5. DETERMINING BEARING SIZE

5.1 General information

A properly installed and lubricated roller-contact bearing will operate under normal conditions, i.e. absent extreme speeds and temperatures, until it fails due to fatigue of materials at acting surfaces. Repeated stress on the contact surfaces between roller-contact surfaces and rings will manifest after a certain period depending on the magnitude of load as a stress fracture. This will expand until a part of the bearing ring material or roller element material breaks off (pitting) and causes failure. Many bearings are also discarded for other reasons than material fatigue, but these failures can be avoided if the bearing is treated properly, if it is properly installed, lubricated, and overloading is avoided.

When a certain number of identical bearings are tested for fatigue under specified operating conditions (load and rpm), there is a large variance of durability between individual bearings. In a group of 30 or more bearings, the ratio between the shortest and longest durability can be 20-fold or more. A durability variance curve can be drawn for each tested group of bearings that illustrates the relationship between the durability and the number of bearings, which were discarded.

The required bearing size is determined on the basis of externally acting forces and based on the durability and reliability demands of the seated bearing. The size, direction, purpose, and nature of the bearing load as well as the revolution operating speed are determinant when selecting the bearing type and size. Meanwhile, other special or important conditions of each individual case must be considered, e.g. operating temperature, spatial allowances, ease of installation, lubrication requirements, packing, etc., which can affect the selection of the most suitable bearing. Various types of bearings may, in many cases, be suitable for the given specific conditions.

In terms of the action of external forces and the function of the bearing in the respective node or unit, we distinguish two types of roller bearing loads in bearing technology:

- If the bearing rings turn in relation to one another and the bearing is exposed, under such conditions, to external forces (which applies for the majority of bearing applications), we refer to this as a dynamic bearing load,

- If the bearing rings do not turn in relation to one another or turn very slowly, the bearing transmits oscillating motion, or external forces act for shorter period than the time of one bearing revolution, we refer to this as a static bearing load.

The durability limited by failure of a particular bearing component (bearing rings, roller elements, cage, lubricant and seal) is, in the first case, decisive for calculating bearing safety. In the second case, permanent deformities of functional surfaces at contact points between rolling elements and orbits is decisive.

5.2 Roller bearing reliability

The reliability of a group of apparently identical roller bearings, operating under identical conditions, is the percentage of the group, expected to achieve or exceed the specified durability.

The reliability of an individual roller bearing is the probability that the bearing will achieve or exceed the specified durability.

The equation for calculating durability includes the effect of stress induced by external loads, lubrication, and surface kinematics at the site of rolling contact. Including the impact of the comprehensive system of stress on bearing durability makes it possible to better anticipate the actual manner, in which a bearing behaves within a specific housing. International standards, such as e.g. ISO 281, are based on the theory of material fatigue at the site of rolling contact. One must keep in mind that a complete bearing can be considered as a system, the individual components of which (bearing rings, rolling elements, cage, lubricant and seal) have the same effect on durability and, in certain cases, are even a decisive factor in determining the bearing durability during operation. The optimal operating durability is theoretically achieved when all of the components achieve the same durability. In other words, the calculated durability corresponds to the actual operating durability if the operating durability of related components is at least as long as the calculated bearing durability. Related components in such case are the cage, seal and lubricant. The most important factor in practise is metal fatigue.

5.3   Dynamic Load Capacity

Dynamic load capacity is, according to ISO 281:1990, a constant invariable load that a bearing can theoretically carry at a basic durability of one million revolutions.

The dynamic load capacity Cr for radial bearings relates to constant, invariable, entirely radial loads. For thrust bearings, the dynamic load capacity Ca relates to the invariable, purely axial load acting in the bearing's axis.

The dynamic load capacity Cr and Ca, whose magnitude depends on the bearing dimensions, the number of rolling elements, the bearing material and design, is provided in the table for each bearing. The dynamic load capacity values were determined in accordance with ISO standard 280. These values are verified on testing equipment and confirmed in operating results.

The numeric values specified in this catalogue apply for chrome steel bearings, heat treated to a minimal hardness of 58 HRC and normal operating conditions. NEW FORCE bearings display, among others, improved material properties and advanced manufacturing processes. To determine the dynamic load capacity in these bearings, thus requires the use of correction factors according to ISO 281. More information about these bearings is available in separate chapter 7.7.
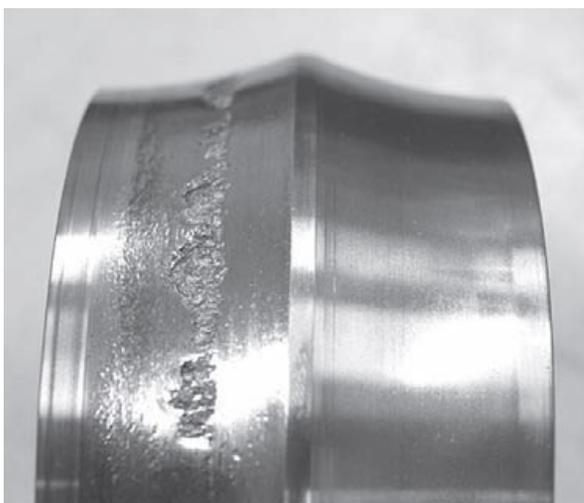

Fig. 5.1 Photo-illustration of fatigue damage on the raceway

Fig. 5.2 Photo-illustration of fatigue damage on the raceway

5.4 Durability

It is the number of revolutions that a bearing lasts, before fatigue of one of its components occurs, which manifests as flaking of material. It is expressed either as the total number of revolutions or operating hours, or in vehicles, by the distance travelled (number of driven km).

The material is primarily responsible for significant variance in durability in a broader range of identical bearings tested under the same conditions. No material or bearing steel is entirely homogenous and contains certain weak points. If a weak point is located on the orbit, where large load (stress) is generated, then the durability of the bearing will be small. The durability is higher where the load is decreased. Poor material has a large amount of weak points and, in all likeliness, some of them lie in areas of greatest load. The variance of durability will thus be less in poor material and larger in first-class material.

Variance of durability is also affected by manufacturing tolerances of individual components. The tolerances of roller diameters and radiuses of raceways significantly affect loads on roller surfaces. For manufacturing reasons, the radial clearance in a bearing varies within a specific tolerance, and as such, it also affects the distribution of pressure on individual roller elements. The distribution of forces within the bearing in the same manner cause expansion and decrease the orbit diameter due to the prescribed placement of rings on the shaft and within the housing.

Adherence to the prescribed material composition, its purity, and heat treatment is also an important indicator of bearing quality. Large variances in durability of large quantities of identical roller bearings, testing under identical conditions is but a natural consequence of the specified individual influences. Current research shows that even the quality of lubrication, its purity, and quantity may significantly impact bearing durability. Lubrication is taken into account in the modified durability calculation, see further.

The results of performed durability tests and practical operating experiences indicate that identical bearings, operating under identical conditions, do not achieve the same durability. The term "durability" must thus be correctly defined.

5.5 Basic durability equation

The basic durability of a bearing is mathematically defined by the durability equation, which applies for all types of bearings.

$$L_{10} = \left(\frac{C}{P}\right)^{p} \qquad \text{or} \qquad \frac{C}{P} = \left(L_{10}\right)^{1/p}$$

L_{10} . . . . . . basic durability [10^6 rev]

C . . . . dynamic load capacity [kN]
(the C_r and C_a values are specified in the product section of the catalogue)

P . . . . bearing equivalent dynamic load capacity [kN]
(the equations for calculating P_r and P_a are provided in the chapter Equivalent Dynamic Load Capacity and for each structural group of bearings)

p . . . . ball bearing exponent $p = 3$

 . . . . For cylindrical roller, needle roller, spherical-roller, and tapered-roller bearings $p = \dfrac{10}{3}$
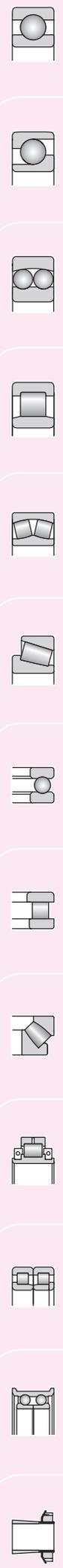
The basic durability of a bearing is thus understood to mean the durability that 90 % of bearings achieve or exceed from a set of identical bearings, working under the same operating conditions. All standard durability calculations are performed for this reliability level. Mean durability L_s is the durability that 50 % of bearing from the same set achieve; it is about 5 times higher than the basic durability. In contrast, the durability achieved by 99 % of bearings is about one fifth when compared with the basic durability. The impact of the degree of reliability on the durability calculation is specified in chapter 5.6.

Table 5.1 lists the relationship of durability L_{10} in millions of revolutions and the corresponding C/P ratio. If the revolution speed is unchanged, then the durability can be calculated using the modified equation, which expresses the basic durability in terms of operating hours:

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

L_{10h} . . . . . basic durability   [h]

n . . . . revolution speed   [min^{-1}]

The relationship of the C/P ratio on basic durability L_{10h} and on the revolution speed for ball bearings is specified in table 5.2 and in table 5.3 for cylindrical roller, needle, spherical-roller, and tapered-roller bearings.

In road and rail vehicle axle supports, we can express the basic durability using the modified relationship in terms of kilometres driven.

$$L_{10km} = \left(\frac{C}{P}\right)^p \cdot \frac{\pi \cdot D}{1000}$$

L_{10km} . . . . basic durability   [10^6 km]

D . . . . wheel diameter   [m]

5.5.1 Standard values of basic durability

In cases, when the required durability for the given housing is not provided in advance, we can appropriately use the values provided in tables 5.4 and 5.5.

