Life time calculation

Even under suitable operating conditions, and even if the high precision rolling bearings have been mounted properly, a bearing may still fail. The service life of high precision rolling bearings is limited by various factors.

The length of time during which a high precision rolling bearing works satisfactorily is called its "life time". Material fatigue, vibrations, contamination or lubrication-failure may occur during this time span. In the following observations we will be ignoring the causes that can lead to the failure of a high precision rolling bearing without noticeable prior indication; these causes include faulty design, inadequate maintenance, faulty mounting, and wrong bearing dimensioning.

Fatigue

Fatigue of the material that high precision rolling bearings are made of is caused by the rising stress to the material that occurs when high precision rolling bearings rotate under load. Towards the end of the life of the bearing, "pittings" appear on the running surfaces or the rolling elements; an initial occurrence may be followed by progressive pitting of the material.

Various factors are responsible for material fatigue; they can really only be covered with reference to statistics. The definition of a theoretical reference value for "fatigue life time" therefore refers to a very large quantity of rolling bearings running under the same conditions. The fatigue life time is reached as soon as 10% of all bearings have failed. The fatigue life time may be specified with reference to the number of revolutions or to the runtime in hours.

In addition to fatigue life time, lubrication life time is another decisive factor in operating a high precision rolling bearing. Particular rules apply for the calculation of this factor. In the case of high precision rolling bearings that are permanently monitored, other parameters may also be crucial in determining their life time, including temperature, bearing noise and vibrations.

Basic rating life

In terms of pure definition, 10% of a great number of high precision rolling bearings of the same type, running under the same conditions, may fail within a basic rating life of L_{10}. If a constant rotational speed is used, the basic rating life may also be specified in terms of a time $L_{10,h}$.

$$L_{10,h} = \left(\frac{C}{P}\right)^p \cdot \frac{10^6}{60 \cdot n} \qquad [h] \qquad [5.1]$$

$L_{10,h}$	basic rating life	[h]
n	operating speed	[min^{-1}]
C	dynamic load rating	[N]
P	equivalent dynamic bearing load	[N]
p	life time exponent for ball bearing p = 3 for roller bearing p = 10/3	

Dynamic load rating

The dynamic load rating is the constant load of a high precision rolling bearing with a rotating inner ring for which 1 million revolutions will be achieved with a probability of 90%. It is considered as a central radial load that is constant with regard to magnitude and direction. It is based on numerous experiments and is a performance figure that may be calculated with the aid of empirical rules.

Static load rating

The static load carrying capacity of a high precision rolling bearing is defined by the static load rating C_0. This is the load that a stationary high precision rolling bearing can bear with a maximum permanent plastic deformation of 1/10,000 of the rolling element diameter.

Equivalent dynamic bearing load

Combined loads, comprising both axial and radial components, can often act on high precision rolling bearings. In order to relate the forces that actually act on a high precision rolling bearing to the dynamic load rating, the equivalent dynamic bearing load P is calculated from the effective force components. This load is hypothetical and corresponds to the dynamic load rating with regard to the point of application of the load and its effective direction.

$$P = X \cdot F_r + Y \cdot F_a \quad [N] \quad [5.2]$$

P	equivalent dynamic bearing load	[N]
F_r	radial force	[N]
F_a	axial force	[N]
X	radial load factor, see table 5.1	
Y	axial load factor, see table 5.1	

Once you have calculated the equivalent dynamic bearing load it may be mathematically related to the dynamic load rating in order to determine the life time.

Equivalent static bearing load

The necessary static load rating C_0 is calculated from the equivalent static bearing load P_0, which is additionally weighted by the static load safety factor s_0.

$$C_0 = P_0 \cdot s_0 \quad [5.3]$$

C_0	static load rating	[N]
P_0	equivalent static bearing load	[N]
s_0	static load safety factor	

Achieving adequate static load safety depends on the operation of the rolling bearings and on their required running smoothness. High precision angular contact ball bearings require a minimum static safety factor of 1; high precision roller bearings require a minimum static safety factor of 1.5.

If high demands are placed on the running smoothness of the rolling bearing, the above mentioned values for static load safety must be at least doubled. If the rolling bearing is subject to shock impact loads, an additional increase of the static load safety by factor 1.5 is required.

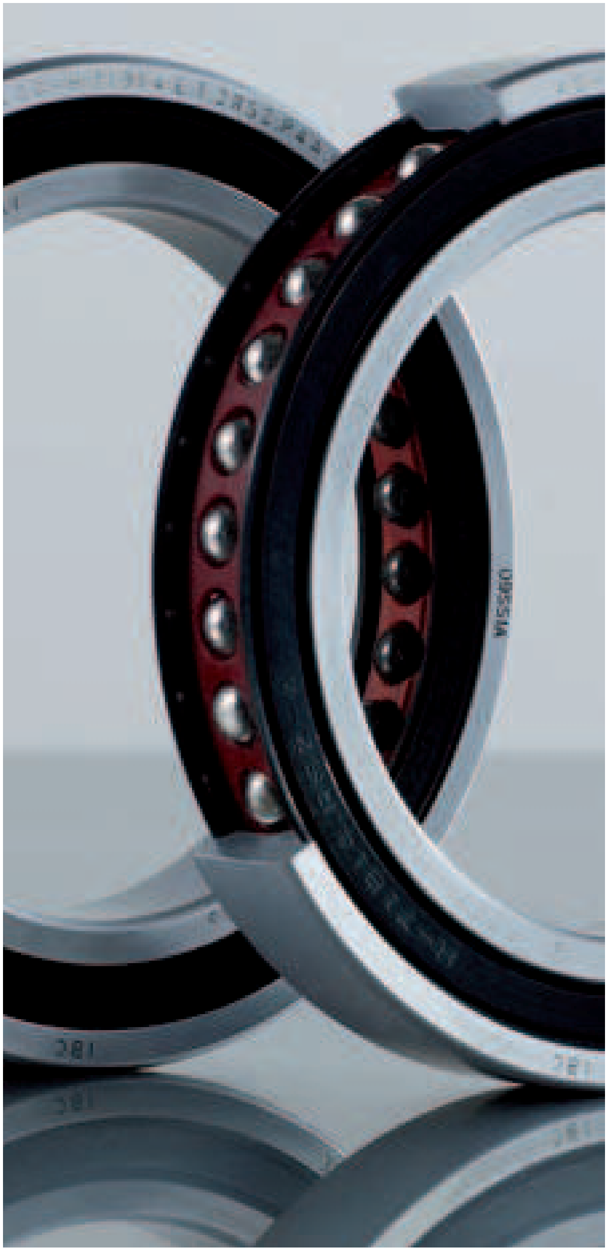
Static load carrying capacity of dynamically designed bearings

