13

Selection and Design of Rolling Bearings

13.1 INTRODUCTION

Several factors must be considered for an appropriate selection of a rolling bearing for a particular application. The most important factors are load characteristics, speed, lubrication, and environmental conditions. Load characteristics include steady or oscillating load and the magnitude of the radial and axial load components. In selecting a rolling-element bearing, the first two steps are (a) to check whether the bearing can resist the static load and (b) the level of fatigue under oscillating stresses. The fatigue life is estimated in order to ensure that the bearing will not fail prematurely.

In Chapter 12, bearing selection based on basic principles was discussed. The selection was based on stress calculations using Hertz equations for calculating the maximum normal compression stress (maximum pressure) of a rolling contact. The purpose of the calculations is to make sure that the actual maximum contact pressure does not exceed a certain allowed stress limit, which depends on the bearing material. In this chapter a simplified approach for bearing selection is presented for bearings made of standard materials. This approach is based on empirical and analytical data that is provided in manufacturers' catalogues.

In addition to stress calculations, Chapter 12 discussed the EHD fluid film equations. The EHD equations are used for optimum selection of a rolling bearing in combination with an appropriate lubricant. However, in this chapter, the selection of a rolling bearing and lubricant are simplified for standard bearings. Empirical data in the form of charts is used for the selection of bearing and lubricant.

13.1.1 Static Load

Selection of bearings by means of static load calculations is necessary only for slow speeds, because at higher speeds, the requirement for fatigue resistance is much more demanding. This means that bearings that are selected via fatigue calculations are usually loaded much below the static load limits.

For standard rolling bearings that are made of hardened steel, the ISO standard has been set to limit the maximum stress in order to prevent excessive permanent (plastic) deformation. In fact, the ISO standard is limiting the total plastic deformation of the rolling element and raceway. The total plastic deformation limits to ($10^{-4} D_c$), where D_c is the circular orbit diameter of a rolling-element center.

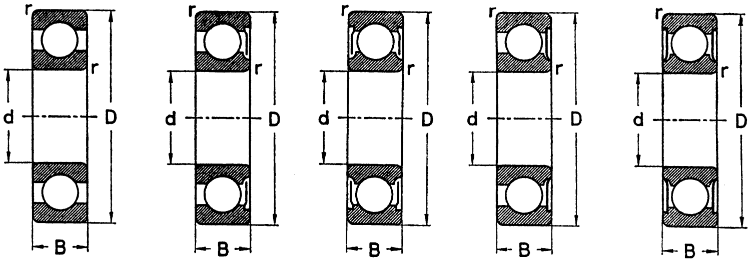
A modified calculation method, which limits the maximum contact stresses, was adopted recently. The international standard ISO 76 was revised, and the Antifriction Bearing Manufacturers Association (AFBMA) in 1986 adopted this method of calculation of static loads. In order to satisfy the plastic deformation limit, the compression stress limits applied to various rolling element bearings made of standard hardened steel are:

Ball bearings:	4200 MPa
Self-aligning ball bearings:	4600 MPa
Roller bearings:	4000 MPa

The basic static load rating C_0 is defined as the static load that results in the calculated stress limit at the center of the rolling contact area where there is maximal compression stress. For radial bearings, the radial static load, C_{0r}, is limited; for thrust bearings, the axial load C_{0a} is limited.

For standard steel bearings, the values of the static load rating C_0 are given in manufacturers' catalogues. Examples are in Tables 13-1 through 13-4. These values are helpful because the designer can rely on the maximum load without resorting to calculations based on Hertz equations. However, better materials are often used; in such cases the designer should use Hertz equations for determining the maximum allowed bearing load from basic principles.

TABLE 13-1 Dimensions and Load Ratings for Deep Ball Bearing Series 6300. (From FAG Bearing Catalogue, with permission)