Table 5.1

C/P ratio depending on durability L_{10h}							
Ball bearings				Cylindrical roller, needle-roller, spherical-roller, and tapered-roller bearings			
L_{10} Durability $\times 10^6$ rev	C/P	L_{10} Durability $\times 10^6$ rev	C/P	L_{10} Durability $\times 10^6$ rev	C/P	L_{10} Durability $\times 10^6$ rev	C/P
0,5	0,79	600	8,43	0,5	0,81	600	6,81
0,75	0,91	650	8,66	0,75	0,92	650	6,98
1	1,00	700	8,88	1	1,00	700	7,14
1,5	1,14	750	9,09	1,5	1,13	750	7,29
2	1,26	800	9,28	2	1,24	800	7,43
3	1,44	850	9,47	3	1,39	850	7,56
4	1,59	900	9,65	4	1,52	900	7,70
5	1,71	950	9,83	5	1,62	950	7,82
6	1,82	1 000	10,00	6	1,71	1 000	7,94
8	2,00	1 100	10,30	8	1,87	1 100	8,17
10	2,15	1 200	10,60	10	2,00	1 200	8,39
12	2,29	1 300	10,90	12	2,11	1 300	8,59
14	2,41	1 400	11,20	14	2,21	1 400	8,79
16	2,52	1 500	11,40	16	2,30	1 500	8,97
18	2,62	1 600	11,70	18	2,38	1 600	9,15
20	2,71	1 700	11,90	20	2,46	1 700	9,31
25	2,92	1 800	12,20	25	2,63	1 800	9,48
30	3,11	1 900	12,40	30	2,77	1 900	9,63
35	3,27	2 000	12,60	35	2,91	2 000	9,78
40	3,42	2 200	13,00	40	3,02	2 200	10,10
45	3,56	2 400	13,40	45	3,13	2 400	10,30
50	3,68	2 600	13,80	50	3,23	2 600	10,60
60	3,91	2 800	14,10	60	3,42	2 800	10,80
70	4,12	3 000	14,40	70	3,58	3 000	11,00
80	4,31	3 500	15,20	80	3,72	3 500	11,50
90	4,48	4 000	15,90	90	3,86	4 000	12,00
100	4,64	4 500	16,50	100	3,98	4 500	12,50
120	4,93	5 000	17,10	120	4,20	5 000	12,90
140	5,19	5 500	17,70	140	4,40	5 500	13,20
160	5,43	6 000	18,20	160	4,58	6 000	13,60
180	5,65	7 000	19,10	180	4,75	7 000	14,20
200	5,85	8 000	20,00	200	4,90	8 000	14,80
250	6,30	9 000	20,80	250	5,24	9 000	15,40
300	6,69	10 000	21,50	300	5,54	10 000	15,80
350	7,05	12 500	23,20	350	5,80	12 500	16,90
400	7,37	15 000	24,70	400	6,03	15 000	17,90
450	7,66	17 500	26,00	450	6,25	17 500	18,70
500	7,94	20 000	27,10	500	6,45	20 000	19,50
550	8,19	25 000	29,20	550	6,64	25 000	20,90

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Table 5.2

L₁₀ₕ Durability	Rotation speed n [min⁻¹]												
Hod	10	16	25	40	63	100	125	160	200	250	320	400	500
100	-	-	-	-	-	-	-	-	1,06	1,15	1,24	1,34	1,45
500	-	-	-	1,06	1,24	1,45	1,56	1,68	1,82	1,96	2,12	2,29	2,47
1 000	-	-	1,15	1,34	1,56	1,82	1,96	2,12	2,29	2,47	2,67	2,88	3,11
1 250	-	1,06	1,24	1,45	1,68	1,96	2,12	2,29	2,47	2,67	2,88	3,11	3,36
1 600	-	1,15	1,34	1,56	1,82	2,12	2,29	2,47	2,67	2,88	3,11	3,36	3,63
2 000	1,06	1,24	1,45	1,68	1,96	2,29	2,47	2,67	2,88	3,11	3,36	3,63	3,91
2 500	1,15	1,34	1,56	1,82	2,12	2,47	2,67	2,88	3,11	3,36	3,63	3,91	4,23
3 200	1,24	1,45	1,68	1,96	2,29	2,67	2,88	3,11	3,36	3,63	3,91	4,23	4,56
4 000	1,34	1,56	1,82	2,12	2,47	2,88	3,11	3,36	3,63	3,91	4,23	4,56	4,93
5 000	1,45	1,68	1,96	2,29	2,67	3,11	3,36	3,63	3,91	4,23	4,56	4,93	5,32
6 300	1,56	1,82	2,12	2,47	2,88	3,36	3,63	3,91	4,23	4,56	4,93	5,32	5,75
8 000	1,68	1,96	2,29	2,67	3,11	3,63	3,91	4,23	4,56	4,93	5,32	5,75	6,20
10 000	1,82	2,12	2,47	2,88	3,36	3,91	4,23	4,56	4,93	5,32	5,75	6,20	6,70
12 500	1,96	2,29	2,67	3,11	3,36	4,23	4,56	4,93	5,32	5,75	6,20	6,70	7,23
16 000	2,12	2,47	2,88	3,36	3,91	4,56	4,93	5,23	5,75	6,20	6,70	7,23	7,81
20 000	2,29	2,67	3,11	3,63	4,23	4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43
25 000	2,47	2,88	3,36	3,91	4,56	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11
32 000	2,67	3,11	3,63	4,23	4,93	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83
40 000	2,88	3,36	3,91	4,56	5,32	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60
50 000	3,11	3,63	4,23	4,93	5,75	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50
63 000	3,36	3,91	4,56	5,32	6,20	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40
80 000	3,36	4,23	4,93	5,75	6,70	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40
100 000	3,91	4,56	5,32	6,20	7,23	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50
200 000	4,93	5,75	6,70	7,81	9,11	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20

C/P ratio dependent on L₁₀ₕ durability and rotation speed n for ball bearings

C/P ratio dependent on L_{10h} durability and rotation speed n for ball bearings															
Rotation speed n [min⁻¹]															
630	800	1 000	1 250	1 600	2 000	2 500	3 200	4 000	5 000	6 300	8 000	10 000	12 500	16 000	
1,56	1,68	1,82	1,96	2,12	2,29	2,47	2,67	2,88	3,11	3,36	3,63	3,91	4,23	4,56
2,67	2,88	3,11	3,36	3,63	3,91	4,23	4,56	4,93	5,32	5,75	6,20	6,70	7,23	7,81
3,36	3,63	3,91	4,23	4,56	4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83
3,63	3,91	4,23	4,56	4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60
3,91	4,23	4,56	4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50
4,23	4,56	4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40
4,56	4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40
4,93	5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50
5,32	5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60
5,75	6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80
6,20	6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20
6,70	7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60
7,23	7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20
7,81	8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90
8,43	9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70
9,11	9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70
9,83	10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80
10,60	11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80	31,10
11,50	12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80	31,10	-
12,40	13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80	31,10	-	-
13,40	14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80	31,10	-	-	-
14,50	15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80	31,10	-	-	-	-
15,60	16,80	18,20	19,60	21,20	22,90	24,70	26,70	28,80	31,10	-	-	-	-	-
19,60	21,20	22,90	24,70	26,70	28,80	31,10	-	-	-	-	-	-	-	-

Table 5.3

C/P ratio dependent on L_{10h} durability and rotation speed n for cylindrical roller, spherical-roller, and tapered-roller bearings													
L_{10h} Durability	Rotation speed n [min^{-1}]												
Hod	10	16	25	40	63	100	125	160	200	250	320	400	500
100	-	-	-	-	-	-	-	-	1,05	1,10	1,21	1,30	1,39
500	-	-	-	1,05	1,21	1,39	1,49	1,60	1,71	1,83	1,97	2,11	2,26
1 000	-	-	1,13	1,30	1,49	1,71	1,83	1,97	2,11	2,26	2,42	2,59	2,78
1 250	-	1,05	1,21	1,39	1,60	1,83	1,97	2,11	2,26	2,42	2,59	52,78	2,97
1 600	-	1,13	1,30	1,49	1,71	1,97	2,11	2,26	2,42	2,59	2,78	2,97	3,19
2 000	1,05	1,21	1,39	1,60	1,83	2,11	2,26	2,42	2,59	2,78	2,97	3,19	3,42
2 500	1,13	1,30	1,49	1,71	1,97	2,26	2,42	2,59	2,78	2,97	3,19	3,42	3,66
3 200	1,21	1,39	1,60	1,83	2,11	2,42	2,59	2,78	2,97	3,19	3,42	3,66	3,92
4 000	1,30	1,49	1,71	1,97	2,26	2,59	2,78	2,97	3,19	3,42	3,66	3,92	4,20
5 000	1,39	1,60	1,83	2,11	2,42	2,78	2,97	3,19	3,42	3,66	3,92	4,20	4,50
6 300	1,49	1,71	1,97	2,26	2,59	2,97	3,19	3,42	3,66	3,92	4,20	4,50	4,82
8 000	1,60	1,83	2,11	2,42	2,78	3,19	3,42	3,66	3,92	4,20	4,50	4,82	5,17
10 000	1,71	1,97	2,26	2,59	2,97	3,42	3,66	3,92	4,20	4,50	4,82	5,17	5,54
12 500	1,83	2,11	2,42	2,78	3,19	3,66	3,92	4,20	4,50	4,82	5,17	5,54	5,94
16 000	1,97	2,26	2,59	2,97	3,42	3,92	4,20	4,50	4,82	5,17	5,54	5,94	6,36
20 000	2,11	2,42	2,78	3,19	3,66	4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81
25 000	2,26	2,59	2,97	3,42	3,92	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30
32 000	2,42	2,78	3,19	3,66	4,20	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82
40 000	2,59	2,97	3,42	3,92	4,50	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38
50 000	2,78	3,19	3,66	4,20	4,82	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98
63 000	2,97	3,42	3,92	4,50	5,17	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62
80 000	3,19	3,66	4,20	4,82	5,54	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30
100 000	3,42	3,92	4,50	5,17	5,94	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00
200 000	4,20	4,82	5,54	6,36	7,30	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60

C/P ratio dependent on L_{10h} durability and rotation speed n for cylindrical roller, spherical-roller, and tapered-roller bearings														
Rotation speed n [min^{-1}]														
630	800	1 000	1 250	1 600	2 000	2 500	3 200	4 000	5 000	6 300	8 000	10 000	12 500	16 000
1,49	1,60	1,71	1,83	1,97	2,11	2,26	2,42	2,59	2,78	2,97	3,19	3,42	3,66	3,92
2,42	2,59	2,78	2,97	3,19	3,42	3,66	3,92	4,20	4,50	4,82	5,70	5,54	5,94	6,36
2,97	3,19	3,42	3,66	3,92	4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82
3,19	3,42	3,66	3,92	4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38
3,42	3,66	3,92	4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98
3,66	3,92	4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62
3,92	4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30
4,20	4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00
4,50	4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80
4,82	5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70
5,17	5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60
5,54	5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60
5,94	6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60
6,36	6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70
6,81	7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90
7,30	7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20
7,82	8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60
8,38	8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60	-
8,98	9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60	-	-
9,62	10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60	-	-	-
10,30	11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60	-	-	-	-
11,00	11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60	-	-	-	-	-
11,80	12,70	13,60	14,60	15,60	16,70	17,90	19,20	20,60	-	-	-	-	-	-
14,60	15,60	16,70	17,90	19,20	20,60	-	-	-	-	-	-	-	-	-