If the static load is known, the equivalent static bearing load should be calculated for the type of rolling bearing that was chosen on the basis of the computation of the dynamic load carrying capacity; the equivalent static bearing load should then be checked against the static load safety given above.

Comparing the effective static bearing load with the static load rating also means that the equivalent static bearing load P_0 has to be calculated. Calculation is done in analogy to the calculation of the equivalent dynamic bearing load.

$$P_0 = X_0 \cdot F_r + Y_0 \cdot F_a \quad [N] \quad [5.4]$$

P_0	equivalent static bearing load	[N]
F_r	radial force	[N]
F_a	axial force	[N]
X_0	radial load factor, see table 5.1	
Y_0	axial load factor, see table 5.1	



Determining the axial and radial load factors

To begin with, the axial and radial load factors from table 5.1 are determined for angular contact ball bearings that permit a combined load. The first step for angular contact ball bearings with a contact angle of 15° is to calculate the ratio F_a / i · C_0.

In so doing, i stands for the number of high precision rolling bearings in a set. The bearing-related factor e for single bearings or bearing sets with the same alignment is given in table 5.1a; for high precision rolling bearings that are set against each other, the factor e is given in table 5.1b next to the value that comes as close as possible to the outcome of the calculation F_a / i · C_0. The factor e may also be interpolated in the relevant manner. In the next step, the ratio of the force components F_a / F_r that impact from the outside is calculated and is compared to the value of the bearing-related factor e that is contained in the respective table.

If the ratio is smaller than e, then the axial component is omitted and only the radial load factor X = 1 is retained. If the ratio is greater than e, then the factors X and Y are read off from the same line in the table as their respective factor e. In the case of angular contact ball bearings with a contact angle of 25°, e always has the value of 0.68, and the ratio F_a / i · C_0 is not calculated. The static radial load factor X_0 and the static axial load factor Y_0 may be taken straight from tables 5.1a and 5.1b for the different kinds of contact angles and bearing arrangements. There is no need to calculate these factors.

Contact angle α	F_a / i · C_0	bearing-related factor e	single bearing and tandem mountings < ; << ; <<< ; <<<< $F_a / F_r \le e$ X	Y	$F_a / F_r > e$ X	Y	X_0	Y_0
15°	0.011	0.38	1	0	0.44	1.47	0.5	0.46
	0.022	0.40				1.40		
	0.045	0.43				1.30		
	0.067	0.46				1.23		
	0.089	0.47				1.19		
	0.134	0.50				1.12		
	0.223	0.55				1.02		
	0.334	0.56				1.00		
	0.446	0.56				1.00		
25°		0.68	1	0	0.41	0.87	0.5	0.38

Table 5.1a: Factors for calculating the equivalent bearing load for single rolling bearings and tandem mountings

Contact angle α	F_a / i · C_0	bearing-related factor e	angular contact ball bearings in X- or O-arrangement, or double row high precision rolling bearings <> ; >< $F_a / F_r \le e$ X	Y	$F_a / F_r > e$ X	Y	X_0	Y_0
15°	0.011	0.38	1	1.65	0.72	2.39	1	0.92
	0.022	0.40		1.57		2.28		
	0.045	0.43		1.46		2.11		
	0.067	0.46		1.38		2.00		
	0.089	0.47		1.34		1.93		
	0.134	0.50		1.26		1.82		
	0.223	0.55		1.14		1.66		
	0.334	0.56		1.12		1.63		
	0.446	0.56		1.12		1.63		
25°		0.68	1	0.92	0.67	1.41	1	0.76

Table 5.1b: Factors for calculating the equivalent bearing load for bearing arrangements with a load that impacts symmetrically

Two angular contact ball bearings that are arranged at a distance from one another in an X- or O-arrangement are considered to be a system if there is a symmetrical load application; the system factors for the calculation of the equivalent bearing load are contained in table 5.1b. In all other cases it will be necessary to calculate the individual bearing forces, and the factors will be taken from table 5.1a.

After applying the formulas 5.2 and 5.3, the equivalent dynamic bearing load P and the equivalent static bearing load P_0 will be known.

If the equivalent static bearing load does not exceed the static load rating multiplied by the static load safety s_0, then the single bearing possesses an adequate static dimension. The equivalent dynamic bearing load for a single bearing is worked out using the life time calculation according to equation 5.1.

The dynamic load rating that is necessary for this calculation is contained in the tables in Chapter 2 (IBC High precision angular contact ball bearings) and Chapter 3 (IBC High precision cylindrical roller bearings) for high precision rolling bearings.

