The Design of Rolling Bearing Mountings

Design Examples covering Machines, Vehicles and Equipment

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This publication presents design examples covering various machines, vehicles and equipment having one thing in common: rolling bearings.

For this reason the brief texts concentrate on the rolling bearing aspects of the applications. The operation of the machine allows conclusions to be drawn about the operating conditions which dictate the bearing type and design, the size and arrangement, fits, lubrication and sealing.

Important rolling bearing engineering terms are printed in italics. At the end of this publication they are summarized and explained in a glossary of terms, some supplemented by illustrations.
Example Title

GLOSSARY ...................... 8/8
Additives

Additives are oil-soluble substances added to mineral oils or mineral oil products. By chemical or physical action, they change or improve lubricant properties (oxidation stability, EP properties, foaming, viscosity-temperature behaviour, setting point, flow properties, etc.). Additives are also an important factor in calculating the attainable life (cp. also Factor K).

Adjusted bearing arrangement/Adjustment

An adjusted bearing arrangement consists of two symmetrically arranged angular contact bearings or thrust bearings. During mounting, one bearing ring (for an O arrangement, the inner ring; for an X arrangement, the outer ring) is displaced on its seat until the bearing arrangement has the appropriate axial clearance or the required preload. This means that the adjusted bearing arrangement is particularly suitable for those cases where a close axial guidance is required, for example, for pinion bearing arrangements with spiral toothed bevel gears.

Alignment

Self-aligning bearings are used to compensate for misalignment and tilting.

Angular contact bearings

The term "angular contact bearing" is collectively used for single-row bearings whose contact lines are inclined to the radial plane. So, angular contact bearings are angular contact ball bearings, tapered roller bearings and spherical roller thrust bearings. Axially loaded deep groove ball bearings also act in the same way as angular contact bearings.

Arcanol (FAG rolling bearing greases)

FAG rolling bearing greases Arcanol are field-proven lubricating greases. Their scopes of application were determined by FAG by means of the latest test methods under a large variety of operating conditions and with rolling bearings of all types. The eight Arcanol greases listed in the table on page 179 cover almost all demands on the lubrication of rolling bearings.

Attainable life $L_{na}$, $L_{hna}$

The FAG calculation method for determining the attainable life ($L_{na}$, $L_{hna}$) is based on DIN ISO 281 (cp. Modified life). It takes into account the influences of the operating conditions on the rolling bearing life and indicates the preconditions for reaching endurance strength.

$$L_{na} = a_1 \cdot a_{23} \cdot L \quad [10^6 \text{ revolutions}]$$

and

$$L_{hna} = a_1 \cdot a_{23} \cdot L_h \quad [\text{h}]$$

$a_1$ factor for failure probability (DIN ISO 281);

for a normal (10%) failure probability $a_1 = 1$.

$a_{23}$ factor (life adjustment factor)

$L$ nominal rating life [$10^6 \text{ revolutions}$]

$L_h$ nominal rating life [h]

If the quantities influencing the bearing life (e. g. load, speed, temperature, cleanliness, type and condition of lubricant) are variable, the attainable life ($L_{hna1}$, $L_{hna2}$, ...) under constant conditions has to be determined for every operating time $q \%$. The attainable life is calculated for the total operating time using the formula

$$L_{hna} = \frac{100}{\frac{q_1}{L_{hna1}} + \frac{q_2}{L_{hna2}} + \frac{q_3}{L_{hna3}}}$$
## Glossary

### Arcanol rolling bearing greases · Chemo-physical data · Directions for use

<table>
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<tr>
<th>Arcanol</th>
<th>Thickener</th>
<th>Base oil viscosity at 40°C</th>
<th>Consistency NLGI-Class</th>
<th>Temperature range</th>
<th>Colour</th>
<th>Main characteristics</th>
<th>Typical applications</th>
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</thead>
<tbody>
<tr>
<td>L12V Polyurea</td>
<td>Mineral oil</td>
<td>ISO VG 100</td>
<td>2</td>
<td>–30...+160</td>
<td>2002 vermillion</td>
<td>Special grease for high temperatures</td>
<td>Couplings, electric machines (motors, generators)</td>
</tr>
<tr>
<td>L71V Lithium soap</td>
<td>Mineral oil</td>
<td>ISO VG 100</td>
<td>3</td>
<td>–30...+140</td>
<td>4008 signal violet</td>
<td>Standard grease for bearings with O.D.s &gt; 62 mm</td>
<td>Large electric motors, wheel bearings for motor vehicles, ventilators</td>
</tr>
<tr>
<td>L74V Special soap</td>
<td>Synthetic oil</td>
<td>ISO VG 22</td>
<td>2</td>
<td>–40...+120</td>
<td>6018 yellow-green</td>
<td>Special grease for high speeds and low temperatures</td>
<td>Machine tools, spindle bearings, instruments</td>
</tr>
<tr>
<td>L78V Lithium soap</td>
<td>Mineral oil</td>
<td>ISO VG 100</td>
<td>2</td>
<td>–30...+130</td>
<td>1018 zinc yellow</td>
<td>Standard grease for bearings with O.D.s ≤ 62 mm</td>
<td>Small electric motors, agricultural and construction machinery, household appliances</td>
</tr>
<tr>
<td>L79V Synthetic</td>
<td>Synthetic oil</td>
<td>390</td>
<td>2</td>
<td>–30...+270</td>
<td>1024 yellow ochre</td>
<td>Special grease for extremely high temperatures and chemically aggressive environments</td>
<td>Track rollers in bakery machines, piston pins in compressors, kiln trucks, chemical plants (please observe safety data sheet)</td>
</tr>
<tr>
<td>L135V Lithium soap with EP additives</td>
<td>Mineral oil</td>
<td>ISO VG 85</td>
<td>2</td>
<td>–40...+150</td>
<td>2000 yellow orange</td>
<td>Special grease for high loads, high speeds, high temperatures</td>
<td>Rolling mills, construction machinery, motor vehicles, rail vehicles, spinning and grinding spindles</td>
</tr>
<tr>
<td>L186V Lithium soap with EP additives</td>
<td>Mineral oil</td>
<td>ISO VG 460</td>
<td>2</td>
<td>–20...+140</td>
<td>7005 mouse-grey</td>
<td>Special grease for extremely high loads, medium speeds, medium temperatures</td>
<td>Heavily stressed mining machinery, construction machinery, machines with oscillating movements</td>
</tr>
<tr>
<td>L223V Lithium soap with EP additives</td>
<td>Mineral oil</td>
<td>ISO VG 1000</td>
<td>2</td>
<td>–10...+140</td>
<td>5005 signal blue</td>
<td>Special grease for extremely high loads, low speeds</td>
<td>Heavily stressed mining machinery, construction machinery, particularly for impact loads and large bearings</td>
</tr>
</tbody>
</table>
Glossary

Axial clearance

The axial clearance of a bearing is the total possible axial displacement of one bearing ring measured without load. There is a difference between the axial clearance of the unmounted bearing and the axial operating clearance existing when the bearing is mounted and running at operating temperature.