Table 5.4

Standard basic durability values in operating hours	
Type of machine	Basic durability L_{10h}
Seldom used machines and tools	1 000
Electrical household appliances, small fans	2 000 to 4 000
tools for intermittent use, hand tools, workshop cranes, agricultural machines	4 000 to 8 000
machines for intermittent use with high reliability demands, auxiliary machines for use in power plants, belt conveyors, transport trolleys, elevators	8 000 to 15 000
rolling mills	6 000 to 12 000
machines for 8-16 hour shifts, stationary motors, gears, spindles for textile machines, plastic processing machinery, printing machinery, cranes	15 000 to 30 000
machine tools, in general	20 000 to 30 000
machines for continuous operation: stationary electrical machines, transportation equipment, roller conveyors, pumps, centrifuges, blowers, compressors, hammer mills, shredders, briquetting presses, mine hoists, cable reels	40 000 to 60 000
machines for continuous operation with high operating safety requirements: power plant machinery, waterworks machines, paper mill machinery, ship machinery	100 000 to 200 000

Table 5.5

Standard values of basic durability in kilometres	
Type of vehicle	Basic durability L_{10km}
Road vehicle wheel bearings	
motorcycles	60 000
personal automobiles	150 000 to 250 000
lorries, buses	400 000 to 500 000
Axle bearings of rail vehicles	
freight cars (according to UIC) under constant maximal load per axle	800 000
trams	1 500 000
personal rail vehicles	3 000 000
motorized vehicles and motorized units	3 000 000 to 4 000 000
locomotives	3 000 000 to 5 000 000

5.6 Modified durability equation

The operating durability, as previously described, depends on many factors. Research and operating results demonstrated that greater durability can be achieved through thorough lubrication, when the roller elements are entirely separated by a layer of grease. It was further demonstrated that greater resistance against stress damage of materials is provided using advanced manufacturing processes. This technical advance was incorporated into standard ISO 281 as a modified durability calculation, which includes reliability a_1, material a_2, and operating condition a_3 factors. Additional test results concluded that the impact of materials on operating conditions, in particular, lubrication, are in close correlation. This led to the merger of both factors into one a_{23}.

The modified durability is thus the modified basic durability which, aside from taking into account load, also considers the impact of bearing material components, the physical and chemical properties of the lubricant, and the temperature regime of the bearing operating environment.

$$L_{na} = a_1 + a_{23} + L_{10}$$

L_{na} .... modified durability for reliability (100 - n) %
and other than normal operating conditions .... $[10^6 \text{ rev}]$

a_1 ..... eliability coefficient for other than 90 % reliability, see table 5.6

a_{23} .... material, lubricant, manufacturing technology, and operating condition coefficient, see fig. 5.3

L_{10} .... basic durability .... $[10^6 \text{ rev}]$

Table 5.6

Coefficient a_1 values		
Reliability (%)	L_n	a_1
90	L_{10}	1,000
95	L_5	0,640
96	L_4	0,550
97	L_3	0,470
98	L_2	0,370
99	L_1	0,250
99,2	$L_{0,8}$	0,220
99,4	$L_{0,6}$	0,190
99,6	$L_{0,4}$	0,160
99,8	$L_{0,2}$	0,120
99,9	$L_{0,1}$	0,093
99,92	$L_{0,08}$	0,087
99,94	$L_{0,06}$	0,080
99,95	$L_{0,05}$	0,077
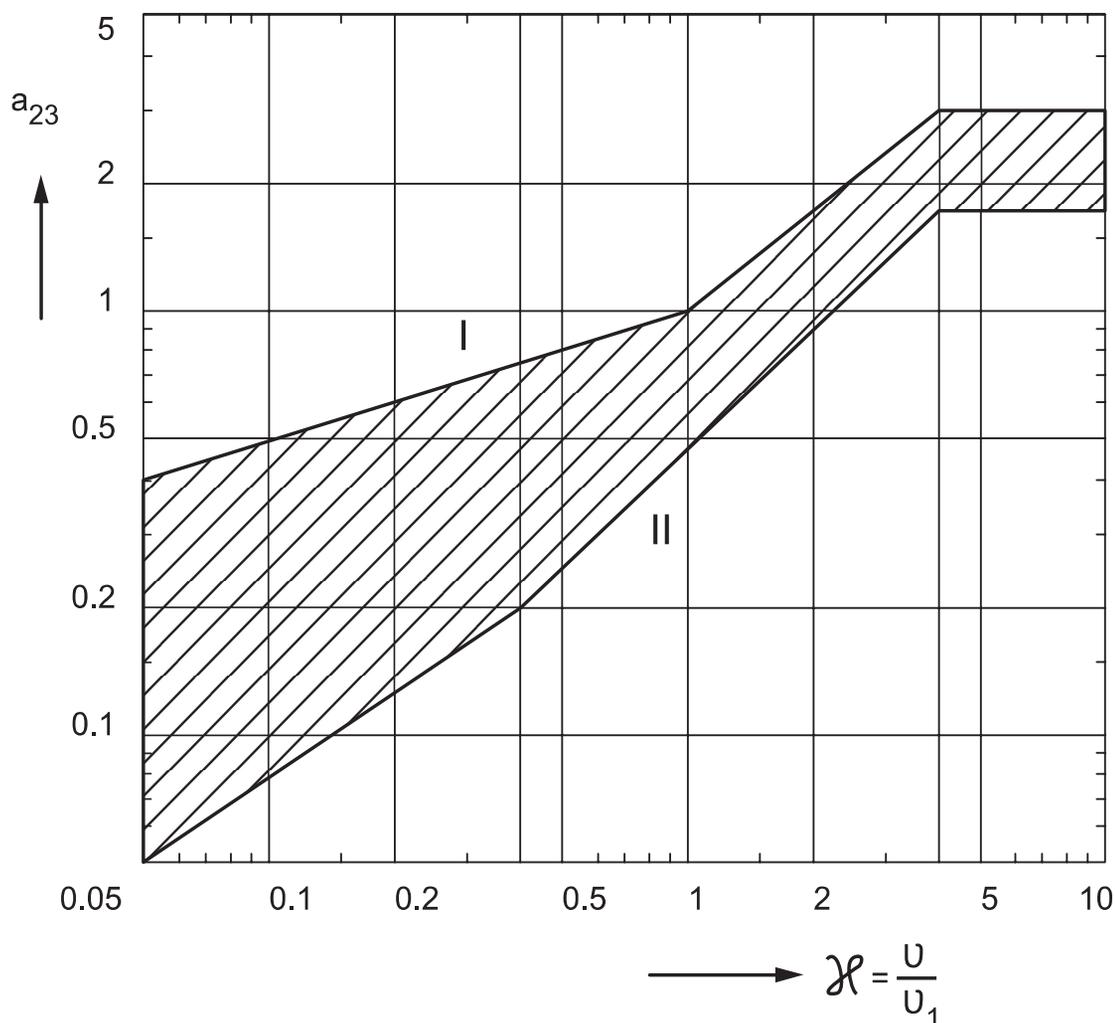
Fig. 5.3

The diagram in fig. 5.4 is used to determine the basic values of coefficient a_{23}.

The quality of the lubrication process is given by the extent of separation of the roller surfaces. Viscosity is a decisive factor for the formation of lubricant film, which is strongly related to temperature. The viscosity ratio, as follows, decides on the use of lubricant:

$$\kappa = \frac{\nu}{\nu_1}$$

ν . . . . . .lubricant kinematic viscosity at bearing operating temperature $[m^2 . s^{-1}]$

ν_1 . . . . .kinematic viscosity for the defined revolution speed
and the given dimension of the bearing $[m^2 . s^{-1}]$

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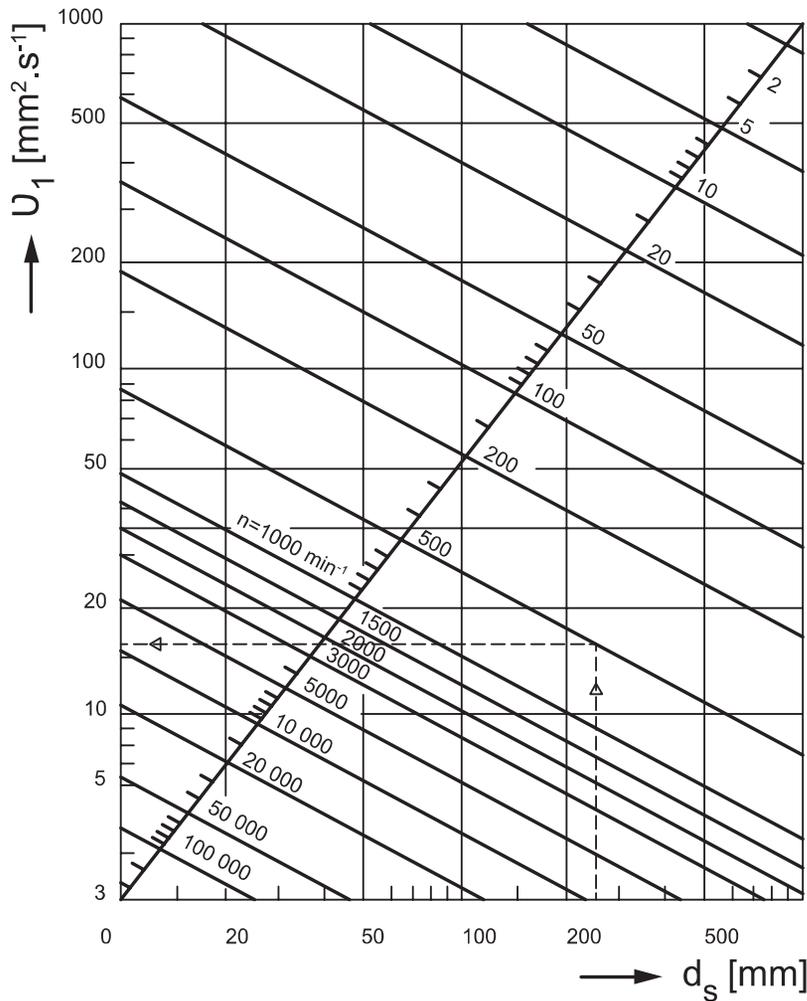
Fig. 5.4

We determine the ν and ν_1 values based on the diagram found in fig. 5.4 and 5.5. In the diagram on fig. 5.3, line I applies for radial ball bearings that operate in a very clean environment. In all other cases, we select a lower a_{23} coefficient, proportional to the cleanliness of the environment, while a decreasing tendency is dependent on the structural group of the bearing in the following order:

- Angular-contact ball bearings
- Tapered-roller bearings
- Cylindrical roller bearings
- Double-row self-aligning bearings
- Spherical-roller bearings

Line II can be used to determine coefficient a_{23} for spherical-roller bearings that operate in a dusty environment.