Bearing combinations

Should multiple single-row precision angular contact ball bearings of the same kind be fitted next to each other in the same arrangement, the static overall load rating of this combination is calculated from the individual static load ratings (see below):

$$C_{0,Satz} = i \cdot C_{0,Einzellager} \qquad [N] \qquad [5.5]$$

$C_{0,Satz}$	static load rating of bearing set	[N]
i	number of bearings	
$C_{0,Einzellager}$	static load rating of single bearing	[N]

The dynamic load rating of a set is calculated as follows:

$$C_{Satz} = i^{0.7} \cdot C_{Einzellager} \qquad [N] \qquad [5.6]$$

C_{Satz}	dynamic load rating of bearing set	[N]
i	number of bearings	
$C_{Einzellager}$	dynamic load rating of single bearing	[N]

The general reduction in the dynamic load rating of a set by the value $i^{0.7}$ according to DIN ISO 281 is based on the assumption that, within a set, bearings with standard tolerances have bore and outside diameters that differ from each other and therefore carry unequal portions of the load.

Sorted and marked high precision angular contact ball bearings with narrower tolerances carry loads in a much more equal way and therefore result in better operational safety.

The life time of a set of high precision rolling bearings can be calculated if the effective forces that act on each single bearing are known, as the failure of a single high precision rolling bearing within the system leads to the failure of the entire system.

As each single high precision rolling bearing has a 90% probability of reaching its life time, the failure probability of the overall system is the product of the failure probabilities of the single high precision rolling bearings.

In effect, this means that the life time of the overall system is shorter than the shortest life time of any high precision rolling bearing that is part of the system. Taking into account the life times of the single high precision rolling bearings, a system life time $L_{10h,ges}$ may be calculated using equation 5.7.

$$L_{10h,ges} = \frac{1}{\left(\frac{1}{L_{10h,1}^{1.1}} + \frac{1}{L_{10h,2}^{1.1}} + \ldots + \frac{1}{L_{10h,n}^{1.1}}\right)^{\frac{1}{1.1}}} \qquad [h] \qquad [5.7]$$

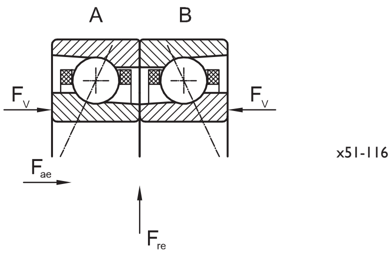
A note on bearing sets with inner preload

The method described above is not valid for preloaded high precision rolling bearings, which are often set against each other in bearing sets, because the preload of these bearings stresses the rolling bearings in addition to the external forces. In order to calculate the life time of a preloaded set of high precision rolling bearings one needs to look at the forces acting on each individual bearing. A method for calculating the life time that takes into account preloading is described on the following pages.

Detailed calculation of bearing forces

In order to determine the radial and axial load components F_r and F_a of each bearing in an arrangement of preloaded angular contact ball bearings, the external radial loads F_{re}, the external axial loads F_{ae} and the axial preload F_v as well as their load allocation, have to be taken into account. The load allocation may be calculated from the spring compression of the rolling elements that conforms to a power law with the exponent 2/3; the assumption is that the radial force is equally divided between the single rolling bearings.

A bearing set with two high precision angular contact ball bearings



Due to the wedge effect produced by contact angle α, an external radial load F_{re} changes the whole preload F_v in the following way:

$$F_{v,ges} = \frac{F_{re} \cdot 1.2 \cdot \tan\alpha + F_v}{2} \quad [N] \quad [5.8]$$

if $F_{v,ges} < F_v$, than apply as correction $F_{v,ges} = F_v$

The axial loads of the individual bearings A and B can now be calculated with the adjusted preload $F_{v,\,ges}$:

$$F_{a,A} = \frac{2}{3} \cdot F_{ae} + F_{v,ges} \quad [N] \quad [5.9]$$

$$F_{a,B} = F_{v,ges} - \frac{1}{3} \cdot F_{ae} \quad [N] \quad [5.10]$$

If the result is $F_{a,A}$ or $F_{a,B} < 0$ the respective high precision rolling bearing is in an unloaded state. The axial load then equals zero, and the axial load of the other bearing corresponds to the external axial load F_{ae}.

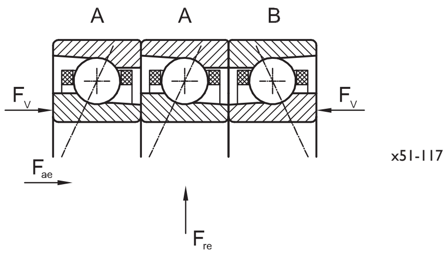
The external radial load F_{re} is now distributed for the single bearings in accordance with the power law of the spring compression mentioned above, taking into account the allocation of the axial loads.

$$F_{r,A} = \frac{F_{a,A}^{2/3}}{F_{a,A}^{2/3} + F_{a,B}^{2/3}} \cdot F_{re} \quad [N] \quad [5.11]$$

$$F_{r,B} = \frac{F_{a,B}^{2/3}}{F_{a,A}^{2/3} + F_{a,B}^{2/3}} \cdot F_{re} \quad [N] \quad [5.12]$$

From the axial loads $F_{a,A}$ and $F_{a,B}$ and the radial loads $F_{r,A}$ and $F_{r,B}$ which have been obtained by the above method, the equivalent dynamic bearing load for each high precision rolling bearing is now determined in accordance with equation 5.2. Factors X and Y are taken from table 5.1a. The application of equation 5.1 gives an individual life time for each high precision rolling bearing, and these life time values are then combined in accordance with equation 5.7 to obtain the system life time.