BEARING OF STANDARD DESIGN

BEARING WITH ONE SHIELD

BEARING WITH TWO SHIELDS

BEARING WITH ONE SEAL

BEARING WITH TWO SEALS

Number					Dimensions				Load ratings	
Bearing of standard design	Bearing with one shield	Bearing with two shields	Bearing with one seal	Bearing with two seals	d	D	B	r	Dynamic C lbs	Static C_0 lbs
						mm				
6300	6300Z	6300.2Z	6300 RS	6300.2RS	10	35	11	1	1400	850
6301	6301Z	6301.2Z	6301 RS	6301.2RS	12	37	12	1.5	1700	1040
6302	6302Z	6302.2Z	6302 RS	6302.2RS	15	42	13	1.5	1930	1200
6303	6303Z	6303.2Z	6303 RS	6303.2RS	17	47	14	1.5	2320	1460
6304	6304Z	6304.2Z	6304 RS	6304.2RS	20	52	15	2	3000	1930
6305	6305Z	6305.2Z	6305 RS	6305.2RS	25	62	17	2	3800	2550
6306	6306Z	6306.2Z	6306 RS	6306.2RS	30	72	19	2	5000	3400
6307	6307Z	6307.2Z	6307 RS	6307.2RS	35	80	21	2.5	5700	4000
6308	6308Z	6308.2Z	6308 RS	6308.2RS	40	90	23	2.5	7350	5300

6309	6309Z	6309.2Z	6309 RS	6309.2RS	45	100	25	2.5	9150	6700
6310	6310Z	6310.2Z	6310 RS	6310.2RS	50	110	27	3	10600	8150
6311	6311Z	6311.2Z			55	120	29	3	12900	10000
6312	6312Z	6312.2Z			60	130	31	3.5	14000	10800
6313	6313Z	6313.2Z			65	140	33	3.5	16000	12500
6314	6314Z	6314.2Z			70	150	35	3.5	18000	14000
6315					75	160	37	3.5	19300	16300
6316					80	170	39	3.5	19600	16300
6317					85	180	41	4	21600	18600
6318					90	190	43	4	23200	20000
6319					95	200	45	4	24500	22400
6320					100	215	47	4	28500	27000
6321					105	225	49	4	30500	30000
6322					110	240	50	4	32500	32500
6324					120	260	55	4	36000	38000
6326					130	280	58	5	39000	43000
6328					140	300	62	5	44000	50000
6330					150	320	65	5	49000	60000

TABLE 13-2 Angular Contact Bearing of Series 909 $\alpha = 25°$, separable. (From FAG Bearing Catalogue, with permission)

EQUIVALENT DYNAMIC LOAD

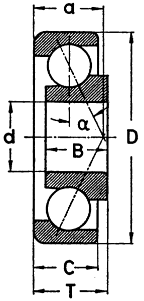
$P = F_r$ when $\frac{F_a}{F_r} \leq 0.68$

$P = 0.41\ F_r + 0.87\ F_a$ when $\frac{F_a}{F_r} > 0.68$

EQUIVALENT STATIC LOAD

$Po = F_r$ when $\frac{F_a}{F_r} \leq 1.3$

$Po = 0.5\ F_r + 0.38\ F_a$ when $\frac{F_a}{F_r} > 1.3$



Number	Dimensions inch						Max. fillet radius inch	Load ratings	
	d	D	B	C	T	a		dynamic C lbs	static C_0 lbs
909001	.7503	2.0800	.5950	.6080	.7080	.65	.060	3250	2240
909002	1.1904	2.9630	.8700	.7700	1.1450	.91	.060	6100	4400
909003	.8128	2.4370	.6880	.7290	.8290	.73	.100	4550	3200
909004	1.2815	3.3750	.9640	.9330	1.3080	1.06	.010	8300	6300
909007	.9379	3.0300	.8440	.9310	1.0310	.96	.060	7500	5600
909008	1.4384	3.9300	1.0580	1.0950	1.4700	1.20	.100	11400	9150
909021	.6875	1.8750	.5630	.5630	.6880	.63	.060	3050	2040
909022	1.1250	2.5000	.8440	.6250	.9840	.75	.060	4150	2900
909023	.7503	2.2500	.6590	.6900	.7900	.73	.100	4500	3200

909024	1.3128	3.1496	.9170	.8510	1.2260	.98	.100	7350	5600
909025	.8440	2.2500	.6590	.6900	.7900	.75	.060	4300	3100
909026	1.4065	3.1496	.9170	.8510	1.2260	.98	.060	6550	5500
909027	.9379	2.8125	.8000	.8500	.9100	.91	.100	6300	4550
909028	1.5000	3.7500	1.0700	1.0150	1.4500	1.18	.100	9800	7650
909029	1.1250	3.1875	.8750	.9730	1.0730	1.02	.100	8300	6300
909030	1.6250	4.0625	1.1875	1.0950	1.5620	1.20	.100	11400	9150
909052	1.2815	2.9630	.8700	.7700	1.1450	.91	.060	6100	4400
909062	1.3750	2.9630	.8700	.7700	1.1450	.91	.060	6000	4550
909067	.7502	2.0800	.4690	.4690	.7080	.69	.040	3900	2750
909070	1.2500	2.6500	.7000	.5150	.8000	.81	.040	5300	3800