Base oil

is the oil contained in a lubricating grease. The amount of oil varies with the type of thickener and the grease application. The penetration number and the frictional behaviour of the grease vary with the amount of base oil and its viscosity.

Basic a_{23II} value

The basic a_{23II} value is the basis for determining factor a_{23}, used in attainable life calculation.

Bearing life

The life of dynamically stressed rolling bearings, as defined by DIN ISO 281, is the operating time until failure due to material fatigue (fatigue life).

By means of the classical calculation method, a comparison calculation, the nominal rating life L or L_{h}, is determined; by means of the refined FAG calculation process, the attainable life L_{na} or L_{hna} is determined (see also factor a_{23}).

Cage

The cage of a rolling bearing prevents the rolling elements from rubbing against each other. It keeps them evenly spaced and guides them through unloaded sections of the bearing circumference.

The cage of a needle roller bearing also has to guide the needle rollers parallel to the axis. In the case of separable bearings the cage retains the rolling element set, thus facilitating bearing mounting. Rolling bearing cages are classified into the categories pressed cages and machined/moulded cages.

Circumferential load

If the ring under consideration rotates in relation to the radial load, the entire circumference of the ring is, during each revolution, subjected to the maximum load. This ring is circumferentially loaded. Bearings with circumferential load must be mounted with a tight fit to avoid sliding (cp. Point load, Oscillating load).

Cleanliness factor s

The cleanliness factor s quantifies the effect of contamination on the attainable life. The product of s and the basic a_{23II} factor is the factor a_{23}.

Contamination factor V is required to determine s. s = 1 always applies to normal cleanliness (V = 1). With improved cleanliness (V = 0.5) and utmost cleanliness (V = 0.3) a cleanliness factor s > 1 is obtained from the right diagram (a) on page 181, based on the stress index f_{s} and depending on the viscosity ratio K.

s = 1 applies to K < 0.4.

With V = 2 (moderately contaminated lubricant) to V = 3 (heavily contaminated lubricant), s < 1 is obtained from diagram (b).

Combined load

This applies when a bearing is loaded both radially and axially, and the resulting load acts, therefore, at the load angle \( \beta \).

Depending on the type of load, the equivalent dynamic load \( P \) or the equivalent static load \( P_0 \) is determined with the radial component \( F_r \) and the thrust component \( F_t \) of the combined load.
Diagram for determining the cleanliness factor $s$

a) Diagram for improved to utmost cleanliness
b) Diagram for moderately contaminated lubricant and heavily contaminated lubricant

Consistency

Measure of the resistance of a lubricating grease to being deformed.
Consistency classification to NLGI, cp. Penetration.

Contact angle $\alpha$

The contact angle $\alpha$ is the angle formed by the contact lines of the rolling elements and the radial plane of the bearing. $\alpha_0$ refers to the nominal contact angle, i.e. the contact angle of the load-free bearing.
Under axial loads the contact angle of deep groove ball bearings, angular contact ball bearings etc. increases.
Under a combined load it changes from one rolling element to the next. These changing contact angles are taken into account when calculating the pressure distribution within the bearing.
Ball bearings and roller bearings with symmetrical rolling elements have identical contact angles at their inner rings and outer rings. In roller bearings with asymmetrical rollers the contact angles at inner ring and outer ring are not identical. The equilibrium of forces in these bearings is maintained by a force component which is directed towards the lip.

Contact line

The rolling elements transmit loads from one bearing ring to the other in the direction of the contact lines.
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Contamination factor V

The contamination factor V indicates the degree of cleanliness in the lubricating gap of rolling bearings based on the oil cleanliness classes defined in ISO 4406.

When determining the factor $a_{23}$ and the attainable life, V is used, together with the stress index $f_s*$ and the viscosity ratio $k$, to determine the cleanliness factor $s$.

V depends on the bearing cross section $(D – d)/2$, the type of contact between the mating surfaces and especially the cleanliness level of the oil.

If hard particles from a defined size on are cycled in the most heavily stressed contact area of a rolling bearing, the resulting indentations in the contact surfaces lead to premature material fatigue. The smaller the contact area, the more damaging the effect of a particle above a certain size when being cycled. Small bearings with point contact are especially vulnerable.

According to today’s knowledge the following cleanliness scale is useful (the most important values are in boldface):

- $V = 0.3$ utmost cleanliness
- $V = 0.5$ improved cleanliness
- $V = 1$ normal cleanliness
- $V = 2$ moderately contaminated lubricant
- $V = 3$ heavily contaminated lubricant

Preconditions for utmost cleanliness ($V = 0.3$):
- bearings are greased and protected by seals or shields against dust by the manufacturer
- grease lubrication by the user who fits the bearings into clean housings under top cleanliness conditions, lubricates them with clean grease and takes care that dirt cannot enter the bearing during operation
- flushing the oil circulation system prior to the first operation of the cleanly fitted bearings and taking care that the oil cleanliness class is ensured during the entire operating time

