>
<th>Reliability, percent</th>
<th>L_{10a}</th>
<th>a_{1}</th>
</tr>
</thead>
<tbody>
<tr>
<td>90</td>
<td>L_{10a}</td>
<td>1</td>
</tr>
<tr>
<td>95</td>
<td>L_{5a}</td>
<td>0.62</td>
</tr>
<tr>
<td>96</td>
<td>L_{4a}</td>
<td>0.53</td>
</tr>
<tr>
<td>97</td>
<td>L_{3a}</td>
<td>0.44</td>
</tr>
<tr>
<td>98</td>
<td>L_{2a}</td>
<td>0.33</td>
</tr>
<tr>
<td>99</td>
<td>L_{1a}</td>
<td>0.21</td>
</tr>
</tbody>
</table>

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 from Table 2.

Table 2.2 Factors a_{23}

<table>
<thead>
<tr>
<th>Type of Bearing</th>
<th>Vacuum Treated Steel</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Values of Viscosity Coefficient c = n / n_{1}</td>
</tr>
<tr>
<td></td>
<td>0.1-0.2</td>
</tr>
<tr>
<td>Radial and Angular Contact Ball Bearings</td>
<td>0.1-0.3</td>
</tr>
<tr>
<td>Roller Spherical Bearings, Double-row</td>
<td>0.1-0.2</td>
</tr>
<tr>
<td>Roller Bearings, with Short Cylindrical Rollers or Needles</td>
<td>0.1-0.4</td>
</tr>
<tr>
<td>Spherical Roller Angular Contact Thrust Bearings</td>
<td>0.1-0.2</td>
</tr>
</tbody>
</table>

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 excessive lubricant contamination with hard particles or poor oil circulation, a_{23} shall be taken to be 0.1.
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 $a_{23}$.

Fig. 1. A nomograph chart to determine lubricant viscosity at operating temperatures when its viscosity at basic temperature is known.
The factor $a_{ij}$ is selected from Table 3, by the ratio of normative and actual kinematic viscosity of the lubricant used:

$$\chi = \frac{v}{v_1} \quad \cdots \quad 2.4$$

Where $\chi$ – 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 viscosity of oil as required to ensure lubrication conditions at a given velocity, mm$^2$·s$^{-1}$;
### Bearing Data

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

<table>
<thead>
<tr>
<th>Machine Type and Employment</th>
<th>$L_{10h}$, hr</th>
<th>$L_{10h}$, mln, km</th>
</tr>
</thead>
<tbody>
<tr>
<td>Devices and mechanisms used at regular intervals, agricultural machines, household appliances</td>
<td>500-4000</td>
<td>-</td>
</tr>
<tr>
<td>Mechanisms used for a short periods of time, erecting cranes, building machines</td>
<td>4000-8000</td>
<td></td>
</tr>
<tr>
<td>Critical mechanisms used intermittently (accessory mechanisms at power plant stations, conveyors for series production, elevators, metal-cutting machine tools used from time to time)</td>
<td>8000-12000</td>
<td></td>
</tr>
<tr>
<td>One-shift operated machines, underloaded (stationery electric motors, reduction gears, crushers (mills))</td>
<td>12000-20000</td>
<td></td>
</tr>
<tr>
<td>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, centrifuges, polygraph equipment.</td>
<td>20000-30000</td>
<td></td>
</tr>
<tr>
<td>Machines to be used on a round-the-clock basis (compressors, pumps, mine lifters, stationery electric motors, equipment used in textile industry)</td>
<td>40000-50000</td>
<td></td>
</tr>
<tr>
<td>Hydropower stations, rotary furnaces, deep-sea vessels engines</td>
<td>60000-100000</td>
<td></td>
</tr>
<tr>
<td>Continuous-operation heavy-duty machines (paper working equipment, power plants, mine pumps, flexible shafts of deep-sea vessels)</td>
<td>100000</td>
<td></td>
</tr>
<tr>
<td>Wheel-hubs of cars</td>
<td>0.2-0.3</td>
<td></td>
</tr>
<tr>
<td>Wheel-hubs of buses, industrial-type vehicles</td>
<td>0.3-0.5</td>
<td></td>
</tr>
<tr>
<td>Railway freight-car journal boxes</td>
<td>0.8</td>
<td></td>
</tr>
<tr>
<td>Suburban car and tram journal boxes</td>
<td>1.5</td>
<td></td>
</tr>
<tr>
<td>Passenger-car journal boxes</td>
<td>3.0</td>
<td></td>
</tr>
<tr>
<td>Locomotive journal boxes</td>
<td>3.0-5.0</td>
<td></td>
</tr>
</tbody>
</table>
The values of the kinematic viscosity of oil, i.e., the operating viscosity, is determined with the help of a nomographic chart, Fig 1. To obtain the operating viscosity, it is necessary to know the bearing temperature and the initial kinematic viscosity of the oil used. For example, to calculate the standard viscosity of oil, \( \nu_1 \), for a bearing with a rotational speed of \( n = 200 \text{ min}^{-1} \) and a mean diameter of \( d_m = 150 \text{ mm} \), it is necessary to find the X-axis of mean diameters-to pass over to the Y-axis the respective value of \( \nu_1 \) (\( \nu_1 = 44 \text{ mm}^2\text{s}^{-1} \) 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 abovemention 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 \( F/F_a \leq e \), is assumed,