TABLE 13-3 Angular Contact Ball Bearings Series 73B, $\alpha = 40°C$, Non-separable. (From FAG Bearing Catalogue, with permission)

EQUIVALENT DYNAMIC LOAD

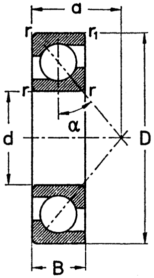
$P = F_r$ when $\frac{F_a}{F_r} \leq 1.14$

$P = 0.35\ F_r + 0.57\ F_a$ when $\frac{F_a}{F_r} > 1.14$

EQUIVALENT STATIC LOAD

$P_o = F_r$ when $\frac{F_a}{F_r} \leq 1.9$

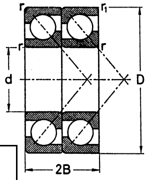
$P_o = 0.5\ F_r + 0.26\ F_a$ when $\frac{F_a}{F_r} > 1.9$



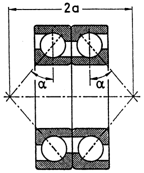
	Dimensions										Max. fillet radius for		Load ratings dynamic	static
	d	D	B	r	r_1	a	d	D	B	a	r	r_1	C	0
Number	mm						inch				inch		lbs	lbs
7300B	10	35	11	1	.5	15	.3937	1.3780	.4331	.59	.025	.012	1460	830
7301B	12	37	12	1.5	.8	16	.4724	1.4567	.4724	.63	.040	.020	1830	1080
7302B	15	42	13	1.5	.8	18	.5906	1.6535	.5118	.71	.040	.020	2240	1340
7303B	17	47	14	1.5	.8	20	.6693	1.8504	.5512	.79	.040	.020	2750	1730
7304B	20	52	15	2	1	23	.7874	2.0472	.5906	.91	.040	.025	3250	2120
7305B	25	62	17	2	1	27	.9842	2.4409	.6693	1.06	.040	.025	4500	3050
7306B	30	72	19	2	1	31	1.1811	2.8346	.7480	1.22	.040	.025	5600	3900
7307B	35	80	21	2.5	1.2	35	1.3780	3.1496	.8268	1.38	.060	.030	6800	4800
7308B	40	90	23	2.5	1.2	39	1.5748	3.5433	.9055	1.54	.060	.030	8650	6300
7309B	45	100	25	2.5	1.2	43	1.7716	3.9370	.9842	1.69	.060	.030	10200	7800
7310B	50	110	27	3	1.5	47	1.9685	4.3307	1.0630	1.85	.080	.040	12000	9300

7311B	55	120	29	3	1.5	51	2.1654	4.7244	1.1417	2.01	.080	.040	13400	10800
7312B	60	130	31	3.5	2	55	2.3622	5.1181	1.2205	2.17	.080	.040	15600	12500
7313B	65	140	33	3.5	2	60	2.5590	5.5118	1.2992	2.36	.080	.040	17600	14300
7314B	70	150	35	3.5	2	64	2.7559	5.9055	1.3780	2.52	.080	.040	19600	16300
7315B	75	160	37	3.5	2	68	2.9528	6.2992	1.4567	2.68	.080	.040	22000	19000
7316B	80	170	39	3.5	2	72	3.1496	6.6929	1.5354	2.83	.080	.040	24000	22000
7317B	85	180	41	4	2	76	3.3464	7.0866	1.6142	2.99	.10	.040	26000	24000
7318B	90	190	43	4	2	80	3.5433	7.4803	1.6929	3.15	.10	.040	28000	27000
7319B	95	200	45	4	2	84	3.7402	7.8740	1.7716	3.31	.10	.040	30000	29000
7320B	100	215	47	4	2	90	3.9370	8.4646	1.8504	3.54	.10	.040	33500	34000
7321B	105	225	49	4	2	94	4.1338	8.8582	1.9291	3.70	.10	.040	35500	38000
7322B	110	240	50	4	2	98	4.3307	9.4488	1.9685	3.86	.10	.040	38000	43000

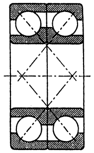
TABLE 13-4 Angular Contact Ball Bearings. (From FAG Bearing Catalogue, with permission)