Guide values for V

<table>
<thead>
<tr>
<th>(D-d)/2 V</th>
<th>Point contact required oil cleanliness class according to ISO 4406</th>
<th>guide values for filtration ratio according to ISO 4572</th>
<th>Line contact required oil cleanliness class according to ISO 4406</th>
<th>guide values for filtration ratio according to ISO 4572</th>
</tr>
</thead>
<tbody>
<tr>
<td>mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>≤ 12.5</td>
<td>0.3 11/8 $\beta_3 \geq 200$</td>
<td>12/9 $\beta_3 \geq 200$</td>
<td>13/10 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.5 12/9 $\beta_3 \geq 200$</td>
<td>14/11 $\beta_3 \geq 75$</td>
<td>15/12 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>1 14/11 $\beta_3 \geq 75$</td>
<td>16/13 $\beta_3 \geq 75$</td>
<td>17/14 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2 15/12 $\beta_3 \geq 75$</td>
<td>16/13 $\beta_3 \geq 75$</td>
<td>17/14 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>3 16/13 $\beta_3 \geq 75$</td>
<td>17/14 $\beta_3 \geq 75$</td>
<td>19/15 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td>&gt; 12.5...20</td>
<td>0.3 12/9 $\beta_3 \geq 200$</td>
<td>13/10 $\beta_3 \geq 75$</td>
<td>14/11 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.5 13/10 $\beta_3 \geq 75$</td>
<td>15/12 $\beta_3 \geq 75$</td>
<td>16/13 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>1 15/12 $\beta_3 \geq 75$</td>
<td>17/14 $\beta_3 \geq 75$</td>
<td>19/15 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2 16/13 $\beta_3 \geq 75$</td>
<td>17/14 $\beta_3 \geq 75$</td>
<td>20/16 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>3 18/14 $\beta_3 \geq 75$</td>
<td>19/15 $\beta_3 \geq 75$</td>
<td>21/17 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td>&gt; 20...35</td>
<td>0.3 13/10 $\beta_3 \geq 75$</td>
<td>14/11 $\beta_3 \geq 75$</td>
<td>15/12 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.5 14/11 $\beta_3 \geq 75$</td>
<td>16/13 $\beta_3 \geq 75$</td>
<td>17/14 $\beta_3 \geq 75$</td>
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</tr>
<tr>
<td></td>
<td>1 16/13 $\beta_3 \geq 75$</td>
<td>18/15 $\beta_3 \geq 75$</td>
<td>20/16 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2 17/14 $\beta_3 \geq 75$</td>
<td>19/16 $\beta_3 \geq 75$</td>
<td>21/17 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>3 19/15 $\beta_3 \geq 75$</td>
<td>20/16 $\beta_3 \geq 75$</td>
<td>22/18 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td>&gt; 35</td>
<td>0.3 14/11 $\beta_3 \geq 75$</td>
<td>15/12 $\beta_3 \geq 75$</td>
<td>16/13 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.5 15/12 $\beta_3 \geq 75$</td>
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<td>18/15 $\beta_3 \geq 75$</td>
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<td></td>
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<td></td>
<td>2 18/15 $\beta_3 \geq 75$</td>
<td>19/16 $\beta_3 \geq 75$</td>
<td>22/18 $\beta_3 \geq 75$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>3 20/16 $\beta_3 \geq 75$</td>
<td>21/17 $\beta_3 \geq 75$</td>
<td>23/20 $\beta_3 \geq 75$</td>
<td></td>
</tr>
</tbody>
</table>

The oil cleanliness class can be determined by means of oil samples by filter manufacturers and institutes. It is a measure of the probability of life-reducing particles being cycled in a bearing. Suitable sampling should be observed (see e.g. DIN 51570). Today, online measuring instruments are available. The cleanliness classes are reached if the entire oil volume flows through the filter within a few minutes.

To ensure a high degree of cleanliness flushing is required prior to bearing operation.

For example, a filtration ratio $\beta_3 \geq 200$ (ISO 4572) means that in the so-called multi-pass test only one of 200 particles ≥ 3 µm passes the filter.

Filters with coarser filtration ratios than $\beta_25 \geq 75$ should not be used due to the ill effect on the other components within the circulation system.
Glossary

Preconditions for normal cleanliness (V = 1):
– good sealing adapted to the environment
– cleanliness during mounting
– oil cleanliness according to V = 1
– observing the recommended oil change intervals

Possible causes of heavy lubricant contamination (V = 3):
– the cast housing was inadequately cleaned
– abraded particles from components which are subject to wear enter the circulating oil system of the machine
– foreign matter penetrates into the bearing due to unsatisfactory sealing
– water which entered the bearing, also condensation water, caused standstill corrosion or deterioration of the lubricant properties

The necessary oil cleanliness class according to ISO 4406 is an objectively measurable level of the contamination of a lubricant.

In accordance with the particle-counting method, the number of all particles > 5 µm and all particles > 15 µm are allocated to a certain ISO oil cleanliness class. For example, an oil cleanliness class 15/12 according to ISO 4406 means that between 16,000 and 32,000 particles > 5 µm and between 2,000 and 4,000 particles > 15 µm are present per 100 ml of a fluid.

A defined filtration ratio $\beta_x$ should exist in order to reach the oil cleanliness required.

The filtration ratio is the ratio of all particles $> x$ µm before passing the filter to the particles $> x$ µm which have passed the filter. For example, a filtration ratio $\beta_3 \geq 200$ means that in the so-called multi-pass test (ISO 4572) only one of 200 particles $\geq 3$ µm passes the filter.

Counter guidance

Angular contact bearings and single-direction thrust bearings accommodate axial forces only in one direction. A second, symmetrically arranged bearing must be used for "counter guidance," i.e. to accommodate the axial forces in the other direction.

Curvature ratio

In all bearing types with a curved raceway profile the radius of the raceway is slightly larger than that of the rolling elements. This curvature difference in the axial plane is defined by the curvature ratio $\kappa$. The curvature ratio is the curvature difference between the rolling element radius and the slightly larger groove radius.

$$\text{curvature ratio } \kappa = \frac{\text{groove radius} - \text{rolling element radius}}{\text{rolling element radius}}$$

Dynamic load rating C

The dynamic load rating C (see FAG catalogues) is a factor for the load carrying capacity of a rolling bearing under dynamic load. It is defined, in accordance with DIN ISO 281, as the load a rolling bearing can theoretically accommodate for a nominal life $L$ of $10^6$ revolutions (fatigue life).

Dynamic stressing/dynamic load

Rolling bearings are dynamically stressed when one ring rotates relative to the other under load. The term "dynamic" does not refer, therefore, to the effect of the load but rather to the operating condition of the bearing. The magnitude and direction of the load can remain constant.

When calculating the bearings, a dynamic stress is assumed when the speed $n$ amounts to at least 10 min$^{-1}$ (see Static stressing).