\[
P_r = F, \quad \text{...... 2.6}
\]

Where \( e \) is the limited value of \( F/F_a \), 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:

\[
P_a = XF, + YF, \quad \text{...... 2.7}
\]

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 known. 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 supported beam resting in bearing supports. Using the momental equation and those for the sum of forces acting upon the beam, and the 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 thot variety of force factors and application of incidental forces. Hence, any mathematical 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 \( P_{\text{min}} \) to \( P_{\text{max}} \), e.g., at the supports of single-sided winding drums, then, the value of \( F \) has the form:

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

If operating duties are of a varying nature, i.e., load \( F_i \) acts within the period \( t_i \), at a rotational speed \( n_i \), while during the period \( t_b \), at a rotation speed \( n_b \) acts the load \( F_b \) and so on, then, the amount \( F \) takes the form:
**Equivalent Dynamic Load Calculation**

\[ F = \left( \frac{n_1 F_{1r} + n_2 F_{2r} + \ldots + n_i F_{ir}}{n_1 F_{1r} + n_2 F_{2r} + \ldots + n_i F_{ir}} \right)^{\frac{1}{p}} \] \quad \text{...... 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 above-mentioned relationships is valid not only for radial loads but, also, for any load of constant direction 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 the radial plane of the 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:

\[ F = m r w^2, \quad H, \quad \text{...... 2.11} \]

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.
### 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**

<table>
<thead>
<tr>
<th>Type of Loading</th>
<th>$K_6$</th>
<th>Field of Application</th>
</tr>
</thead>
<tbody>
<tr>
<td>Light jerks, short-time overloads up to 125% of the rated (nominal) load</td>
<td>1.0-1.2</td>
<td>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 blowers.</td>
</tr>
<tr>
<td>Moderate jerks, vibratory load; short-time overloads up to 150% of the rated (nominal) load</td>
<td>1.2-1.5</td>
<td>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.</td>
</tr>
<tr>
<td>Same, under conditions of improved reliability</td>
<td>1.5-1.8</td>
<td>Centrifuges and separators. Journal boxes and traction engines of electric locomotives. Mechanisms of crane positioning. Wheels of trucks, tractors, prime movers, locomotives, crane and road-building machines,. Power electric machines. Power-generating plants.</td>
</tr>
<tr>
<td>Loads with considerable jerks and vibrations; short-time overloads up to 200% of the rated (nominal) load</td>
<td>1.8-2.5</td>
<td>Gears. Crushers and pile driver. Crank mechanisms. Ball and impact mills. Frame saws. Rolling mill rollers. High-power ventilators and exhaustors.</td>
</tr>
</tbody>
</table>
In a number of cases, it is not quite easy to perform accurate calculations of loading a bearing. For example, journal boxes of the 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, inertial 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 production.