TANDEM ARRANGEMENT



BACK-TO-BACK
(O ARRANGEMENT)


FACE-TO-FACE
(X ARRANGEMENT)

EQUIVALENT DYNAMIC LOAD

Tandem arrangement $P = F_r$ when $\frac{F_a}{F_r} \leq 1.14$

$P = 0.35\ F_r + 0.57\ F_a$ when $\frac{F_a}{F_r} > 1.14$

O and X arrangements $P = F_r + 0.55\ F_a$ when $\frac{F_a}{F_r} \leq 1.14$

$P = 0.57\ F_r + 0.93\ F_a$ when $\frac{F_a}{F_r} > 1.14$

EQUIVALENT STATIC LOAD

Tandem arrangement $P_o = F_r$ when $\frac{F_a}{F_r} \leq 1.9$

$P = 0.5\ F_r + 0.26\ F_a$ when $\frac{F_a}{F_r} > 1.9$

O and X arrangements $P_o = F_r + 0.52\ F_a$

Bearing pair number			Dimensions mm						Dimensions inch				Max. fillet radius for inch		Load ratings for bearing pair dynamic	static
			d	D	2B	r	r_1	2a	d	D	2B	2a	r	r_1	C^1 lbs	C_0 lbs
2 × 7300 B.UA	2 × 7300 B.UO	2 × 7300 B.UL	10	35	22	1	.5	30	.3937	1.3780	.8661	1.18	.025	.012	2360	1660
2 × 7301 B.UA	2 × 7301 B.UO	2 × 7301 B.UL	12	37	24	1.5	.8	33	.4724	1.4567	.9449	1.26	.040	.020	3050	2160
2 × 7302 B.UA	2 × 7302 B.UO	2 × 7302 B.UL	15	42	26	1.5	.8	37	.5906	1.6535	1.0236	1.42	.040	.020	3600	2750
2 × 7303 B.UA	2 × 7303 B.UO	2 × 7303 B.UL	17	47	28	1.5	.8	41	.6693	1.8504	1.1024	1.57	.040	.020	4500	3400
2 × 7304 B.UA	2 × 7304 B.UO	2 × 7304 B.UL	20	52	30	2	1	45	.7874	2.0472	1.1811	1.81	.040	.025	5300	4250
2 × 7305 B.UA	2 × 7305 B.UO	2 × 7305 B.UL	25	62	34	2	1	53	.9842	2.4409	1.3386	2.13	.040	.025	7350	6100
2 × 7306 B.UA	2 × 7306 B.UO	2 × 7306 B.UL	30	72	38	2	1	62	1.1811	2.8346	1.4961	2.44	.040	.025	9150	7800

2 × 7307 B.UA	2 × 7307 B.UO	2 × 7307 B.UL	35	80	42	2.5	1.2	69	1.3780	3.1496	1.6535	2.76	.060	.030	11000	9650
2 × 7308 B.UA	2 × 7308 B.UO	2 × 7308 B.UL	40	90	46	2.5	1.2	78	1.5748	3.5433	1.8110	3.07	.060	.030	13700	12500
2 × 7309 B.UA	2 × 7309 B.UO	2 × 7309 B.UL	45	100	50	2.5	1.2	86	1.7716	3.9370	1.9685	3.39	.060	.030	17000	15300
2 × 7310 B.UA	2 × 7310 B.UO	2 × 7310 B.UL	50	110	54	3	1.5	94	1.9685	4.3307	2.1260	3.70	.080	.040	19600	18600
2 × 7311 B.UA	2 × 7311 B.UO	2 × 7311 B.UL	55	120	58	3	1.5	102	2.1654	4.7244	2.2835	4.02	.080	.040	22000	21600
2 × 7312 B.UA	2 × 7312 B.UO	2 × 7312 B.UL	60	130	62	3.5	2	111	2.3622	5.1181	2.4409	4.33	.080	.040	25000	25000
2 × 7313 B.UA	2 × 7313 B.UO	2 × 7313 B.UL	65	140	66	3.5	2	119	2.5590	5.5118	2.5984	4.72	.080	.040	28500	28500
2 × 7314 B.UA	2 × 7314 B.UO	2 × 7314 B.UL	70	150	70	3.5	2	127	2.7559	5.9055	2.7559	5.04	.080	.040	32000	32500
2 × 7315 B.UA	2 × 7315 B.UO	2 × 7315 B.UL	75	160	74	3.5	2	136	2.9528	6.2992	2.9134	5.35	.080	.040	35500	37500
2 × 7316 B.UA	2 × 7316 B.UO	2 × 7316 B.UL	80	170	78	3.5	2	144	3.1496	6.6929	3.0709	5.67	.080	.040	39000	44000
2 × 7317 B.UA	2 × 7317 B.UO	2 × 7317 B.UL	85	180	82	4	2	152	3.3464	7.0866	3.2283	5.98	.10	.040	42500	48000
2 × 7318 B.UA	2 × 7318 B.UO	2 × 7318 B.UL	90	190	86	4	2	160	3.5433	7.4803	3.38583	6.30	.10	.040	45000	54000
2 × 7319 B.UA	2 × 7319 B.UO	2 × 7319 B.UL	95	200	90	4	2	169	3.7402	7.8740	3.5433	6.61	.10	.040	48000	58500
2 × 7320 B.UA	2 × 7320 B.UO	2 × 7320 B.UL	100	215	94	4	2	179	3.9370	8.4646	3.7008	7.09	.10	.040	54000	69500
2 × 7321 B.UA	2 × 7321 B.UO	2 × 7321 B.UL	105	225	98	4	2	187	4.1338	8.8582	3.8583	7.40	.10	.040	58500	76500
2 × 7322 B.UA	2 × 7322 B.UO	2 × 7322 B.UL	110	240	100	4	2	197	4.3307	9.4488	3.9370	7.72	.10	.040	64000	86500