Endurance strength

Tests by FAG and field experience have proved that, under the following conditions, rolling bearings can be fail-safe:

– utmost cleanliness in the lubricating gap (contamination factor $V = 0.3$)
– complete separation of the components in rolling contact by the lubricating film (viscosity ratio $\kappa \geq 4$)
– load according to stress index $f_r \geq 8$
Glossary

**EP additives**

Wear-reducing additives in lubricating greases and lubricating oils, also referred to as extreme pressure lubricants.

**Equivalent dynamic load P**

For dynamically loaded rolling bearings operating under a combined load, the calculation is based on the equivalent dynamic load. This is a radial load for radial bearings and an axial and centric load for axial bearings, having the same effect on fatigue as the combined load. The equivalent dynamic load \( P \) is calculated by means of the following equation:

\[
P = X \cdot F_r + Y \cdot F_a \quad [\text{kN}]
\]

- \( F_r \) radial load \([\text{kN}]\)
- \( F_a \) axial load \([\text{kN}]\)
- \( X \) radial factor (see FAG catalogues)
- \( Y \) thrust factor (see FAG catalogues)

**Equivalent static load \( P_0 \)**

Statically stressed rolling bearings which operate under a combined load are calculated with the equivalent static load. It is a radial load for radial bearings and an axial and centric load for thrust bearings, having the same effect with regard to permanent deformation as the combined load.

The equivalent static load \( P_0 \) is calculated with the formula:

\[
P = X_0 \cdot F_r + Y_0 \cdot F_a \quad [\text{kN}]
\]

- \( F_r \) radial load \([\text{kN}]\)
- \( F_a \) axial load \([\text{kN}]\)
- \( X_0 \) radial factor (see FAG catalogues)
- \( Y_0 \) thrust factor (see FAG catalogues)

**Factor \( a_{23} \) (life adjustment factor)**

The \( a_{23} \) factor is used to calculate the attainable life. FAG use \( a_{23} \) instead of the mutually dependent adjustment factors for material \( (a_2) \) and operating conditions \( (a_3) \) indicated in DIN ISO 281.

\[
a_{23} = a_2 \cdot a_3
\]

The \( a_{23} \) factor takes into account effects of:

- amount of load (stress index \( f_s \)),
- lubricating film thickness (viscosity ratio \( \kappa \)),
- lubricant additives (value \( K \)),
- contaminants in the lubricating gap (cleanliness factor \( s \)),
- bearing type (value \( K \)).

The diagram on page 185 is the basis for the determination of the \( a_{23} \) factor using the basic \( a_{23II} \) value. The \( a_{23} \) factor is obtained from the equation \( a_{23II} \cdot s \) (\( s \) being the cleanliness factor).

The viscosity ratio \( \kappa = v / v_1 \) and the value \( K \) are required for locating the basic value. The most important zone (II) in the diagram applies to normal cleanliness \((s = 1)\).

The viscosity ratio \( \kappa \) is a measure of the lubricating film development in the bearing.

\[ v \quad \text{operating viscosity of the lubricant, depending on the nominal viscosity (at 40 °C) and the operating temperature \( t \) (fig. 1). In the case of lubricating greases, \( v \) is the operating viscosity of the base oil.} \]

\[ v_1 \quad \text{rated viscosity, depending on mean bearing diameter \( d_m \) and operating speed \( n \) (fig. 2).} \]

The diagram (fig. 3) for determining the basic \( a_{23II} \) factor is subdivided into zones I, II and III.

Most applications in rolling bearing engineering are covered by zone II. It applies to normal cleanliness \((\text{contamination factor } V = 1)\). In zone II, \( a_{23} \) can be determined as a function of \( \kappa \) by means of value \( K \).

With \( K = 0 \) to 6, \( a_{23II} \) is found on one of the curves in zone II of the diagram.

With \( K > 6 \), \( a_{23} \) must be expected to be in zone III. In such a case conditions should be improved so that zone II can be reached.
Fatigue life

The fatigue life of a rolling bearing is the operating time from the beginning of its service until failure due to material fatigue. The fatigue life is the upper limit of service life.

The classical calculation method, a comparison calculation, is used to determine the nominal life \( L \) or \( \bar{L}_4 \); by means of the refined FAG calculation process the attainable life \( L_{na} \) or \( L_{hna} \) is determined (see also \( a_{23} \) factor).

Fits

The tolerances for the bore and for the outside diameter of rolling bearings are standardized in DIN 620 (cp. Tolerance class). The seating characteristics required for reliable bearing operation, which are dependent on the operating conditions of the application, are obtained by the correct selection of shaft and housing machining tolerances.

For this reason, the seating characteristics of the rings are indicated by the shaft and housing tolerance symbols.

Three factors should be borne in mind in the selection of fits:
Glossary

1. Safe retention and uniform support of the bearing rings
2. Simplicity of mounting and dismounting
3. Axial freedom of the floating bearing

The simplest and safest means of ring retention in the circumferential direction is achieved by a tight fit. A tight fit will support the rings evenly, a factor which is indispensable for the full utilization of the load-carrying capacity. Bearing rings accommodating a circumferential load or an oscillating load are always fitted tightly. Bearing rings accommodating a point load may be fitted loosely. The higher the load the tighter should be the interference fit provided, particularly for shock loading. The temperature gradient between bearing ring and mating component should also be taken into account. Bearing type and size also play a role in the selection of the correct fit.

Floating bearing

In a locating/floating bearing arrangement the floating bearing compensates for axial thermal expansion. Cylindrical roller bearings of NU and N designs, as well as needle roller bearings, are ideal floating bearings. Differences in length are compensated for in the floating bearing itself. The bearing rings can be given tight fits. Non-separable bearings, such as deep groove ball bearings and spherical roller bearings, can also be used as floating bearings. In such a case one of the two bearing rings is given a loose fit, with no axial mating surface so that it can shift freely on its seat.

Floating bearing arrangement

A floating bearing arrangement is an economical solution where no close axial shaft guidance is required. The design is similar to that of an adjusted bearing arrangement. In a floating bearing arrangement, however, the shaft can shift relative to the housing by the axial clearance s. The value s is determined depending on the required guiding accuracy in such a way that detrimental axial preloading of the bearings is prevented even under unfavourable thermal conditions.