A bearing set with three high precision angular contact ball bearings



Taking into account an external radial load, the following is true for the preload of the individual high precision rolling bearings:

$$F_{v,A} = \frac{F_{re} \cdot 1.2 \cdot \tan\alpha + F_v}{4} \quad [N] \quad [5.13]$$

$$F_{v,B} = \frac{F_{re} \cdot 1.2 \cdot \tan\alpha + F_v}{2} \quad [N] \quad [5.14]$$

if $F_{v,A} < F_v / 2$, then apply as correction $F_{v,A} = F_v / 2$
and $F_{v,B} < F_v$ then apply as correction $F_{v,B} = F_v$

Therefore the axial loads of the individual high precision rolling bearings are obtained from the following calculation:

$$F_{a,A} = 0.4 \cdot F_{ae} + F_{v,A} \quad [N] \quad [5.15]$$

$$F_{a,B} = F_{v,B} - 0.2 \cdot F_{ae} \quad [N] \quad [5.16]$$

If $F_{a,B} < 0$ then there is no more preload. The axial load of high precision rolling bearing B is zero, and the two high precision rolling bearings A each carry half the external axial load. In the opposite case, both high precision rolling bearings A are unloaded and are no longer subject to axial load. In this case, high precision rolling bearing B carries the full external axial load F_{ae}.

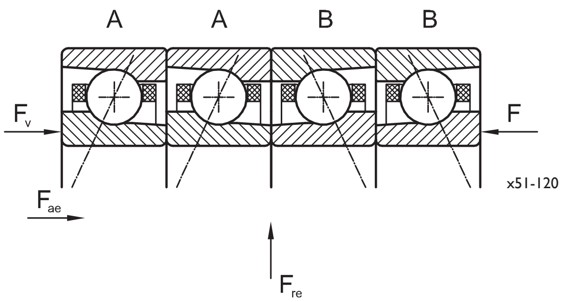
The radial load rate for each high precision rolling bearing with regard to the external radial force is as follows:

$$F_{r,A} = \frac{F_{a,A}^{2/3}}{2 \cdot F_{a,A}^{2/3} + F_{a,B}^{2/3}} \cdot F_{re} \qquad [N] \quad [5.17]$$

$$F_{r,B} = \frac{F_{a,B}^{2/3}}{2 \cdot F_{a,A}^{2/3} + F_{a,B}^{2/3}} \cdot F_{re} \qquad [N] \quad [5.18]$$

As in the case mentioned above, the equivalent dynamic bearing load is obtained by further calculation using equation 5.2 in conjunction with table 5.1a that contains the axial and radial load factors. From there, equation 5.1 leads to the life time of the individual bearings, and equation 5.7 then produces the life time of the overall system.

A bearing set with four high precision angular contact ball bearings



Because of the contact angle of angular contact ball bearings, the radial load influences the preload of the individual high precision rolling bearings as follows:

$$F_{v,ges} = \frac{F_{re} \cdot 1.2 \cdot \tan\alpha + F_v}{4} \qquad [N] \quad [5.19]$$

if $F_{v,ges} < F_v / 2$, than apply as correction $F_{v,ges} = F_v / 2$

The axial loads of the individual high precision rolling bearings are thus obtained by doing the following calculation:

$$F_{a,A} = \frac{1}{3} \cdot F_{ae} + F_{v,ges} \qquad [N] \quad [5.20]$$

$$F_{a,B} = F_{v,ges} - \frac{1}{6} \cdot F_{ae} \qquad [N] \quad [5.21]$$

If $F_{a,B} < 0$ then there is no more preload. The axial load of high precision rolling bearings B is zero, and the two high precision rolling bearings A each carry half the external axial load. In the opposite case, both high precision rolling bearings A are unloaded and are no longer subject to axial load. In this case the high precision rolling bearings B each carry half the axial load F_{ae}.

The radial load rate for each high precision rolling bearing with regard to the external radial force is as follows:

$$F_{r,A} = \frac{F_{a,A}^{2/3}}{F_{a,A}^{2/3} + F_{a,B}^{2/3}} \cdot \frac{F_{re}}{2} \qquad [N] \quad [5.22]$$

$$F_{r,B} = \frac{F_{a,B}^{2/3}}{F_{a,A}^{2/3} + F_{a,B}^{2/3}} \cdot \frac{F_{re}}{2} \qquad [N] \quad [5.23]$$

Further calculation is carried out by applying equation 5.2 in conjunction with table 5.1a, as well as equation 5.1 and 5.7; this gives the life time of the overall bearing system.

Further combinations of angular contact ball bearings

Departing from the above calculation, and taking into account the uneven distribution of the load in accordance with equation 5.24, if axial load predominates, the calculation of the life time may be carried out with defined load parameters according to table 5.2.

In order to determine the resulting axial bearing load, the bearing preload F_v needs to be taken into account along with the external load F_{ae}. In bearing arrangements with an unequal number of bearings per load direction, this results in variable axial stiffness, axial load ratings and life times, depending on the number of bearings for each direction. The proportion of the preload and the external load that needs to be taken into account for each bearing is contained in table 5.2.

For bearing sets made up of universal bearings with more than two bearings and a rigid preload F_v that cannot to be ignored, the life time for each single bearing should be calculated as follows.

The radial load is distributed between all the high precision rolling bearings in a set:

$$F_{r,\,Einzellager} = \frac{F_r}{i_{ges}^{0.7}} \qquad [N] \quad [5.24]$$

Number of bearings in a set

i	1	2	3	4	5	6
$i^{0.7}$	1	1.62	2.12	2.64	3.09	3.51

The axial load $F_{a,Einzellager}$ results from the formulas in accordance with table 5.2. The high precision rolling bearings in load direction experience the greatest stress. The high precision rolling bearings in counter-load direction only carry part of the preload or are completely unloaded.