We recommend that these issues be resolved in consultation with the ZKL technical and consultation services department.
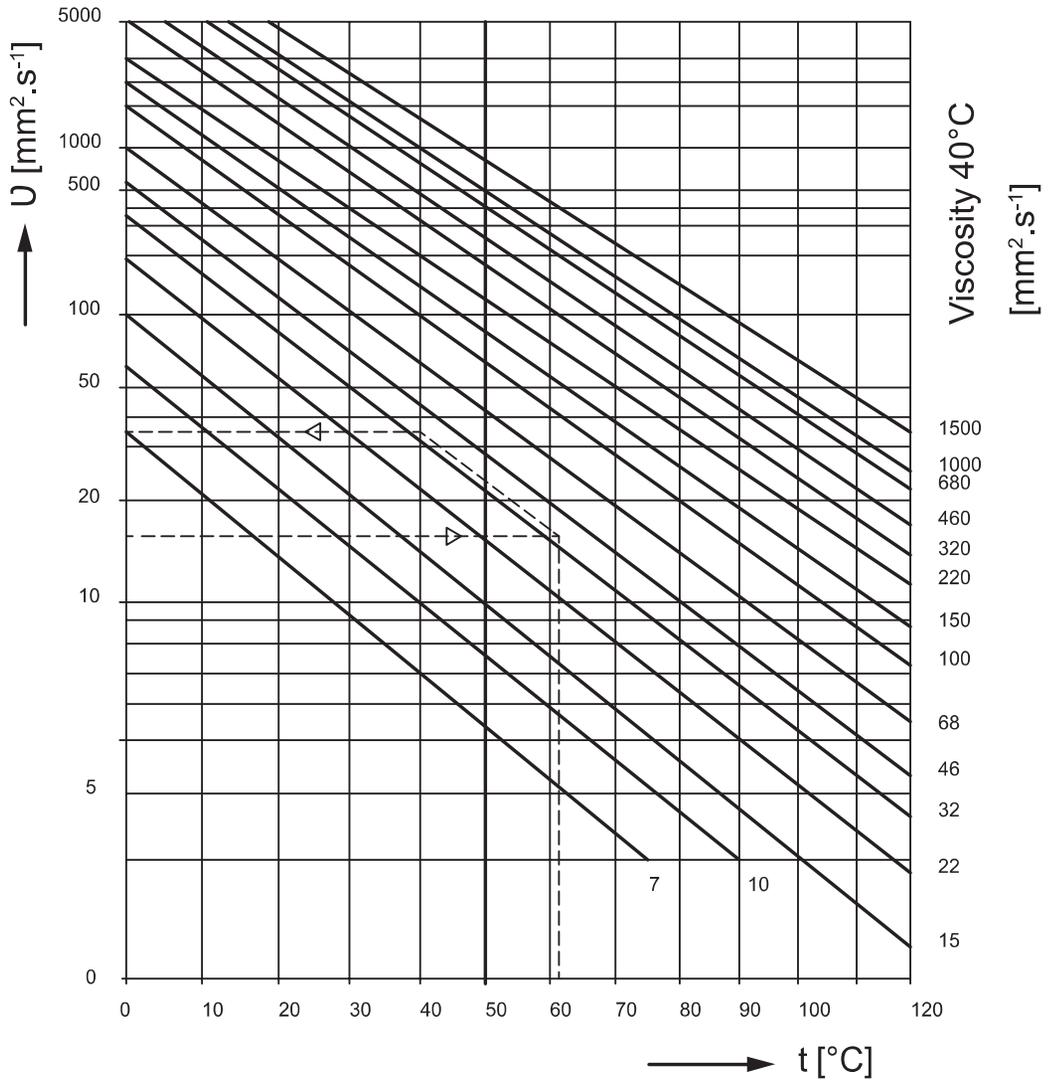
Fig. 5.5

5.7   Durability according ZKL

The use of the L_{10} calculation of basic durability as bearing performance parameter criteria has demonstrated, over many years, to be satisfactory. This calculation is associated with 90 % reliability in conjunction with the use of superior materials, a superior technological design, and under normal operating conditions.

Notwithstanding, many applications require that the calculation be performed for a different reliability level or for more precise lubrication and contamination conditions. It was determined, with the use of advanced high quality bearing steel, that under favourable operating conditions and when contact stresses fall below the limit values and provided that the bearing steel fatigue stress limit is not exceeded, a higher durability than L_{10} can be achieved. Under unfavourable operating conditions, on the other hand, the bearing durability can in fact be shorter than L_{10}.

A system approach of fatigue-related durability was applied when creating the method of calculating ZKL modified durability. The impact on the durability of the system (bearing) is described in the following text and considers the influence of variance and the interaction of mutually related factors on the overall life. These factors are demonstrated through increased contact stress in the contact area, which leads to decreased service life.

These factors are used in the modified durability equation.

$$L_m = a_1 \cdot a_{ZKL} \cdot L_{10}$$

a_1. . . . . . .reliability coefficient for other than 90 % reliability, see table 5.6

a_{ZKL} . . . . . .modified life coefficient

L_{10} . . . . . .basic durability   [10^6 rev]

Provided that the lubrication conditions, cleanliness of the environment, and other operation conditions are favourable, an advanced, high-quality bearing can, under a certain load, achieve infinite service life. The fatigue load limit for bearings manufactured from generally high-quality bearing material and workmanship is such a load, that the contact pressure exerted on roller elements in the bearing is approximately 1500 MPa. This stress value takes into account the additional stresses caused by manufacturing tolerances and operating conditions. Decreased product precision and quality of materials leads to a lower fatigue load limit.

The contact stress in many applications is greater than 1500 MPa. Such operating conditions lead to reduced bearing life.

The operating influences can be related to the applied stress and rigidity of the material.

- Notches lead to the formation of edge stresses.
- A thin film of oil increases the stress at the contact area between the raceway and the roller element.
- Increased temperature decreases the fatigue load limit (its strength) of the material.
- A static inner ring (increased overlap) leads to increased orbital stress

Various influences on bearing durability are mutually dependant. Consequently, a systemic approach to calculating fatigue durability is entirely appropriate.
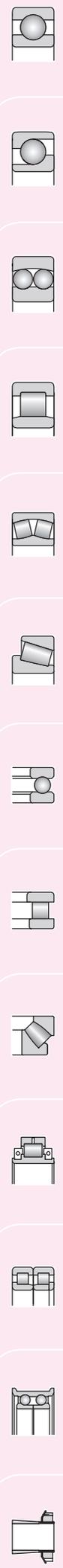
A theoretical explanation of how to incorporate additional influences, such as the radial clearance during operation and the variable stress on raceways from tilting, is explained in ISO/TS 16281.

5.7.1 Fatigue load limit

The modified durability coefficient a_{ZKL} can be expressed as function

$$\frac{\sigma_u}{\sigma}$$

(fatigue load limit divided by the real stress σ, while considering all potential influencing factors).

If the actual stress decreases to fatigue stress limit, then a_{ZKL} asymptotically approaches infinity. Generally, the orthogonal shear stress is used as a fatigue criterion. The diagram on fig. 5.6 is also based on the shear fatigue limit.
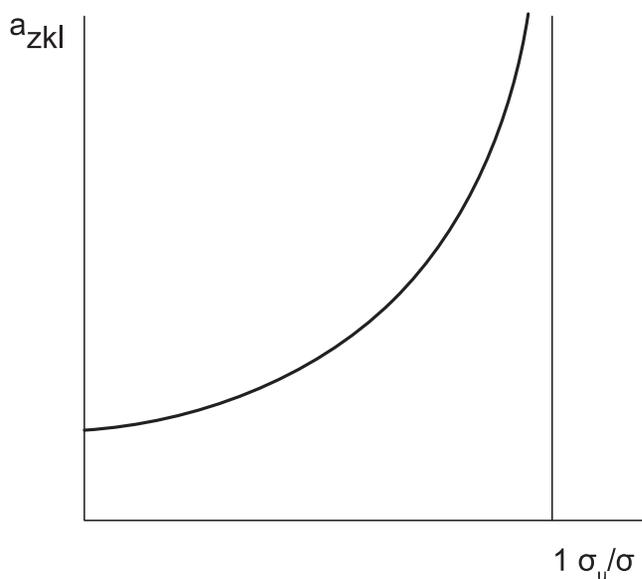
Fig. 5.6

Analogous to the C_{or} static load rating, defined in ISO 76, the fatigue load limit is defined as the load, during which the stress fatigue limit is reached at the most burdened point on the orbit.

The ratio $\dfrac{\sigma_u}{\sigma}$ can then be estimated according to the ratio $\dfrac{C_{or}}{P}$

and the modified life coefficient can be expressed as:

$$a_{ZKL} = f\left(\frac{C_{or}}{P}\right)$$

The following must be considered when calculating the C_{or} static load rating:

- The type, size, and internal geometry of the bearing
- The profile of rolling elements and the raceways
- The quality of technological processes
- The fatigue limit for the raceway materials

5.7.2 Determining the modified durability coefficient

The modified durability coefficient takes into consideration the following:

- The fatigue load and bearing load
- Lubrication (type of lubricant, viscosity, revolution speed, bearing size, additives)
- Environment (degree of contamination, packing)
- Contaminating particles (strength and size of particles in relation to bearing size, lubrication and filtration method)
- Installation (cleanliness during installation)

The effect of bearing clearance and the effect of tilt on bearing durability is described in ISO/TS 16281.

The a_{ZKL} Fatigue life coefficient is derived from the following equation:

$$a_{ZKL} = f\left[\frac{e_C \cdot C_{or}}{P}, \kappa\right]$$

Factors e_C and κ adjust for contamination and lubrication conditions.

5.7.3 Contamination factor

If the grease is contaminated with sold particles, notches may form in the orbit due to rolling. Stress points (concentrations) form later on these notches, which results in decreased bearing life. The given decrease in life caused by the contamination of lubricant is adjusted for in the e_C contamination factor.