Based on the analysis of their operation, there has been derived a so-called loading factor, \( k_w \), to be multiplied into the 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 \( k_w = 1 \). 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 \( k_w \).

With the equivalent load (\( P \)) known, the basic rated resource (\( L_{10} \)) selected, the basic dynamic load-carrying capacity (\( C \)) is determined 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 \( L = 0, \) \( P = \infty \), the bearing cannot accommodate load as high as is wished. At a low rotational speed (\( \alpha \leq 20 \text{ min}^{-1} \)), \( P \) values turn out to be overstated. Consequently, for bearings which rotate at low speeds, if at all, -especially when operated under impact loads-the allowable load depends on residual deformation originating 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 \( C_r \) or \( C_a \) 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-carrying capacity are given as calculated on the above bases.

When testing a stationary (non-rotating) bearing for static load-carrying capacity under a load applied in any direction, it is necessary to calculate the 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, \( P_o \), is found from the formula:

\[
P_o = X_o F_r + Y_o F_a \quad \ldots \ldots \quad 2.12
\]

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

\[
P_o = F_a + 2,3F_r \tan \alpha \quad \ldots \ldots \quad 2.13
\]

Where

- \( P_o \) is the equivalent static radial load, H;
- \( P_{oa} \) is the equivalent static axial load, H;
- \( F_r \) is the radial load or the radial component of the load acting upon the bearing, H;
- \( F_a \) is the axial load or the axial component of the load acting upon 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 \((a = 90^\circ)\) are capable to withstand axial loads, only. The equal load for these types of bearings is calculated from the formula \( P_o = F_a \).

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 the present Catalogue.

It is necessary that the 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 \( S_o \left( S_o = \frac{C_o}{P_o} \right) \) 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 conditions, \( S_o = 1-1.5 \) 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 \( s = 1.5-2.5 \).
3. Tolerances

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

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

Table 3.1 Comparison of tolerance classifications of national standards

<table>
<thead>
<tr>
<th>Standard</th>
<th>Tolerance Class</th>
<th>Bearing Types</th>
</tr>
</thead>
<tbody>
<tr>
<td>ISO 492</td>
<td>Normal Class  Class 6  Class 6X</td>
<td>Class 6  Class 5  Class 4  Class 2</td>
</tr>
<tr>
<td>ISO 199</td>
<td>Normal Class  Class 6</td>
<td>Class 5  Class 4</td>
</tr>
<tr>
<td>ISO 578</td>
<td>Class 4</td>
<td>-</td>
</tr>
<tr>
<td>ISO 1224</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>JIS B 1514</td>
<td>class 0  class 6X</td>
<td>Class 6</td>
</tr>
<tr>
<td>DIN 620</td>
<td>P0</td>
<td>P6</td>
</tr>
<tr>
<td>ANSI/AFBMA (Std.201)</td>
<td>ABEC-1</td>
<td>ABEC-3</td>
</tr>
<tr>
<td>ANSI/AFBMA (Std. 19.1)</td>
<td>Class K</td>
<td>Class N</td>
</tr>
<tr>
<td>ANSI / B 3.19 AFBMA Std.19</td>
<td>Class 4</td>
<td>Class 2</td>
</tr>
<tr>
<td>ANSI/AFBMA (Std. 12.1)</td>
<td>-</td>
<td>Class 3P</td>
</tr>
<tr>
<td>ANSI/AFBMA (Std. 12.2)</td>
<td>-</td>
<td>Class 3P</td>
</tr>
</tbody>
</table>