The equivalent bearing loads P are then determined in each case with the help of the resulting bearing loads $F_{r,\,Einzellager}$ and $F_{a,\,Einzellager}$ according to formula 5.2.

If a variable spring preload is used instead of a rigid preload, the following is true for the high precision rolling bearing (or bearing set) that experiences more stress:

$$F_a = F_{Feder} + F_{ae} \quad [N] \quad [5.25]$$

$$F_{a,Einzellager} = \frac{1}{i^{0.7}} \cdot (F_{Feder} + F_{ae}) \quad [N] \quad [5.26]$$

Load spectrums

For a load spectrum made up of forces that change over time, and possibly including variable speeds, a mean equivalent bearing load is determined in accordance with equation 5.27.

The mean rotational speed itself is also made up of portions of time for the respective speeds that are expressed as a percentage.

$$P_m = \sqrt[3]{\frac{P_1^3 \cdot t_1 \cdot n_1 + \ldots + P_n^3 \cdot t_n \cdot n_n}{n_m \cdot 100}} \quad [N] \quad [5.27]$$

$$n_m = \frac{t_1 \cdot n_1 + \ldots + t_n \cdot n_n}{100} \quad [min^{-1}]$$

P_m	mean bearing load	[N]
$P_1 \ldots P_n$	equivalent load per load case	[N]
$t_1 \ldots t_n$	time portion of bearing load	[%]
$n_1 \ldots n_n$	service speeds	[min^{-1}]
n_m	mean operating speed	[min^{-1}]

Determining the bearing size

After preselecting a possible arrangement of high precision angular contact ball bearings, and after applying the equations mentioned above to the given external bearing loads and to the assumed preload of the bearings, all individual equivalent dynamic bearing loads P are known.

After adjusting equation 5.1 as follows

$$C = P \cdot \sqrt[p]{\frac{L_{10,h} \cdot 60 \cdot n}{10^6}} \quad [N] \quad [5.28]$$

C	dynamic load rating	[N]
P	equivalent dynamic load	[N]
p	life time exponent	
$L_{10,h}$	basic rating life	[h]
n	operating speed	[min^{-1}]

the required dynamic load rating may be determined by assuming a life time for a given speed; the dynamic load rating may then be used for purposes of comparison with the table.

Load direction	Arrangement bearing position		Load direction	Unloading from $F_{ae} > X \cdot F_v$	Load distribution with regard to single bearing ($F_{a,\ Einzellager}$) taking into account preload F_v and external load F_{ae}: up to unloading through external load $F_{ae} < X \cdot F_v$		after unloading at $F_{ae} > X \cdot F_v$	
	A	B		X	A	B	A	B
$F_{ae} \rightarrow$	<	>		2.83	$F_v + 0.67\ F_{ae}$	$F_v - 0.33\ F_{ae}$	F_{ae}	
$F_{ae} \rightarrow$	<<	>		5.66	$0.84\ F_v + 0.47\ F_{ae}$	$1.36\ F_v - 0.24\ F_{ae}$	$0.617\ F_{ae}$	0
	<<	>	$\leftarrow F_{ae}$	2.83	$0.84\ F_v - 0.30\ F_{ae}$	$1.36\ F_v + 0.52\ F_{ae}$	0	F_{ae}
$F_{ae} \rightarrow$	<<<	>		8.49	$0.73\ F_v + 0.38\ F_{ae}$	$1.57\ F_v - 0.18\ F_{ae}$	$0.463\ F_{ae}$	
	<<<	>	$\leftarrow F_{ae}$	2.83	$0.73\ F_v - 0.26\ F_{ae}$	$1.57\ F_v + 0.45\ F_{ae}$	0	F_{ae}
$F_{ae} \rightarrow$	<<<<	>		11.3	$0.65\ F_v + 0.32\ F_{ae}$	$1.71\ F_v - 0.15\ F_{ae}$	$0.379\ F_{ae}$	
	<<<<	>	$\leftarrow F_{ae}$	2.83	$0.65\ F_v - 0.23\ F_{ae}$	$1.71\ F_v + 0.45\ F_{ae}$	0	F_{ae}
$F_{ae} \rightarrow$	<<	>>		5.66	$0.84\ F_v + 0.40\ F_{ae}$	$0.84\ F_v - 0.22\ F_{ae}$	$0.617\ F_{ae}$	0
$F_{ae} \rightarrow$	<<<	>>		8.49	$1.12\ F_v + 0.33\ F_{ae}$	$1.49\ F_v - 0.18\ F_{ae}$	$0.463\ F_{ae}$	0
	<<<	>>	$\leftarrow F_{ae}$	5.66	$1.12\ F_v - 0.20\ F_{ae}$	$1.49\ F_v + 0.35\ F_{ae}$	0	$0.617\ F_{ae}$
$F_{ae} \rightarrow$	<<<<	>>		11.3	$1.03\ F_v + 0.29\ F_{ae}$	$1.68\ F_v - 0.15\ F_{ae}$	$0.379\ F_{ae}$	0
	<<<<	>>	$\leftarrow F_{ae}$	8.49	$1.03\ F_v - 0.18\ F_{ae}$	$1.68\ F_v + 0.33\ F_{ae}$	0	$0.617\ F_{ae}$

Table 5.2: Resulting axial load F_a of the single bearing

Expanded modified life time calculation

Over the years, the method of calculating the life time of a high precision rolling bearing has become more sophisticated by the inclusion of new criteria. The calculating method that was applied used to consist solely of a function of load rating C, of the equivalent dynamic bearing load P and of the mean operating speed n_m. But various other life time coefficients have been added over time.