Decreased bearing life caused by the effect of sold particles in the lubricant film depends on:

- The type, size, strength, and amount of particles
- The lubricating film thickness (relative viscosity)
- Bearing size

Approximate contamination factor values can be taken from table 5.7.

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Table 5.7

Contamination level	e_c	
	$D_{pw} < 100\,mm$	$D_{pw} \geq 100\,mm$
Extremely clean Particle size in the order of lubricating film thickness, Laboratory conditions	1	1
Highly clean Oil filtered through a very fine filter, typical conditions for a bearing with plastic housing and lifetime lubricant filling	0,8 to 0,6	0,9 to 0,8
Normally clean Oil filtered through a fine filter, typical conditions for a bearing with metal-sheet housing and lifetime lubricant filling	0,6 to 0,5	0,8 to 0,6
Mild contamination Minor contamination in lubricant	0,5 to 0,3	0,6 to 0,4
Typical contamination Typical bearing conditions without integrated bearing glands, particles causing wear enter bearing from vicinity	0,3 to 0,1	0,4 to 0,2
Strong contamination The bearing environment is strongly contaminated, bearing housing with insufficient bearing glands	0,1 to 0	0,1 to 0
Very strong contamination	0	0

Detailed calculation of the contamination factor

Table 5.7 lists the approximate contamination factor values. If the situation requires the use of more detailed calculations, the more precise calculation, provided below, must be used.

A contamination factor may be established for the following types of lubricants:

- Circulating oil lubrication with on-line filtration

- Oil bath lubrication or circulating lubrication with off-line filtration

- Grease

Definition of the β_x filtration ratio:

$$\beta_x = \frac{n_1}{n_2}$$

β_x . . . . . . filtration ratio for particles of determined size x

n_1 . . . . . . number of particles per unit of volume (100 ml) larger than x, prior to passage through filter

n_2 . . . . . . number of particles per unit of volume (100 ml) larger than x, after passage through filter

The filter ratio determined the filter efficiency.

Circulating lubrication with on-line filtration

The β_x filter ratio with particles of size x in µm according to standard ISO 16889 is the most influential factor when choosing the corresponding diagram.
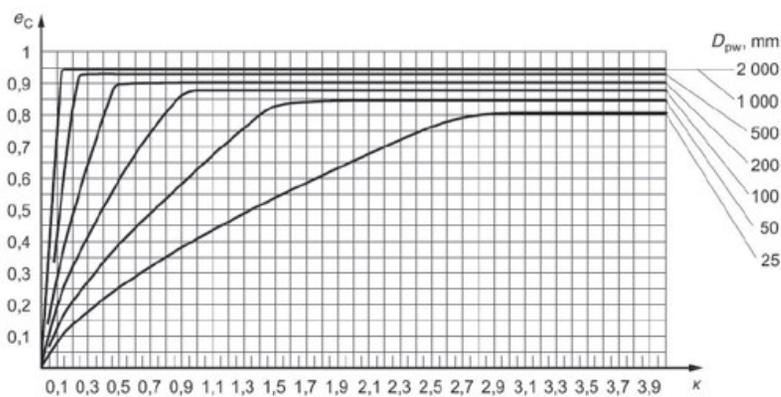
Fig. 5.7   Fouling factor for a circulating oil lubrication system with on-line filtration $\beta_6 = 200$
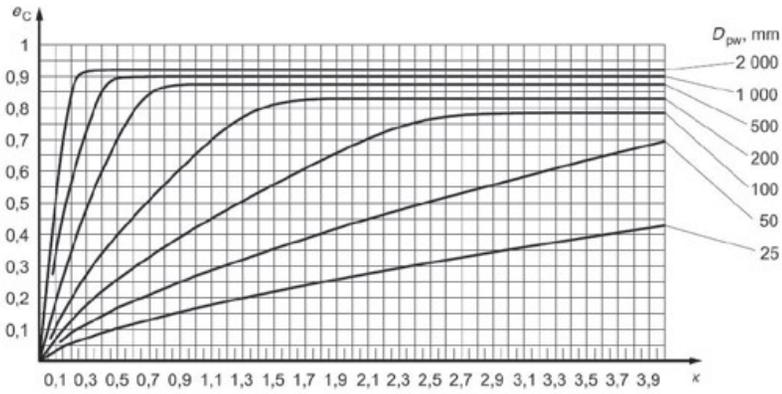
Fig. 5.8   Fouling factor for a circulating oil lubrication system with on-line filtration $\beta_{12} = 200$
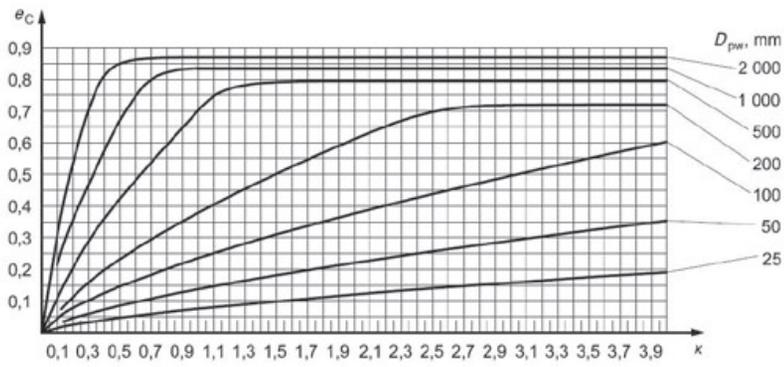
Fig. 5.9   Fouling factor for a circulating oil lubrication system with on-line filtration $\beta_{25} = 75$
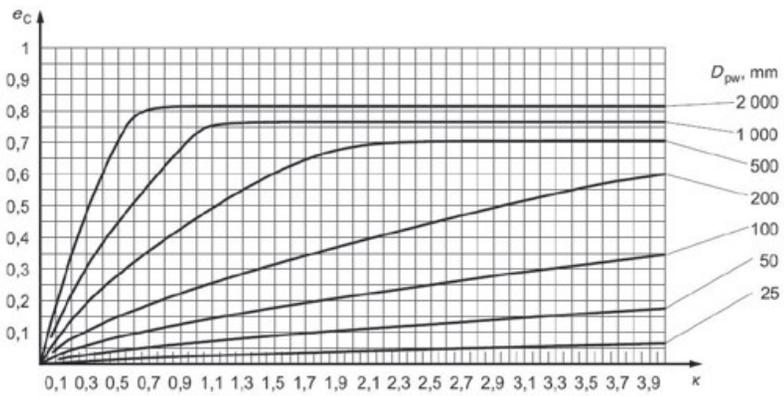
Fig. 5.10   Fouling factor for a circulating oil lubrication system with on-line filtration $\beta_{40} = 75$

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Oil bath lubrication or circulating lubrication with off-line filtration
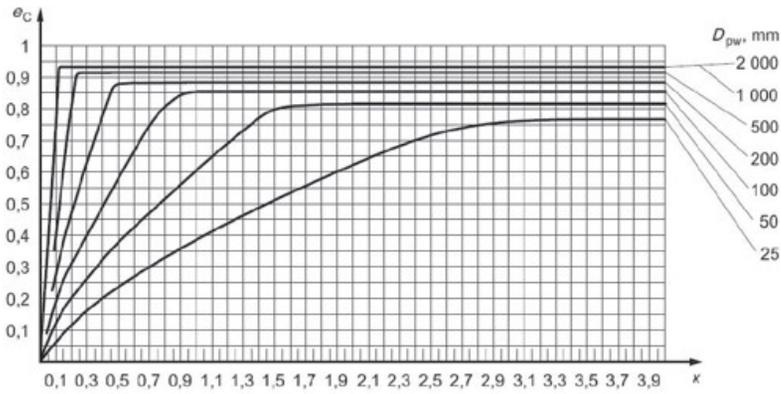
Fig. 5.11   *Fouling factor for oil bath lubrication or for oil lubrication with offline filtration*
             ISO 4406 – degree of contamination by solid particles -13/10
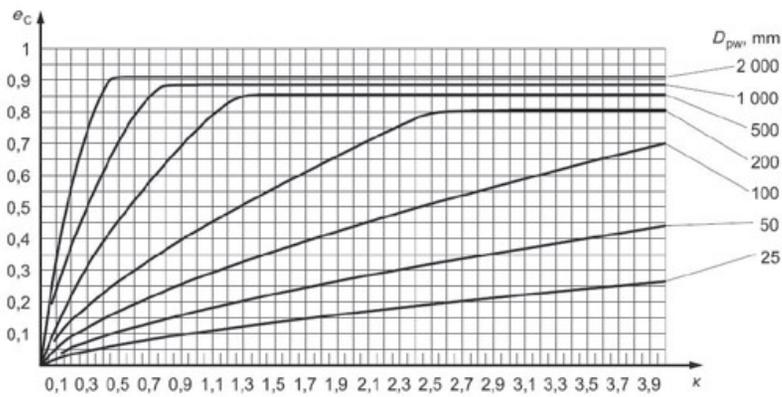
Fig. 5.12   *Fouling factor for oil bath lubrication or for oil lubrication with offline filtration*
             ISO 4406 – degree of contamination by solid particles -15/12
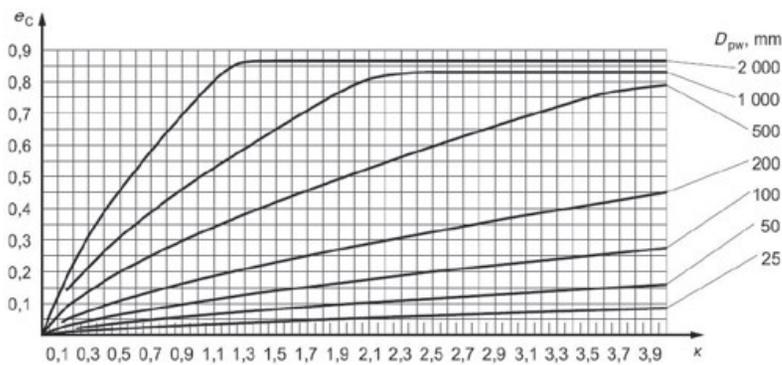
Fig. 5.13   *Fouling factor for oil bath lubrication or for oil lubrication with offline filtration*
             ISO 4406 – degree of contamination by solid particles -17/14