13.1.2 Permissible Static Load and Safety Coefficients

The operation of most machines is associated with vibrations and disturbances. The vibrations result in dynamic forces: in turn, the actual maximum stress can be much higher than that calculated by the static load. Therefore, engineers always use a safety coefficient, f_s. In addition, whenever there is a requirement for low noise, the maximum permissible load is reduced to much lower value than C_0. Low loads would result in a significant reduction of permanent deformation of the races and rolling-element surfaces. Plastic deformation distorts the bearing geometry and causes noise during bearing operation.

The permissible static load on a bearing, P_0, is usually less than the basic static load rating, C_0, according to the equation

$$P_0 = \frac{C_0}{f_s} \tag{13-1}$$

The safety coefficient, f_s, depends on the operating conditions and bearing type. Common guidelines for selecting a safety coefficient, f_s are in Table 13-5.

13.1.3 Static Equivalent Load

Most bearings in machinery are subjected to combined radial and thrust loads. It is necessary to establish the combination of radial and thrust loads that would result in the limit stress of a particular bearing. Static equivalent load is introduced to allow bearing selection under combined radial and thrust forces. It is defined as a hypothetical load (radial or axial) that results in a maximum contact stress equivalent to that under combined radial and thrust forces. In radial bearings, the static equivalent load is taken as a radial equivalent load, while in thrust bearings the static equivalent load is taken as a thrust equivalent load.

TABLE 13-5 Safety Coefficient, f_s for Rolling Element Bearings (From FAG 1998)

	For ball bearings	For roller bearings
Standard operating conditions	$f_s = 1$	$f_s = 1.5$
Bearings subjected to vibrations	$f_s = 1.5$	$f_s = 2$
Low-noise applications	$f_s = 2$	$f_s = 3$

13.1.4 Static Radial Equivalent Load

For radial bearings, the higher of the two values calculated by the following two equations is taken as the static radial equivalent load:

$$P_0 = X_0 F_r + Y_0 F_a \tag{13-2}$$

$$P_0 = F_r \tag{13-3}$$

Here,

P_0 = static equivalent load
F_r = static radial load
F_a = static thrust (axial) load
X_0 = static radial load factor
Y_0 = static thrust load factor

Values of X_0 and Y_0 for several bearing types are listed in Table 13-6.