In floating bearing arrangements with NJ cylindrical roller bearings, length variations are compensated for in the bearings. Inner and outer rings can be fitted tightly. Non-separable radial bearings such as deep groove ball bearings, self-aligning ball bearings and spherical roller bearings can also be used. One ring of each bearing — generally the outer ring — is given a loose fit.

Grease, grease lubrication

cp. Lubricating grease

Grease service life

The grease service life is the period from start-up until the failure of a bearing as a result of lubrication breakdown. The grease service life is determined by the
- amount of grease
- grease type (thickener, base oil, additives)
- bearing type and size
- type and amount of loading
- speed index
- bearing temperature

Index of dynamic stressing $f_L$

The value recommended for dimensioning can be expressed, instead of in hours, as the index of dynamic stressing $f_L$. It is calculated from the dynamic load rating $C$, the equivalent dynamic load $P$ and the speed factor $f_n$.

$$f_L = \frac{C}{P} \cdot f_n$$

The $f_L$ value to be obtained for a correctly dimensioned bearing arrangement is an empirical value obtained from field-proven identical or similar bearing mountings.

The values indicated in various FAG publications take into account not only an adequate fatigue life but also other requirements such as low weight for light-weight constructions, adaptation to given mating parts, higher-than-usual peak loads, etc. The $f_L$ values conform with the latest standards resulting from technical progress. For comparison with a field-proven bearing mounting the calculation of stressing must, of course, be based on the same former method.

Based on the calculated $f_L$ value, the nominal rating life $L_h$ in hours can be determined.
**Glossary**

\[ L_h = 500 \cdot f_L^p \quad [h] \]

\[ p = 3 \quad \text{for ball bearings} \]

\[ p = \frac{10}{3} \quad \text{for roller bearings and needle roller bearings} \]

**Index of static stressing \( f_s \)**

The index of static stressing \( f_s \) for *statically loaded bearings* is calculated to ensure that a bearing with an adequate load carrying capacity has been selected. It is calculated from the *static load rating* \( C_0 \) and the *equivalent static load* \( P_0 \).

\[ f_s = \frac{C_0}{P_0} \]

The index \( f_s \) is a safety factor against permanent deformations of the contact areas between raceway and the most heavily loaded rolling element. A high \( f_s \) value is required for bearings which must run smoothly and particularly quietly. Smaller values suffice where a moderate degree of running quietness is required. The following values are generally recommended:

\[ f_s = 1.5 \ldots 2.5 \quad \text{for a high degree} \]

\[ f_s = 1 \ldots 1.5 \quad \text{for a normal degree} \]

\[ f_s = 0.7 \ldots 1 \quad \text{for a moderate degree} \]

**K value**

The K value is an auxiliary quantity needed to determine the *basic \( a_{23II} \) factor* when calculating the *attainable life* of a bearing.

\[ K = K_1 + K_2 \]

\( K_1 \) depends on the bearing type and the *stress index* \( f_s \), see diagram.

\( K_2 \) depends on the *stress index* \( f_s \) and the *viscosity ratio* \( \kappa \). The values in the diagram (below) apply to lubricants without *additives* and lubricants with *additives* whose effects in rolling bearings was not tested.

**Value \( K_1 \)**

\[ a \) ball bearings
(b) tapered roller bearings, cylindrical roller bearings
(c) spherical roller bearings, spherical roller thrust bearings, cylindrical roller thrust bearings \(^{1, 3} \)
(d) full complement cylindrical roller bearings \(^{1, 2} \)

\(^{1} \) Attainable only with lubricant filtering corresponding to \( V < 1 \), otherwise \( K_1 > 6 \) must be assumed.

\(^{2} \) To be observed for the determination of \( \nu \): the friction is at least twice the value in caged bearings. This results in higher bearing temperature.

\(^{3} \) Minimum load must be observed.

**Value \( K_2 \)**

\[ \kappa = 0.2** \]

\[ \kappa = 0.25** \]

\[ \kappa = 0.3** \]

\[ \kappa = 0.4** \]

\[ \kappa = 0.7 \]

\[ \kappa = 1 \]

\[ \kappa = 2 \]

\[ \kappa = 4 \]

**Kinematically permissible speed**

The kinematically permissible speed is indicated in the FAG catalogues also for bearings for which – according to DIN 732 – no *thermal reference speed* is defined. Decisive criteria for the kinematically permissible speed are e.g. the strength limit of the bearing components or the permissible sliding velocity of rubbing seals. The kinematically permissible speed can be reached, for example, with

- specially designed lubrication
- bearing clearance adapted to the operating conditions
- accurate machining of the bearing seats
- special regard to heat dissipation

**Life**

Cp. also *Bearing life*. 
**Load angle**

The load angle $\beta$ is the angle between the resultant applied load $F$ and the radial plane of the bearing. It is the resultant of the radial component $F_r$ and the axial component $F_a$:

$$\tan \beta = \frac{F_a}{F_r}$$

**Lubricating grease**

Lubricating greases are consistent mixtures of thickeners and base oils. The following grease types are distinguished:
- metal soap base greases consisting of metal soaps as thickeners and lubricating oils,
- non-soap greases comprising inorganic gelling agents or organic thickeners and lubricating oils
- synthetic greases consisting of organic or inorganic thickeners and synthetic oils.

**Lubricating oil**

Rolling bearings can be lubricated either with mineral oils or synthetic oils. Today, mineral oils are most frequently used.

**Lubrication interval**

The lubrication interval corresponds to the minimum grease service life of standard greases (see FAG publication WL 81 115). This value is assumed if the grease service life for the grease used is not known.

**Machined/moulded cages**

Machined cages of metal and textile laminated phenolic resin are produced in a cutting process. They are made from tubes of steel, light metal or textile laminated phenolic resin, or cast brass rings. Cages of polyamide 66 (polyamide cages) are manufactured by injection moulding. Like pressed cages, they are suitable for large-series bearings.

Machined cages of metal and textile laminated phenolic resin are mainly eligible for bearings of which only small series are produced. Large, heavily loaded bearings feature machined cages for strength reasons. Machined cages are also used where lip guidance of the cage is required. Lip-guided cages for high-speed bearings are often made of light materials, such as light metal or textile laminated phenolic resin to minimize the inertia forces.