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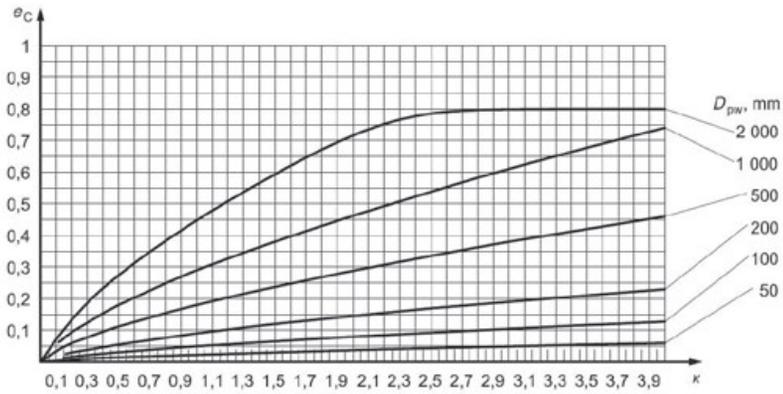
Fig. 5.14   Fouling factor for oil bath lubrication or for oil lubrication with offline filtration
ISO 4406 – degree of contamination by solid particles -19/16
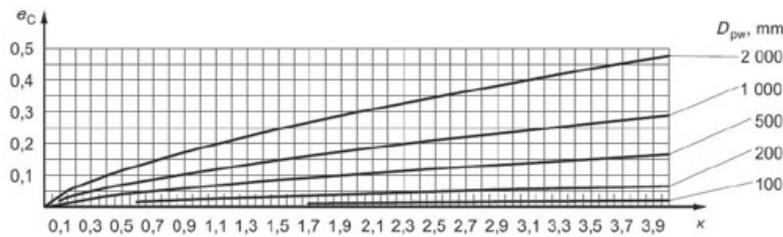
Fig. 5.15   Fouling factor for oil bath lubrication or for oil lubrication with offline filtration
ISO 4406 – degree of contamination by solid particles -21/18

Grease

Table 5.8

Operating conditions	Contamination level
Very clean installation, very good packing relative to operating conditions, continuous lubrication or lubrication in short intervals (Bearings with integrated bearing glands)	Highly clean
Clean installation, good packing, additional lubrication per manufacturer specifications (Bearings with integrated bearing glands)	Normally clean
Clean installation, average sealing capacity relative to operating conditions	Mild contamination
On-site-installation, bearing and housing insufficiently washed following installation, poor sealing capacity relative to operating conditions, re-lubrication intervals longer than recommended	Strong contamination
Installation in a contaminated environment, insufficient gland packaging, long re-lubrication intervals	Very strong contamination
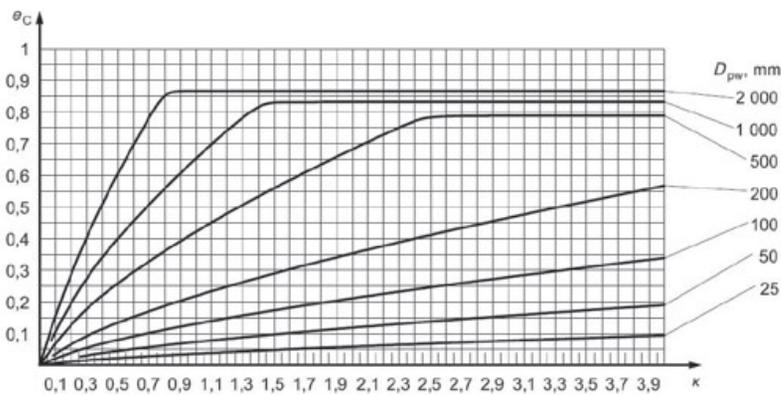
Fig. 5.16   Fouling factor for grease lubrication – moderate pollution
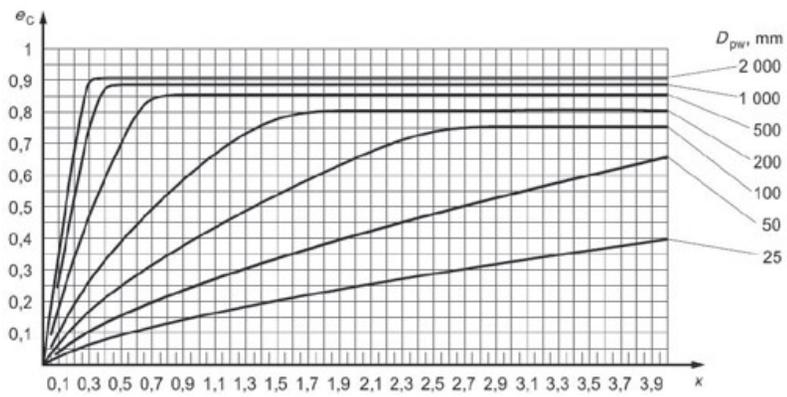
Fig. 5.17   Fouling factor for grease lubrication– usual purity
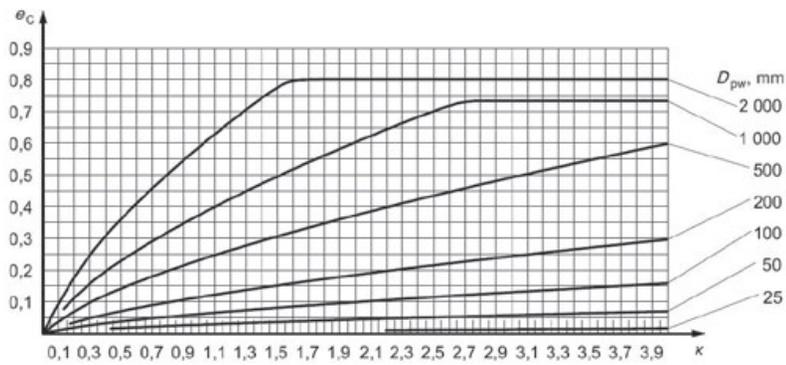
Fig. 5.18   Fouling factor for grease lubrication - strong contamination
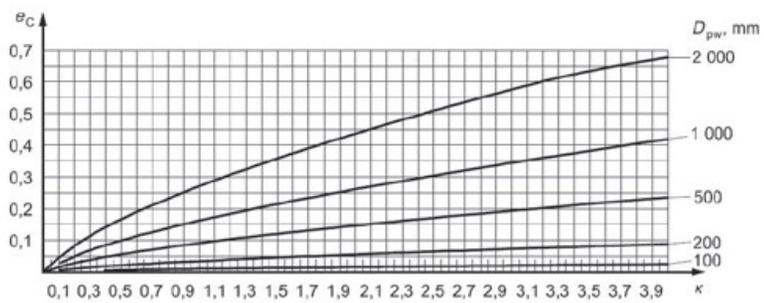
Fig. 5.19   Fouling factor for grease lubrication – very strong contamination
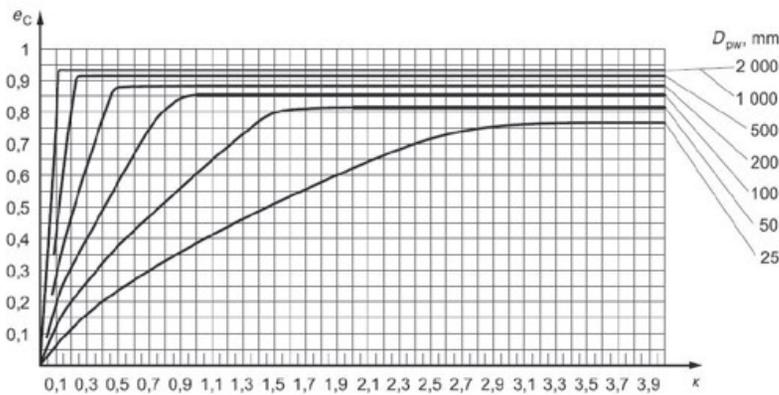
Fig. 5.20   Fouling factor for grease lubrication – high purity

5.7.4 Viscosity ratio

The effectiveness of the lubricant is primarily given by the degree of separation of contact elements. The formation of adequate lubricating film is subject to the given minimal viscosity that the lubricant must possess, when the application achieves its operating temperature. A requirement for the formation of lubricating film is specified by the viscosity ratio κ, which is defined as the ratio between the real (actual) kinematic viscosity ν and the reference kinematic viscosity ν_1. The kinematic viscosity ν is the viscosity of the lubricant, when the given lubricant achieves its operating temperature.

$$\kappa = \frac{\nu}{\nu_1}$$

In order to create sufficient lubricating film, the lubricant must maintain a certain minimal viscosity at operating temperature. The bearing life may be increased by increasing the operating viscosity ν.

The reference kinematic viscosity can be determined from figure 5.4 or by using the following equations:

$$\nu_1 = 45\ 000 \cdot n^{-0.8} \cdot D_{pw}^{-0.5} \qquad \text{for } n < 1\ 000 \text{ rev/min}$$

$$\nu_1 = 45\ 000 \cdot n^{-0.5} \cdot D_{pw}^{-0.5} \qquad \text{for } n \geqq 1\ 000 \text{ rev/min}$$

$D_{pw} = 0.5 \cdot (d + D)$ is the bearing mean diameter

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5.7.5 Calculating the modified durability coefficient