Expanded modified life time calculation

The so-called modified life time L_{na} with its factors a_1, a_2 and a_3 that was commonly used in the past has been superseded by the expanded modified life time calculation $L_{10,nm}$ according to DIN ISO 281 as of 1993.

$$L_{10,nm} = a_1 \cdot a_{DIN} \cdot L_{10} \quad [h] \quad [5.29]$$

$L_{10,nm}$	expanded modified life time	[h]
a_1	life time expectation	
a_{DIN}	life cycle coefficient as in equation 5.37 to 5.42	
L_{10}	life time as in equation 5.1	[h]

life time expectation a_1		
life time expectation %	L_{na}	a_1
90	L_{10a}	1.00
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

Table 5.3: Coefficients a_1

The probability factor a_1 remains. This factor permits the conversion of the life time expectation of 90%, or the failure rate of 10%, to other, higher life time expectations. The a_{DIN} factor that has now been introduced covers various parameters that are listed below.

In addition to load, the following parameters are taken into account, which in turn are based on multiple factors:

Parameter	Influenced by
Lubrication	bearing size, speed, viscosity and type of lubricant, additives
Material	surface, purity, hardness, temperature resistance, fatigue limit
Bearing design	friction conditions, internal load distribution
Tension	manufacturing, heat treatment, press fit
Bearing ambience	moisture, contamination of lubricant
Mounting	displacement, damage

Table 5.4: Parameters

The expanded modified life time calculation is based on the basic rating life L_{10} in accordance with equation 5.10, weighted by the life time expectation factor a_1 in accordance with table 5.3 and the life cycle coefficient a_{DIN}.

Determining the life cycle coefficient

The life cycle coefficient a_{DIN} is read off from diagrams 5.3 and 5.4, after the parameters $e_c \cdot P_u / P$ and κ have been determined, or it is calculated in accordance with equations 5.37 to 5.42.

$$a_{DIN} = f\left(\frac{e_c \cdot P_u}{P}, \kappa\right) \quad [5.30]$$

a_{DIN}	life cycle coefficient	
e_c	contamination coefficient (table 5.5)	
P_u	fatigue limit load	[N]
P	equivalent dynamic bearing load	[N]
κ	viscosity ratio	

Fatigue limit load

This takes into account the fatigue limit of the ring material. Without going into detail here, the fatigue limit load is roughly defined with the aid of the following formula for rolling bearings up to a mean bearing diameter d_m of 150 mm:

$P_u \cong C_0 / 27$	ball bearing	[N]	[5.31]
$P_u \cong C_0 / 8.2$	roller bearing	[N]	[5.32]
P_u	fatigue limit load	[N]	

Contamination coefficient

Hard and firm contaminants that are present in the lubricant can lead to lasting impressions being made in the raceways during the rolling-over process. Local stress peaks caused by over rolling the contaminations reduce the life time of the high precision rolling bearings. A reduction in life time due to hard particles will depend on the size of the bearing, the lubricant film thickness (viscosity ratio κ) and on the size, type, hardness and number of particles. Other sorts of contamination, like the inflow of fluids, cannot, however, be considered in connection with this coefficient.
If severe contamination occurs ($e_c \rightarrow 0$), failure due to wear will become a likelihood. The actual life time will then be much shorter than the calculated life time.

Level of contamination	Coefficient e_c	
	D_{pw} < 100 mm	D_{pw} > 100 mm
Extreme cleanness Particle size same as lubricant film thickness, laboratory conditions	1	1
High cleanness Extra-fine filtering of oil feeding, sealed, greased bearings	0.8...0.6	0.9...0.8
Normal cleanness Fine filtering of oil feeding, greased bearings with shields	0.6...0.5	0.8...0.6
Light contamination Light contaminations in oil feeding	0.5...0.3	0.6...0.4
Moderate contamination Bearing is contaminated with abrasion from other machine components	0.3...0.1	0.4...0.2
Severe contamination Heavily polluted bearing ambience, inadequate sealing	0.1...0	0.1...0
Very severe contamination	0	0

Table 5.5: Contamination coefficients e_c

Viscosity ratio

The viscosity ratio κ is used to rate the quality of the lubricant film formation. It defines the ratio of lubricant viscosity ν at operating temperature to the reference viscosity ν_1.
With κ = 1, the separative lubricant film will be exactly achieved. Procedural method: with the help of diagram 5.1 initially determine the reference viscosity ν_1 subject to the reference diameter D_{pw} and the speed n.