**Mineral oils**

Crude oils and/or their liquid derivates.
Cp. also Synthetic lubricants.
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Modified life

The standard Norm DIN ISO 281 introduced, in addition to the nominal rating life $L_{10}$, the modified life $L_{na}$ to take into account, apart from the load, the influence of the failure probability ($factor a_1$), of the material ($factor a_2$) and of the operating conditions ($factor a_3$).

DIN ISO 281 indicates no figures for the factor $a_{23}$ ($a_{23} = a_2 \cdot a_3$). With the FAG calculation process for the attainable life ($L_{na}$, $L_{hna}$), however, operating conditions can be expressed in terms of figures by the $factor a_{23}$.

NLGI class

Cp. Penetration.

Nominal rating life

The standardized calculation method for dynamically stressed rolling bearings is based on material fatigue (formation of pitting) as the cause of failure. The life formula is:

$$L_{10} = L = \left(\frac{C}{P}\right)^p \text{ [10}^6\text{ revolutions]}$$

$L_{10}$ is the nominal rating life in millions of revolutions which is reached or exceeded by at least 90 % of a large group of identical bearings.

In the formula,

- $C$ dynamic load rating [kN]
- $P$ equivalent dynamic load [kN]
- $p$ life exponent
  - $p = 3$ for ball bearings
  - $p = 10/3$ for roller bearings and needle roller bearings.

Where the bearing speed is constant, the life can be expressed in hours.

$$L_{h10} = L_h = \frac{L \cdot 10^6}{n \cdot 60} \text{ [h]}$$

$n$ speed [min$^{-1}$]

$L_h$ can also be determined by means of the index of dynamic stressing $f_1$.

The nominal rating life $L$ or $L_h$ applies to bearings made of conventional rolling bearing steel and the usual operating conditions (good lubrication, no extreme temperatures, normal cleanliness).

The nominal rating life deviates more or less from the really attainable life of rolling bearings. Influences such as lubricating film thickness, cleanliness in the lubricating gap, lubricant additives and bearing type are taken into account in the adjusted rating life calculation by the $factor a_{23}$.

O arrangement

In an O arrangement (adjusted bearing mounting) two angular contact bearings are mounted symmetrically in such a way that the pressure cone apex of the left-hand bearing points to the left and the pressure cone apex of the right-hand bearing points to the right.

With the O arrangement one of the bearing inner rings is adjusted. A bearing arrangement with a large spread is obtained which can accommodate a considerable tilting moment even with a short bearing distance. A suitable fit must be selected to ensure displaceability of the inner ring.

Oil/oil lubrication

see Lubricating oil.

Operating clearance

There is a distinction made between the radial or axial clearance of the bearing prior to mounting and the radial or axial clearance of the mounted bearing at operating temperature (operating clearance). Due to tight fits and temperature differences between inner and outer ring the operating clearance is usually smaller than the clearance of the unmounted bearing.

Operating viscosity $\nu$

Kinematic viscosity of an oil at operating temperature. The operating viscosity $\nu$ can be determined by means of a viscosity-temperature diagram if the viscosities at two temperatures are known. The operating viscosity of mineral oils with average viscosity-temperature behaviour can be determined by means of diagram 1 (page 185).

For evaluating the lubricating condition the viscosity ratio $\kappa$ (operating viscosity $\nu$ / rated viscosity $\nu_r$) is formed when calculating the attainable life.

Oscillating load

In selecting the fits for radial bearings and angular contact bearings the load conditions have to be considered. With relative oscillatory motion between the radial
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load and the ring to be fitted, conditions of "oscillating load" occur. Both bearing rings must be given a tight fit to avoid sliding (cp. *circumferential load*).

**Penetration**

Penetration is a measure of the **consistency** of a *lubricating grease*. Worked penetration is the penetration of a grease sample that has been worked, under exactly defined conditions, at 25 °C. Then the depth of penetration – in tenths of a millimetre – of a standard cone into a grease-filled vessel is measured.

Penetration of common rolling bearing greases

<table>
<thead>
<tr>
<th>NLGI class</th>
<th>Worked penetration (Penetration classes)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.1 mm</td>
</tr>
<tr>
<td>1</td>
<td>310...340</td>
</tr>
<tr>
<td>2</td>
<td>265...295</td>
</tr>
<tr>
<td>3</td>
<td>220...250</td>
</tr>
<tr>
<td>4</td>
<td>175...205</td>
</tr>
</tbody>
</table>

**Point load**

In selecting the **fits** for the bearing rings of *radial bearings* and *angular contact bearings* the load conditions have to be considered. If the ring to be fitted and the radial load are stationary relative to each other, one point on the circumference of the ring is always subjected to the maximum load. This ring is point-loaded. Since, with point load, the risk of the ring sliding on its seat is minor, a tight fit is not absolutely necessary. With *circumferential load* or *oscillating load*, a tight fit is imperative.

**Polyamide cage**

Moulded cages of glass fibre reinforced polyamide PA66-GF25 are made by injection moulding and are used in numerous large-series bearings. Injection moulding has made it possible to realize *cage* designs with an especially high load carrying capacity. The elasticity and low weight of the cages are of advantage where shock-type bearing loads, great accelerations and decelerations as well as tilting of the bearing rings relative to each other have to be accommodated. Polyamide cages feature very good sliding and dry running properties. *Cages* of glass fibre reinforced polyamide 66 can be used at operating temperatures of up to 120 °C for extended periods of time. In *oil*-lubricated bearings, *additives* contained in the *oil* may reduce the *cage* life. At increased temperatures, aged oil may also have an impact on the cage life so that it is important to observe the oil change intervals.

**Precision bearings/precision design**

In addition to bearings of normal precision (*tolerance class PN*), bearings of precision design (*precision bearings*) are produced for increased demands on working precision, speeds or quietness of running. For these applications the tolerance classes P6, P6X, P5, P4 and P2 were standardized. In addition, some bearing types are also produced in the *tolerance classes* P4S, SP and UP in accordance with an FAG company standard.

**Pressed cage**

Pressed cages are usually made of steel, but sometimes of brass, too. They are lighter than *machined metal cages*. Since a pressed cage barely closes the gap between inner ring and outer ring, *lubricating grease* can easily penetrate into the bearing. It is stored at the *cage*.