The modified durability coefficient a_{ZKL} may be easily determined from the following graphs:
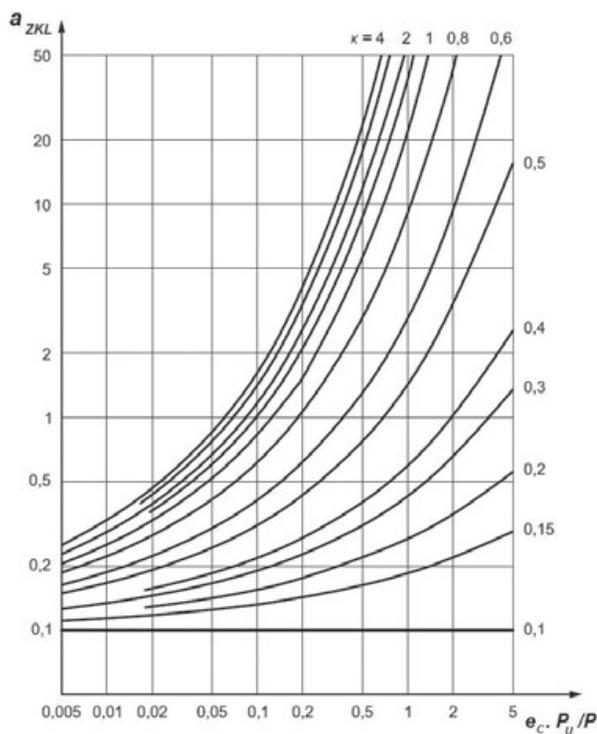
Fig. 5.21   Coefficient of life modification factor for thrust ball bearings
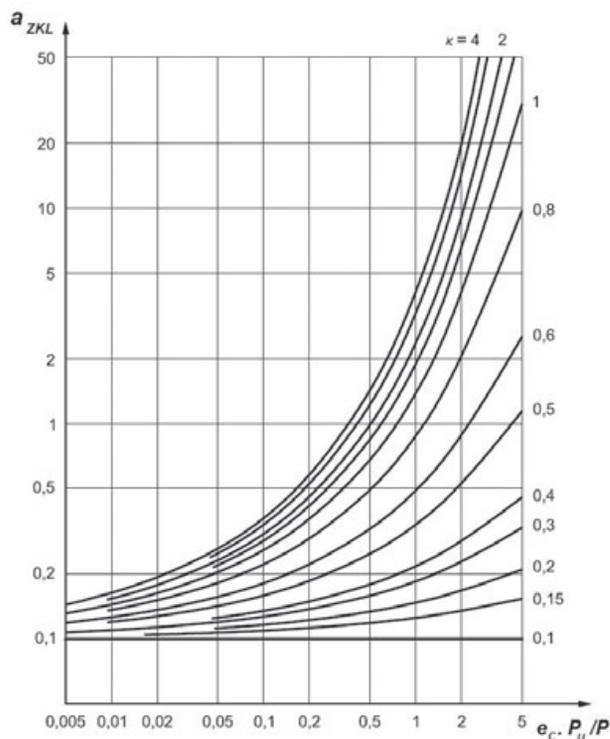

Fig. 5.22   Coefficient of life modification factor for thrust rolling bearings
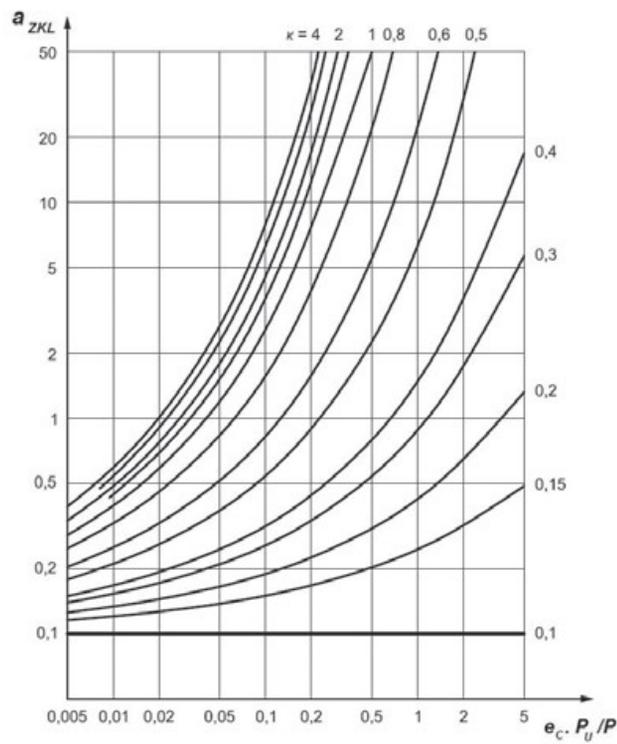
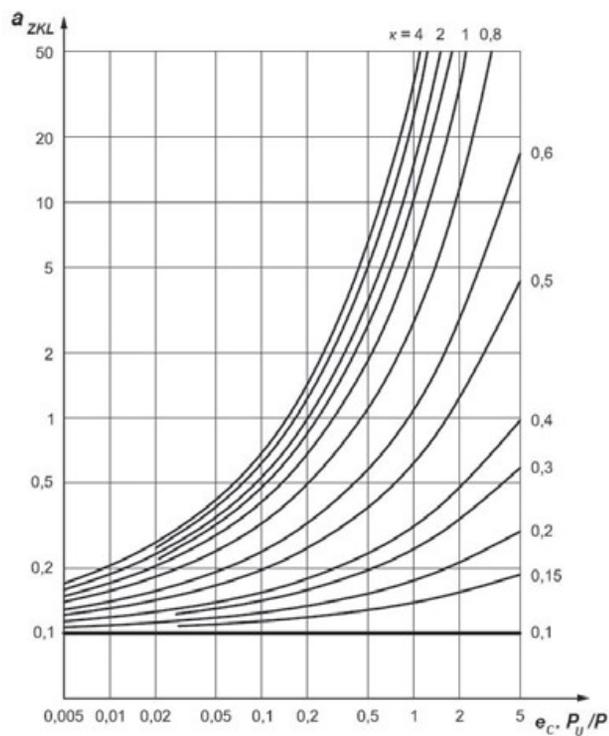Fig. 5.23   Coefficient of life modification factor for radial ball bearings

Fig. 5.24   Coefficient of life modification factor for radial rolling bearing

5.8 Equivalent dynamic load

The bearing in the structural node is exposed generally to acting forces of various magnitudes at various revolution speeds and with various periods of action. In terms of the calculation method, the applied forces must be recalculated at constant load, during which the bearing has the same durability as achieved under actual load. This recalculated constant radial or axial load is called equivalent load P, or P_r (radial) or P_a (axial), resp.

5.8.1 Combined loads

Constant load method

The external forces applied on the bearing do not change in size or in relation to time.

Radial bearings

If constant radial or axial forces simultaneously act on a radial bearing, the following equation for calculating the radial dynamic load applies:

$$P_r = X \cdot F_r + Y \cdot F_a \qquad \text{[kN]}$$

P_r . . . . . . radial equivalent dynamic load	[kN]
F_r . . . . . . radial force acting on the bearing	[kN]
F_a . . . . . . axial force acting on the bearing	[kN]
X . . . . . . radial load coefficient	
Y . . . . . . axial load coefficient	

Coefficients X and Y are dependent on the ratio F_a / F_r. The values X and Y are provided in the table or in the commentary preceding each structural group, where further information is provided for bearing calculations of the respective structural group.

Thrust bearings

Thrust ball bearings can only transfer forces acting axially and the following equation applied for calculating the axial equivalent dynamic load:

$$P_a = F_a \qquad \text{[kN]}$$

P_a . . . . . . axial equivalent dynamic load	[kN]
F_a . . . . . . axial bearing load	[kN]

Spherical-roller thrust bearings can also transfer certain radial loads, however, only when a simultaneous axial load is applied, while observing the following condition:

$$P_a = F_a + 1.2 \cdot F_r \qquad [kN]$$

Variable loading method

A real variable load, whose time course is known, is replaced by a mean intended load to enable calculation. This intended load has the same effect on the bearing as an actual variable load.

5.8.2 Change in load magnitude at constant revolution speed

If a load acts on a bearing in a constant direction, whose size changes in relation to time, while the revolution speed is constant (fig. 5.25), we calculate the mean intended load F_s according to the equation

$$F_s = \sum_{i=1}^{n} F^3 \cdot \left[\frac{q_i}{100} \right]^{\frac{1}{3}} \qquad [kN]$$

F_s . . . . intended mean constant load   [kN]

$F_s = F_1, ..., F_n$ . . constant partial actual load   [kN]
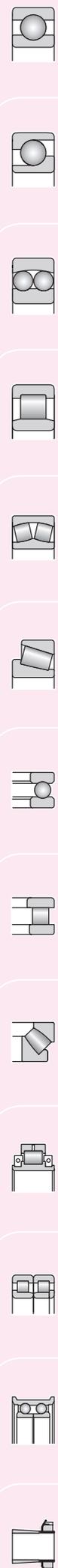
$q_i = q_1, ..., q_n$ . . . proportion of partially acting loads   [%]

If a variable load acts on a bearing, while the rotation speed meanwhile changes (fig. 5.26), we calculate the mean intended load using the equation

$$F_s = \frac{F_{min} + 2 \cdot F_{max}}{3} \qquad [kN]$$

Provided that the actual load has a sinusoid shape (fig. 5.27), the mean intended load is given by
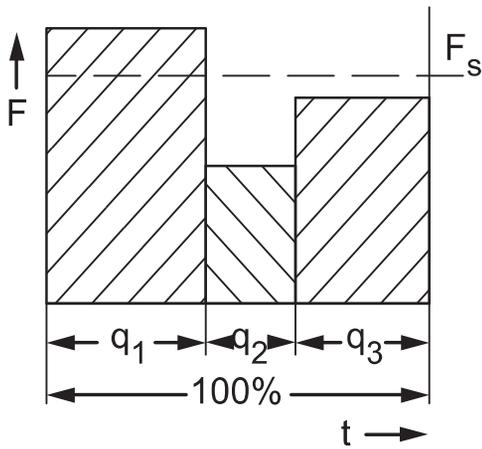
$$F_s = 0.75 \cdot F_{max} \qquad [kN]$$
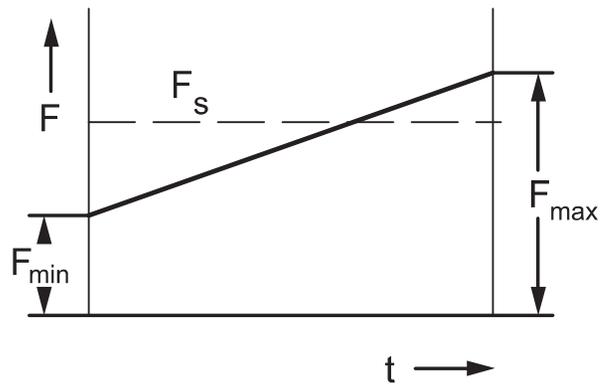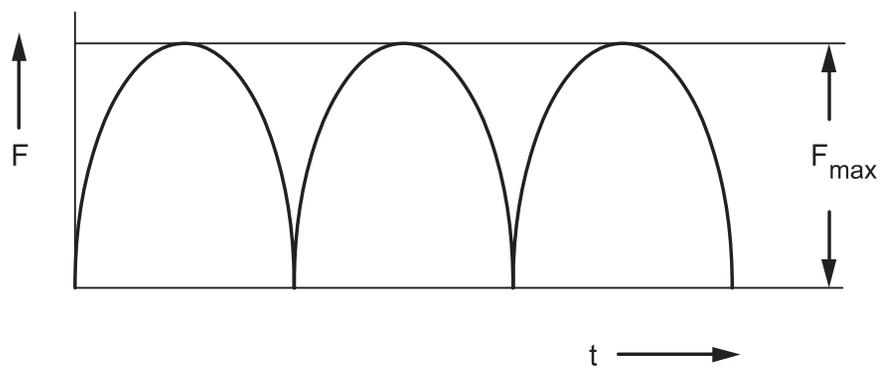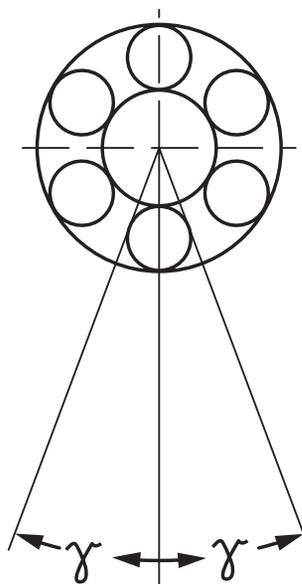
Fig. 5.25