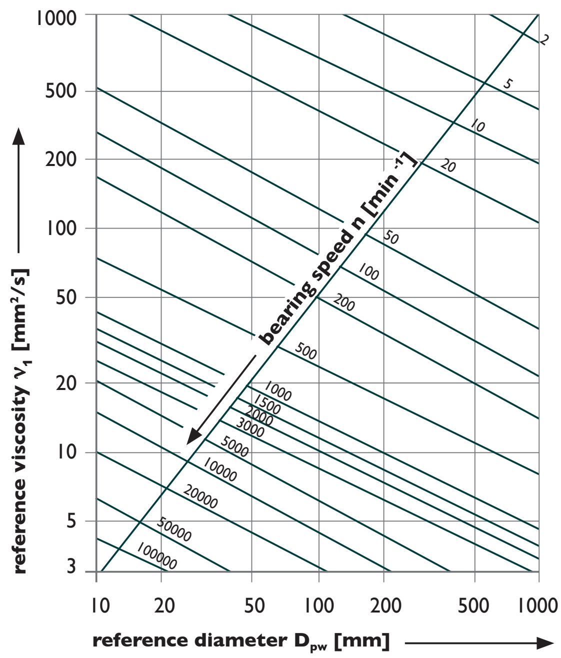


Diagram 5.1: Required kinematic viscosity ν_1

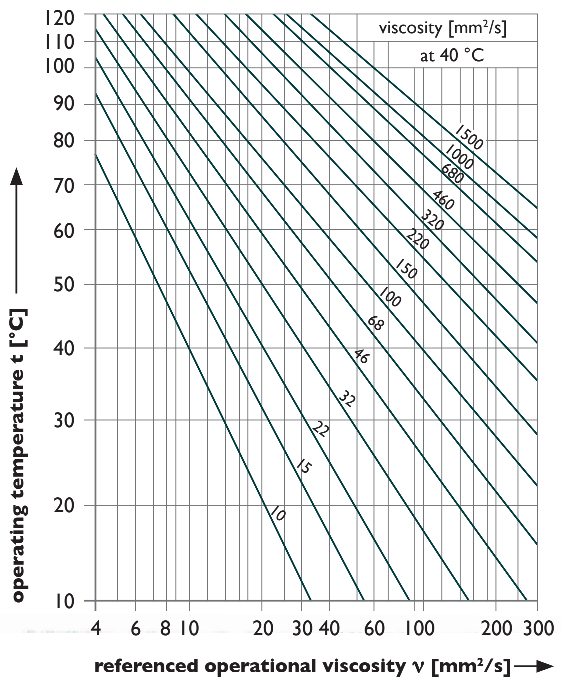


Diagram 5.2: Viscosity at operating temperature for mineral oils

From diagram 5.2, relating to viscosity and temperature, the referenced operational viscosity ν is then read off below the point at which the anticipated operating temperature t intersects with the diagonal graph of the reference viscosities at 40 °C.

From this, the viscosity ratio κ is then established:

$$\kappa = \frac{\nu}{\nu_1} \qquad [5.33]$$

κ	viscosity ratio	
ν	referenced operational viscosity	[mm²/s]
ν_1	reference viscosity	[mm²/s]

In the mixed friction range $\kappa < 1$, higher values may be achieved with the help of suitable additives: Wear can be reduced, corrosion be counteracted, friction reduced and the adhesion of the lubricant in the lubrication gaps improved with the help of special additives like solids,or polar and/or polymer active substances.

Additives should be used if the referenced operational viscosity for ball bearings is < 13 mm²/s, if it is < 20 mm²/s for roller bearings, and if the basic speed rating is relatively low, at $d_m \cdot n < 10{,}000$.

The viscosity ratio κ, in accordance with equation 5.33, is determined by means of diagrams 5.1 and 5.2; but it may also be calculated.

The following is valid for the reference viscosity ν_1:

$$\nu_1 = 45000 \cdot n^{-0.83} \cdot D_{pw}^{-0.5} \quad \text{for } n < 1000 \text{ min}^{-1} \qquad [5.34]$$

$$\nu_1 = 4500 \cdot n^{-0.5} \cdot D_{pw}^{-0.5} \quad \text{for } n > 1000 \text{ min}^{-1} \qquad [5.35]$$

D_{pw} reference diameter of the high precision rolling bearing $= d_m = (d + D)/2$ [mm]

For lubricants with a density that deviates from the reference density of ρ_1 = 0.89 g/cm³ at 20 °C the following formula applies:

$$\kappa = \frac{\nu}{\nu_1} \cdot \left(\frac{\rho}{\rho_1}\right)^{0.83} \qquad [5.36]$$

ρ	density of lubricant used	[g/cm³]
ρ_1	reference density	[g/cm³]

For a viscosity ratio $\kappa < 1$ and a contamination coefficient $e_c > 0.2$ the value $\kappa = 1$ in calculations can be used, if a lubricant with active EP additives is used. The life cycle coefficient must then, however, be limited to $a_{DIN} < 3$. In the case of severe contamination ($e_c < 0.2$) the effectiveness of the use of additives must be demonstrated.

The determining of the value a_{DIN} for IBC high precision angular contact ball bearings and IBC high precision cylindrical roller bearings by graph and by calculation is summarised on the following page.

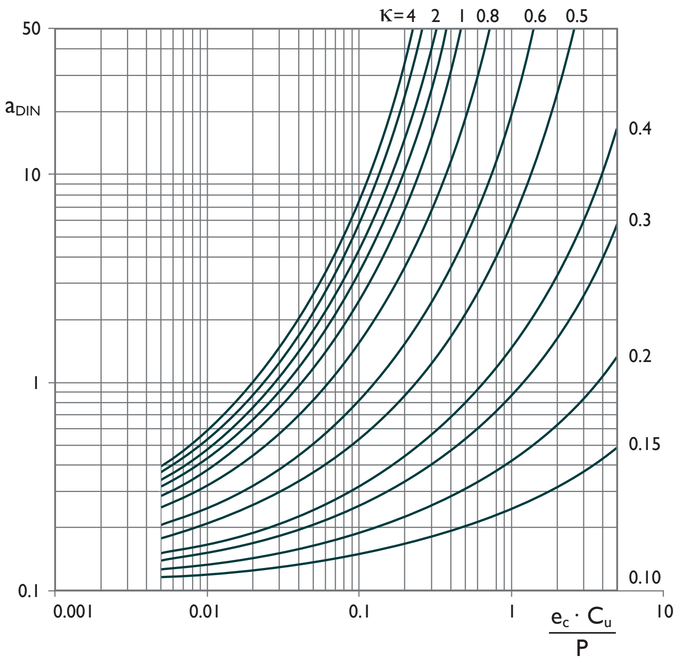


Diagram 5.3: Life cycle coefficient a_{DIN} for IBC high precision angular contact ball bearings