13.1.5 Static Thrust Equivalent Load

For thrust bearings, the static thrust equivalent load is obtained via the following equation:

$$P_0 = X_0 F_r + F_a \tag{13-4}$$

This equation can be applied to thrust bearings for contact angles lower than 90°. The value of X_0 is available in bearing tables in catalogues provided by bearing

TABLE 13-6 Values of Coefficients X_0 and Y_0 (From SKF, 1992, with permission)

Bearing type	Single row bearings X_0	Single row bearings Y_0	Double row bearings X_0	Double row bearings Y_0
Deep groove ball bearings*	0.6	0.5	0.6	0.5
Angular contact ball bearings				
$\alpha = 15°$	0.5	0.46	1	0.92
$\alpha = 20°$	0.5	0.42	1	0.84
$\alpha = 25°$	0.5	0.38	1	0.76
$\alpha = 30°$	0.5	0.33	1	0.66
$\alpha = 35°$	0.5	0.29	1	0.58
$\alpha = 40°$	0.5	0.26	1	0.52
$\alpha = 45°$	0.5	0.22	1	0.44
Self-aligning ball bearings	0.5	0.22ctgα	1	0.44ctgα

*Permissible maximum value of F_a/C_0 depends on bearing design (internal clearance and raceway groove depth).

manufacturers. For a contact angle of 90°, the static thrust equivalent load is $P_0 = F_a$.

13.2 FATIGUE LIFE CALCULATIONS

The rolling elements and raceways are subjected to dynamic stresses. During operation, there are cycles of high contact stresses oscillating at high frequency that cause metal fatigue. The fatigue life—that is, the number of cycles (or the time in hours) to the initiation of fatigue damage in identical bearings under identical load and speed—has a statistical distribution. Therefore, the fatigue life must be determined by considering the statistics of the measured fatigue life of a large number of dimensionally identical bearings.

The method of estimation of fatigue life of rolling-element bearings is based on the work of Lundberg and Palmgren (1947). They used the fundamental theory of the maximum contact stress, and developed a statistical method for estimation of the fatigue life of a rolling-element bearing. This method became a standard method that was adopted by the American Bearing Manufacturers' Association (ABMA). For ball bearings, this method is described in standard ANSI/ABMA-9, 1990; for roller bearings it is described in standard ANSI/ABMA-11, 1990.

13.2.1 Fatigue Life, L_{10}

The *fatigue life*, L_{10}, (often referred to as *rating life*) is the number of revolutions (or the time in hours) that 90% of an identical group of rolling-element bearings will complete or surpass its life before any fatigue damage is evident. The tests are conducted at a given constant speed and load.

Extensive experiments have been conducted to understand the statistical nature of the fatigue life of rolling-element bearings. The experimental results indicated that when fatigue life is plotted against load on a logarithmic scale, a negative-slope straight line could approximate the curve. This means that fatigue life decreases with load according to power-law function. These results allowed the formulation of a simple equation with empirical parameters for predicting the fatigue life of each bearing type.

The following fundamental equation considers only bearing load. Life adjustment factors for operating conditions, such as lubrication, will be discussed later. The fatigue life of a rolling-element bearing is determined via the equation

$$L_{10} = \left(\frac{C}{P}\right)^k \qquad \text{[in millions of revolutions]} \qquad (13\text{-}5)$$

Here, C is the *dynamic load rating* of the bearing (also referred to as the *basic load rating*), P is the equivalent radial load, and k is an empirical exponential parameter ($k = 3$ for ball bearings and $10/3$ for roller bearings). The units of C and P can be pounds or newtons (SI units) as long as the units for the two are consistent, since the ratio C/P is dimensionless.

Engineers are interested in the life of a machine in hours. In industry, machines are designed for a minimum life of five years. The number of years depends on the number of hours the machine will operate per day. Equation (13-5) can be written in terms of hours:

$$L_{10} = \frac{10^6}{60N}\left(\frac{C}{P}\right)^k \qquad \text{[in hours]} \tag{13-6}$$

13.2.2 Dynamic Load Rating, *C*

The dynamic load rating, C, is defined as the radial load on a rolling bearing that will result in a fatigue life of 1 million revolutions of the inner ring. Due to the statistical distribution of fatigue life, at least 90% of the bearings will operate under load C without showing any fatigue damage after 1 million revolutions. The value of C is determined empirically, and it depends on bearing type, geometry, precision, and material. The dynamic load rating C is available in bearing catalogues for each bearing type and size. The actual load on a bearing is always much lower than C, because bearings are designed for much longer life than 1 million revolutions.

The dynamic load rating C has load units, and it depends on the design and material of a specific bearing. For a radial ball bearing, it represents the experimental steady radial load under which the radial bearing endured a fatigue life, L_{10}, of 10^6 revolutions.

To determine the dynamic load rating, C, a large number of identical bearings are subjected to fatigue life tests. In these tests, a steady load is applied, and the inner ring is rotating while the outer ring is stationary. The fatigue life of a large number of bearings of the same type is tested under various radial loads.