Fig. 5.26

Fig. 5.27

Fig. 5.28

5.8.3 Change in load magnitude when rotation speed changes

If a variable load acts on a bearing along with variable rotation speed, the intended mean load is derived from the equation

$$F_s = \left(\frac{\sum\limits_{i=1}^{n} F^3 \cdot q_i \cdot n_i}{\sum\limits_{i=1}^{n} q_i \cdot n_i} \right)^{\frac{1}{3}} \qquad [kN]$$

$n_i = n_1, ..., n_n$ . . .constant rotation speed during action of partial loads $F_1, ..., F_n$ [min⁻¹]

$q_i = q_1, ..., q_n$ . . .proportion of partially acting loads and frequencies   [%]

If the rotation speed only changes in relation to time, the intended mean rotation speed is calculated using the equation

$$n_s = \left(\frac{\sum\limits_{i=1}^{n} q_i \cdot n_i}{100} \right) \qquad [min^{-1}]$$

n_s. . . . . . .mean rotation speed   [min⁻¹]

5.8.4 Oscillating motion of the bearing

During oscillating motion with oscillating amplitude γ (fig. 5.28), it is easiest to substitute the oscillating motion by the notion of rotation, provided that the frequency of rotation is equal to the oscillating frequency. For radial bearings, we calculate the mean intended load using the equation

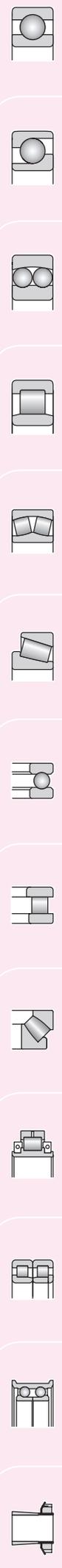$$F_s = F_r \cdot \left[\frac{\gamma}{90} \right]^{\frac{1}{p}} \qquad [kN]$$

F_s. . . . . . .mean intended load   [kN]

F_r. . . . . . .actual radial load   [kN]

γ . . . . . . .amplitude of oscillation   [°]

p . . . . . . .ball bearing exponent $p = 3$

For roller, needle roller, spherical-roller, and tapered-roller bearings $p = \dfrac{10}{3}$

5.9 Effect of temperature

The supplied range of bearings is designated for use in environments with a temperature of up to 120 ºC. Larger spherical roller bearings are manufactured, by default, for operation in temperatures up to 200 ºC. The exception are particular double row spherical roller bearings with polyamide races and single row ball bearings equipped with seals (RS, 2RS, RSR, 2RSR), which may be used short-term in temperatures up to 150 ºC. More information about these bearings is available in chapter 12 "Manufacturer data".

Rolling bearings designed for higher operating temperatures are manufactured to ensure their required physical and mechanical properties and dimensional stability. Housing solutions at higher operating temperatures should be consulted with the supplier.

The and dynamic load rating values C_r and C_a provided within the tables of the publication must, in the case of higher operating temperatures, be multiplied by the coefficient f_t, as specified in table 5.9.

Table 5.9

f_t Coefficient values				
operating temperature up to [°C]	150	200	250	300
f_t coefficient	0,95	0,9	0,75	0,6

5.10 Static Load Rating

The radial static load rating C_{or} and axial static load rating C_{oa} for each bearing is specified in the table section of the publication. The values C_{or} and C_{oa} were determined by calculation according to international standard ISO 76.

The static load rating is the load that corresponds to the calculated contact stress in the roller element and raceway contact zone, under the greatest load.

- 4600 MPa for double row self-aligning ball bearings
- 4200 MPa for other ball bearings
- 4000 MPa For roller, needle roller, spherical roller, and tapered roller bearings

This stress permanently deforms the rolling elements and raceways by approximately 0.0001 the diameter of the rolling element. The load is purely radial for radial bearings and purely axial within the bearing axis for thrust bearings.

The static load rating C_{or} is used for calculations, if the bearings

- rotate at very low speeds (n < 10 min^{-1})
- perform very slow oscillating motions
- under load do not move for a particular, extended period.

It is equally very important to check the safety in short-acting loads, such as e.g. shock loads and peak loads that act on a rotating bearing (dynamic load) or on a stationary bearing.

The maximum load that can act on a bearing should be used when calculating the equivalent static load of a bearing.

5.10.1   Equivalent static load

The equivalent static load is the recalculated radial load P_{or} for radial bearings and the axial load P_{oa} for thrust bearings.

$$P_{or} = X_0 \cdot F_r + Y_0 \cdot F_a \qquad\qquad [kN]$$

$$P_{oa} = Y_0 \cdot F_a \qquad\qquad [kN]$$

P_{or} . . . . . . radial equivalent static load $\qquad\qquad$ [kN]

P_{oa} . . . . . . axial equivalent static load $\qquad\qquad$ [kN]

F_r . . . . radial load $\qquad\qquad$ [kN]

F_a . . . . axial load $\qquad\qquad$ [kN]

X_0 . . . . . . radial load coefficient

Y_0 . . . . . . axial load coefficient

Table 5.10

s_0 Coefficient			
Bearing motion	**Load bearing method, bearing operation requirements**	**s_0**	
		Ball bearings	**Cylindrical roller, needle-roller, spherical-roller, and tapered-roller bearings**
rotational	significant impact loads, high demands on quite operation	2	4
	after static loading, bearing turns at lower loads	1,5	3
	normal demands for quiet operation		
	normal operating conditions and normal operating requirements	1	1,5
	quiet operation without vibration(s)	0,5	1
Oscillating	small oscillating angle with large frequency with occasional uneven loads	2	3,5
	large oscillating angle with small frequency with relatively constant periodical loads	1,5	2,5
non-rotating (at rest)	considerable impact loads	1,5 to 1	3 to 2
	normal and low loads, bearing operation unburdened by increased demands	1 to 0,4	2 to 0,8
	spherical-roller thrust bearings during all types of motion and loading	-	4

Coefficients X_0 and Y_0 are specified in the table section of the publication. Detailed information is also provided here for determining the equivalent static load of bearings of a particular structural group.

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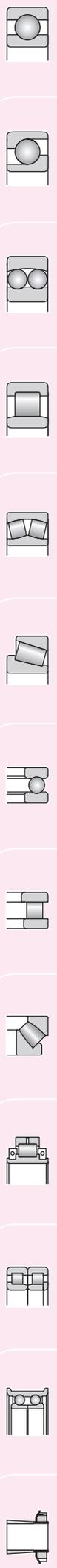
5.10.2   Bearing safety during static loading

In practice, the bearing safety under static load is determined from the ratio C_{or}/P_{or} or C_{oa}/P_{oa} and compared with the data in table 5.10, where the smallest permissible coefficient values S_0 are specified for various operating conditions.

$$S_0 = \frac{C_{or}}{P_{or}} \qquad \text{and/or} \qquad \frac{C_{oa}}{P_{oa}}$$

S_0 . . . . . . safety coefficient under static load   [kN]

C_{or} . . . . . . radial dynamic load capacity   [kN]

C_{oa} . . . . . . axial dynamic load capacity   [kN]

P_{or} . . . . . . radial equivalent static load or max. acting force F_{rmax} (fig. 5.29)
            under significant impact load, resp.   [kN]

P_{oa} . . . . . . axial equivalent static load or max. acting force F_{rmax} (fig. 5.29)
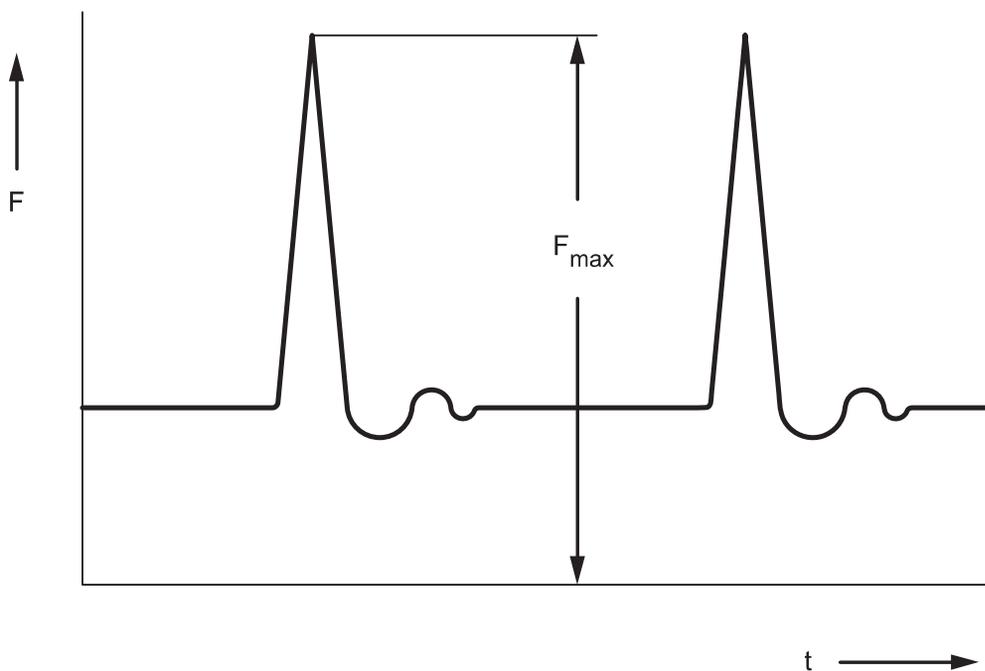   under significant impact load, resp.   [kN]

Fig. 5.29