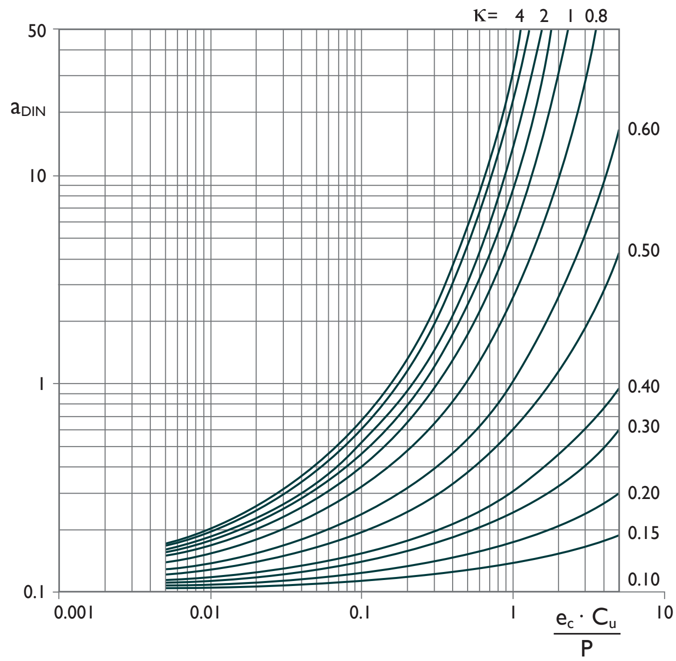


Diagram 5.4: Life cycle coefficient a_{DIN} for IBC high precision cylindrical roller bearings

Calculation rule for a_{DIN} for IBC High precision angular contact ball bearings:

$$a_{DIN} = 0.1 \cdot \left[1 - \left(2.56705 - \frac{2.26492}{\kappa^{0.0543806}}\right)^{0.83} \cdot \left(\frac{e_c \cdot C_u}{P}\right)^{\frac{1}{3}}\right]^{-9.3}$$ for $0,1 \leq \kappa < 0.4$ [5.37]

$$a_{DIN} = 0.1 \cdot \left[1 - \left(2.56705 - \frac{1.99866}{\kappa^{0,190870}}\right)^{0.83} \cdot \left(\frac{e_c \cdot C_u}{P}\right)^{\frac{1}{3}}\right]^{-9.3}$$ for $0.4 \leq \kappa < 1$ [5.38]

$$a_{DIN} = 0.1 \cdot \left[1 - \left(2.56705 - \frac{1.99866}{\kappa^{0.0717391}}\right)^{0.83} \cdot \left(\frac{e_c \cdot C_u}{P}\right)^{\frac{1}{3}}\right]^{-9.3}$$ for $1 \leq \kappa \leq 4$ [5.39]

Calculation rule for a_{DIN} for IBC High precision cylindrical roller bearings:

$$a_{DIN} = 0.1 \cdot \left[1 - \left(1.58592 - \frac{1.39926}{\kappa^{0.0543806}}\right)^{0.83} \cdot \left(\frac{e_c \cdot C_u}{P}\right)^{0.4}\right]^{-9,185}$$ for $0,1 \leq \kappa < 0.4$ [5.40]

$$a_{DIN} = 0.1 \cdot \left[1 - \left(1.58592 - \frac{1.23477}{\kappa^{0,190870}}\right)^{0.83} \cdot \left(\frac{e_c \cdot C_u}{P}\right)^{0.4}\right]^{-9,185}$$ for $0.4 \leq \kappa < 1$ [5.41]

$$a_{DIN} = 0.1 \cdot \left[1 - \left(1.58592 - \frac{1.23477}{\kappa^{0.0717391}}\right)^{0.83} \cdot \left(\frac{e_c \cdot C_u}{P}\right)^{0.4}\right]^{-9,185}$$ for $1 \leq \kappa \leq 4$ [5.42]

IBC-specific factors

For options that exceed the expanded life time calculation according to DIN ISO 281, IBC has defined further factors that lead to a different life time and that may be taken into account optionally. These factors are assumed under “material-related factors” a_{lb} and a_{wk}.

The factor a_{tr} recognises the positive characteristics of the ATCoat coating if the coating is applied to the raceways. (This type of coating may also be applied solely to the outer diameter in order to eliminate fretting corrosion to the rolling bearing); see Chapter 9 (Materials).

The factor a_{wk} makes reference to the material of the rolling elements and takes into account the considerably longer life time of ceramic rolling elements; there are several reasons for this.

Material-related factors			
Track material	**a_{lb}**	**Rolling element material**	**a_{wk}**
uncoated	1.00	100Cr6	1.0
IR ATCoat	1.15	Si_3N_4	2.0
OR ATCoat	1.05		
IR & OR ATCoat	1.20		

Table 5.6: Coefficients a_{lb} and a_{wk}

When using rolling bearings that have been thermally treated to a higher degree, all components used must be matched to the continuous operation temperature.

The IBC-specific factors enter into the calculation of the expanded life time in multiplicative form:

$$L_{10,erw.,IBC} = a_{lb} \cdot a_{wk} \cdot L_{10,nm} \quad [h] \qquad [5.43]$$

$L_{10,erw.,IBC}$	IBC-specific modified life time	[h]
a_{lb}, a_{wk}	material-related factors	
$L_{10,nm}$	modified life time	[h]

In addition, the life time of the grease should be compared to the calculated life time of the high precision rolling bearings in order to ascertain if permanent lubrication is an option, or in order to develop a strategy for continuous or cyclical re-lubrication.