13.2.3 Combined Radial and Thrust Loads

The *equivalent radial load P* is the radial load, which is equivalent to combined radial and thrust loads. This is the constant radial load that, if applied to a bearing with rotating inner ring and stationary outer ring, would result in the same fatigue life the bearing would attain under combined radial and thrust loads, and different rotation conditions.

In Eq. (13-5), P is the equivalent dynamic radial load, similar to the static radial load. If the load is purely radial, P is equal to the bearing load. However, when the bearing is subjected to combined radial and axial loading, the equivalent load, P, is determined by:

$$P = XVF_r + YF_a \tag{13-7}$$

Here,

$P =$ equivalent radial load

$F_r =$ bearing radial load

$F_a =$ bearing thrust (axial) load

$V =$ rotation factor: 1.0 for inner ring rotation, 1.2 for outer ring rotation and for a self-aligning ball bearing use 1 for inner or outer rotation

$X =$ radial load factor

$Y =$ thrust load factor

The factors X and Y differ for various bearings (Table 13-7).

The equivalent load (P), is defined by the Anti-Friction Bearings Manufacturers Association (AFBMA). It is the constant stationary radial load that, if applied to a bearing with rotating inner and stationary outer ring, would give the same life as what the bearing would attain under the actual conditions of load and rotation.

13.2.4 Life Adjustment Factors

Recent high-speed tests of modern ball and roller bearings, which combine improved materials and proper lubrication, show that fatigue life is, in fact, longer than that predicted previously from Eq. (12-5). It is now commonly accepted that an improvement in fatigue life can be expected from proper lubrication, where the rolling surfaces are completely separated by an elastohydrodynamic lubrication film. In Sec. 13.4 the principles of rolling-element bearing lubrication are discussed. For a rolling bearing with adequate EHD lubrication, adjustments to the fatigue life should be applied. The adjustment factor is dependent on the operating speed, bearing temperature, lubricant viscosity, size and type of bearing, and bearing material.

In many applications, higher reliability is required, and 10% probability of failure is not acceptable. Higher reliability, such as L_5 (5% failure probability) or L_1 (failure probability of 1%), is applied. As defined in the AFBMA Standards, fatigue life is calculated according to the equation

$$L_{na} = a_1 a_2 a_3 \left(\frac{C}{P}\right)^P \times 10^6 \text{ (revolutions)} \tag{13-8}$$

TABLE 13-7 Factors X and Y for Radial Bearings. (From FAG Bearing Catalogue, with permission)

			Single row bearings[1]		Double row bearings[2]				
			$\frac{F_a}{F_r} > e$[1]		$\frac{F_a}{F_r} \le e$		$\frac{F_a}{F_r} > e$		e
Bearing type	[3] $\frac{F_a}{C_0}$	[3] $\frac{F_a}{iZD_w^2}$	X	Y	X	Y_1	X	Y_2	
Radial Contact Groove Ball Bearings	0.014	25	0.56	2.30	1	0	0.56	2.30	0.19
	0.028	50		1.99				1.99	0.22
	0.056	100		1.71				1.71	0.26
	0.084	150		1.55				1.55	0.28
	0.11	200		1.45				1.45	0.30
	0.17	300		1.31				1.31	0.34
	0.28	500		1.15				1.15	0.38
	0.42	750		1.04				1.04	0.42
	0.56	1000		1.00				1.00	0.44
20°			0.43	1.00		1.09	0.70	1.63	0.57
25°			0.41	0.87		0.92	0.67	1.44	0.68

(*continued*)

TABLE 13-7 Continued.

	Single row bearings[1]		Double row bearings[2]				
	$\frac{F_a}{F_r} > e^1$		$\frac{F_a}{F_r} \le e$		$\frac{F_a}{F_r} > e$		e
Bearing type	X	Y	X	Y_1	X	Y_2	
30°	0.39	0.76	1	0.78	0.63	1.24	0.80
35°	0.37	0.66		0.66	0.60	1.07	0.95
40°	0.35	0.57		0.55	0.57	0.93	1.14
Self-Aligning[6] Ball Bearings	0.40	$0.4 \cot\alpha$	1	$0.42 \cot\alpha$	$0.65 \cot\alpha$	$1.5 \tan\alpha$	
Spherical[6] and Tapered[4,5] Roller Bearings	0.40	$0.4 \cot\alpha$	1	$0.45 \cot\alpha$	0.67	$0.67 \cot\alpha$	$1.5 \tan\alpha$

[1] For single row bearings, when $\frac{F_a}{F_r} \le e$ use $X = 1$ and $Y = 0$.