**Pressure cone apex**

The pressure cone apex is that point on the bearing axis where the contact lines of an *angular contact bearing* intersect. The contact lines are the generatrices of the pressure cone. In *angular contact bearings* the external forces act, not at the bearing centre, but at the pressure cone apex. This fact has to be taken into account when calculating the *equivalent dynamic load* $P$ and the *equivalent static load* $P_0$. 

---

![Diagram of point load on inner ring](image1)

**Point load on inner ring**

- Weight
- Stationary inner ring
- Constant load direction

![Diagram of point load on outer ring](image2)

**Point load on outer ring**

- Weight
- Stationary outer ring
- Constant load direction

![Diagram of imbalance](image3)

**Imbalance**

- Rotating inner ring
- Direction of load rotating with inner ring

![Diagram of imbalance](image4)

**Imbalance**

- Rotating outer ring
- Direction of load rotating with outer ring
Radial bearings

Radial bearings are those primarily designed to accommodate radial loads; they have a nominal contact angle $\alpha_0 \leq 45^\circ$. The dynamic load rating and the static load rating of radial bearings refer to pure radial loads (see Thrust bearings).

Radial clearance

The radial clearance of a bearing is the total distance by which one bearing ring can be displaced in the radial plane, under zero measuring load. There is a difference between the radial clearance of the unmounted bearing and the radial operating clearance of the mounted bearing running at operating temperature.

Radial clearance group

The radial clearance of a rolling bearing must be adapted to the conditions at the bearing location (fits, temperature gradient, speed). Therefore, rolling bearings are assembled into several radial clearance groups, each covering a certain range of radial clearance.

The radial clearance group CN (normal) is such that the bearing, under normal fitting and operating conditions, maintains an adequate operating clearance. The other clearance groups are:

- C2 radial clearance less than normal
- C3 radial clearance larger than normal
- C4 radial clearance larger than C3.

Rated viscosity $\nu_1$

The rated viscosity is the kinematic viscosity attributed to a defined lubricating condition. It depends on the speed and can be determined with diagram 2 (page 185) by means of the mean bearing diameter and the bearing speed. The viscosity ratio $\kappa$ (operating viscosity $\nu$/rated viscosity $\nu_1$) allows the lubricating condition to be assessed (see also factor $\alpha_{23}$).

Relubrication interval

Period after which the bearings are relubricated. The relubrication interval should be shorter than the lubrication interval.

Rolling elements

This term is used collectively for balls, cylindrical rollers, barrel rollers, tapered rollers or needle rollers in rolling contact with the raceways.

Seals/Sealing

On the one hand the sealing should prevent the lubricant (usually lubricating grease or lubricating oil) from escaping from the bearing and, on the other hand, prevent contaminants from entering into the bearing. It has a considerable influence on the service life of a bearing arrangement (cp. Wear, Contamination factor $V$).

A distinction is made between non-rubbing seals (e.g. gap-type seals, labyrinth seals, shields) and rubbing seals (e.g. radial shaft seals, V-rings, felt rings, sealing washers).

Self-aligning bearings

Self-aligning bearings are all bearing types capable of self-alignment during operation to compensate for misalignment as well as shaft and housing deflection. These bearings have a spherical outer ring raceway. They are self-aligning ball bearings, barrel roller bearings, spherical roller bearings and spherical roller thrust bearings.

Thrust ball bearings with seating rings and S-type bearings are not self-aligning bearings because they can compensate for misalignment and deflections only during mounting and not in operation.

Separable bearings

These are rolling bearings whose rings can be mounted separately. This is of advantage where both bearing rings require a tight fit.

Separable bearings include four-point bearings, cylindrical roller bearings, tapered roller bearings, thrust ball bearings, cylindrical roller thrust bearings and spherical roller thrust bearings.

Non-separable bearings include deep groove ball bearings, single-row angular contact ball bearings, self-
aligning ball bearings, barrel roller bearings and spherical roller bearings.

Service life

This is the life during which the bearing operates reliably. The fatigue life of a bearing is the upper limit of its service life. Often this limit is not reached due to wear or lubrication breakdown (cp. Grease service life).

Speed factor \( f_n \)

The auxiliary quantity \( f_n \) is used, instead of the speed \( n \text{ [min}^{-1}\text{]} \), to determine the index of dynamic stressing, \( f_L \).

\[
f_n = \sqrt{\frac{33.1/3}{n}}
\]

\( p = 3 \) for ball bearings

\( p = 10 \) for roller bearings and needle roller bearings

Speed index \( n \cdot d_m \)

The product from the operating speed \( n \text{ [min}^{-1}\text{]} \) and the mean bearing diameter \( d_m \text{ [mm]} \) is mainly used for selecting suitable lubricants and lubricating methods.

\[
d_m = \frac{D + d}{2}
\]

\( D \) bearing outside diameter [mm]

\( d \) bearing bore [mm]

Speed suitability

Generally, the maximum attainable speed of rolling bearings is dictated by the permissible operating temperatures. This limiting criterion takes into account the thermal reference speed. It is determined on the basis of exactly defined, uniform criteria (reference conditions) in accordance with DIN 732, part 1 (draft). In catalogue WL 41 520 "FAG Rolling Bearings” a reference is made to a method based on DIN 732, part 2, for determining the thermally permissible operating speed on the basis of the thermal reference speed for cases where the operating conditions (load, oil viscosity or permissible temperature) deviate from the reference conditions.

The kinematically permissible speed is indicated also for bearings for which – according to DIN 732 – no thermal reference speed is defined, e. g. for bearings with rubbing seals.

Spread

Generally, the spread of a machine component supported by two rolling bearings is the distance between the two bearing locations. While the distance between deep groove ball bearings etc. is measured between the bearing centres, the spread with single-row angular contact ball bearings and tapered roller bearings is the distance between the pressure cone apexes.

Static load/static stressing

Static stress refers to bearings carrying a load when stationary (no relative movement between the bearing rings). The term "static", therefore, relates to the operation of the bearings but not to the effects of the load. The magnitude and direction of the load may change. Bearings which perform slow slewing motions or rotate at a low speed (\( n < 10 \text{ min}^{-1} \)) are calculated like statically stressed bearings (cp. Dynamic stressing).

Static load rating \( C_0 \)

The static load rating \( C_0 \) is that load acting on a stationary rolling bearing which causes, at the centre of the contact area between the most heavily loaded rolling element and the raceway, a total plastic deformation of about 1/10,000 of the rolling element diameter. For the normal curvature ratios this value corresponds to a Hertzian contact pressure of about 4,000 N/mm\(^2\) for roller bearings, 4,600 N/mm\(^2\) for self-aligning ball bearings and 4,200 N/mm\(^2\) for all other ball bearings. \( C_0 \) values, see FAG rolling bearing catalogues.

Stress index \( f_* \)

In the attainable life calculation the stress index \( f_* \) represents the maximum compressive stress occurring in the rolling contact areas.

\[
f_* = \frac{C_0}{P_0^*}
\]

\( C_0 \) static load rating [kN]

\( P_0^* \) equivalent bearing load [kN]

\[
P_0^* = X_0 \cdot F_r + Y_0 \cdot F_a
\]

\( F_r \) dynamic radial force [kN]

\( F_a \) dynamic axial force [kN]

\( X_0 \) radial factor (see catalogue)

\( Y_0 \) thrust factor (see catalogue)
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**Synthetic lubricants/synthetic oils**

*Lubricating oils* produced by chemical synthesis; their properties can be adapted to meet special requirements: very low setting point, good V-T behaviour, small evaporation losses, long life, high oxidation stability.

**Tandem arrangement**

A tandem arrangement consists of two or more angular contact bearings which are mounted adjacent to each other facing in the same direction, i.e. asymmetrically. In this way, the axial force is distributed over all bearings. An even distribution is achieved with universal-design angular contact bearings.

**Thermal reference speed**

The thermal reference speed is a new index of the speed suitability of rolling bearings. In the draft of DIN 732, part 1, it is defined as the speed at which the reference temperature of 70 °C is established. In FAG catalogue WL 41 520 the standardized reference conditions are indicated which are similar to the normal operating conditions of the current rolling bearings (exceptions are, for example, spindle bearings, four-point bearings, barrel roller bearings, thrust ball bearings). Contrary to the past (limiting speeds), the thermal reference speed values indicated in the FAG catalogue WL 41 520 now apply equally to oil lubrication and grease lubrication.

For applications where the operating conditions deviate from the reference conditions, the thermally permissible operating speed is determined.

In cases where the limiting criterion for the attainable speed is not the permissible bearing temperature but, for example, the strength of the bearing components or the sliding velocity of rubbing seals the kinematically permissible speed has to be used instead of the thermal reference speed.

**Thermally permissible operating speed**

For applications where the loads, the oil viscosity or the permissible temperature deviate from the reference conditions for the thermal reference speed the thermally permissible operating speed can be determined by means of diagrams.

The method is described in FAG catalogue WL 41 520.

**Thickener**

*Thickener* and *base oil* are the constituents of lubricating greases. The most commonly used thickeners are metal soaps (e.g. lithium, calcium) as well as polyurea, PTFE and magnesium aluminium silicate compounds.

**Thrust bearings**

Bears designed to transmit pure or predominantly thrust loading, with a nominal contact angle $\alpha_0 > 45^\circ$, are referred to as thrust bearings.

The dynamic load rating and the static load rating of thrust bearings refer to pure thrust loads (cp. Radial bearings).

**Tolerance class**

In addition to the standard tolerance (tolerance class PN) for rolling bearings there are also the tolerance classes P6, P6X, P5, P4 and P2 for precision bearings.

The standard of precision increases with decreasing tolerance number (DIN 620).

In addition to the standardized tolerance classes FAG also produces rolling bearings in tolerance classes P4S, SP (super precision) and UP (ultra precision).

**Universal design**

Special design of FAG angular contact ball bearings. The position of the ring faces relative to the raceway bottom is so closely tolerated that the bearings can be universally mounted without shims in $O$, $X$ or tandem arrangement.

Bears suffixed UA are matched together in such a way that unmounted bearing pairs in $O$ or $X$ arrangement have a small axial clearance. Under the same conditions, bearings suffixed UO feature zero axial clearance, and bearings suffixed UL a light preload. If the bearings are given tight fits the axial clearance of the bearing pair is reduced or the preload increased.
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Viscosity

Viscosity is the most important physical property of a lubricating oil. It determines the load carrying capacity of the oil film under elastohydrodynamic lubricating conditions. Viscosity decreases with rising temperature and vice-versa (see V-T behaviour). Therefore it is necessary to specify the temperature to which any given viscosity value applies. The nominal viscosity \( \nu_{40} \) of an oil is its kinematic viscosity at 40 °C.

SI units for the kinematic viscosity are m\(^2\)/s and mm\(^2\)/s. The formerly used unit Centistoke (cSt) corresponds to the SI unit mm\(^2\)/s. The dynamic viscosity is the product of the kinematic viscosity and the density of a fluid (density of mineral oils: 0.9 g/cm\(^3\) at 15 °C).

Viscosity ratio \( \kappa \)

The viscosity ratio, being the quotient of the operating viscosity \( \nu \) and the rated viscosity \( \nu_1 \), is a measure of the lubricating film development in a bearing, cp. factor \( d_{35} \).

Viscosity-temperature behaviour (V-T behaviour)

The term V-T behaviour refers to the viscosity variations in lubricating oils with temperature. The V-T behaviour is good if the viscosity varies little with changing temperatures.

Wear

The life of rolling bearings can be terminated, apart from fatigue, as a result of wear. The clearance of a worn bearing gets too large.

One frequent cause of wear are foreign particles which penetrate into a bearing due to insufficient sealing and have an abrasive effect. Wear is also caused by starved lubrication and when the lubricant is used up. Therefore, wear can be considerably reduced by providing good lubrication conditions (viscosity ratio \( \kappa > 2 \) if possible) and a good degree of cleanliness in the rolling bearing. Where \( \kappa \leq 0.4 \) wear will dominate in the bearing if it is not prevented by suitable additives (EP additives).

X arrangement

In an X arrangement, two angular contact bearings are mounted symmetrically in such a way that the pressure cone apex of the left-hand bearing points to the right and that of the right-hand bearing points to the left.

With an X arrangement, the bearing clearance is obtained by adjusting one outer ring. This ring should be subjected to point load because, being displaceable, it cannot be fitted tightly (Fits). Therefore, an X arrangement is provided where the outer ring is subjected to point load or where it is easier to adjust the outer ring than the inner ring. The effective bearing spread in an X arrangement is less than in an O arrangement.