For two single row angular contact ball or roller bearings mounted "face-to-face" or "back-to-back" use the values of X and Y which apply to double row bearings. For two or more single row bearings mounted "in tandem" use the values of X and Y which apply to single row bearings.

[2] Double row bearings are presumed to be symmetrical.

[3] C_0 = static load rating, i = number of rows of rolling elements. Z = number of rolling elements/row, D_w = ball diameter.

[4] Y values for tapered roller bearings are shown in the bearing tables.

[5] $e = \frac{0.6}{Y}$ for single row tapers, and $e = \frac{1}{Y_2}$ for double tow tapers.

TABLE 13-8 Life Adjustment Factor a_1 for Different Failure Probabilities

Failure probability, n					
10	5	4	3	2	1
1	0.62	0.53	0.44	0.33	0.1

where

L_{na} = adjusted fatigue life for a reliability of $(100 - n)\%$, where n is a failure probability (usually, $n = 10$)
a_1 = life adjustment factor for reliability ($a_1 = 1.0$ for $L_n = L_{10}$) (Table 13-8)
a_2 = life adjustment factor for bearing materials made from steel having a higher impurity level
a_3 = life adjustment factor for operating conditions, particularly lubrication (see Sec. 13.4)

Example Problem 13-2 demonstrates the calculation of adjusted rating life; see Sec. 13.4 on bearing lubrication. Experience indicated that the value of the two parameters a_2 and a_3 ultimately depends on proper lubrication conditions. Without proper lubrication, better materials will have no significant benefit in improvement of bearing life. However, better materials have merit only when combined with adequate lubrication. Therefore, the life adjustment factors a_2 and a_3 are often combined, $a_{23} = a_2 a_3$.

13.3 BEARING OPERATING TEMPERATURE

Advanced knowledge of rolling bearing operating temperature is important for bearing design, lubrication, and sealing. Attempts have been made to solve for the bearing temperature at steady-state conditions. The heat balance equation was used, equating the heat generated by friction (proportional to speed and load) to the heat transferred (proportional to temperature rise). It is already recognized that analytical solutions do not yield results equal to the actual operating temperature, because the bearing friction coefficient and particularly the heat transfer coefficients are not known with an adequate degree of precision. For these reasons, we can use only approximations of average bearing operating temperature for design purposes. The temperature of the operating bearing is not uniform. The point of maximum temperature is at the contact of the races with the rolling elements. At the contact with the inner race, the temperature is higher than that of the contact with the outer race. However, for design purposes, an average (approximate) bearing temperature is considered. The average oil temperature is lower than that of the race surface. It is the average of inlet and outlet oil temperatures.

Several attempts to present precise computer solutions are available in the literature. Harris (1984) presented a description of the available numerical methods for solving the temperature distribution in a rolling bearing. Numerical calculation of the bearing temperature is quite complex, because it depends on a large number of heat transfer parameters.

For simplified calculations, it is possible to estimate an average bearing temperature by considering the bearing friction power losses and heat transfer. Friction power losses are dissipated in the bearing as heat and are proportional to the product of friction torque and speed. The heat is continually transferred away by convection, radiation, and conduction. This heat balance can be solved for the temperature rise, bearing temperature minus ambient (atmospheric) temperature $(T_b - T_a)$.

More careful consideration of the friction losses and heat transfer characteristics through the shaft and the housing can only help to estimate the bearing temperature rise. This data can be compared to bearings from previous experience where the oil temperature has been measured. It is relatively easy to measure the oil temperature at the exit from the bearing. (The oil temperature at the contact with the races during operation is higher and requires elaborate experiments to be determined).

It is possible to control the bearing operating temperature. In an elevated-temperature environment, the oil circulation assists in transferring the heat away from the bearing. The final bearing temperature rise, above the ambient temperature, is affected by many factors. It is proportional to the bearing speed and load, but it is difficult to predict accurately by calculation. However, for predicting the operating temperature, engineers rely mostly on experience with similar machinery. A comparative method to estimate the bearing temperature is described in Sec. 13.3.1.

A lot of data has been derived by means of field measurements. The bearing temperature for common moderate-speed applications has been measured, and it is in the range of 40°–90°C. The relatively low bearing temperature of 40°C is for light-duty machines such as the bench drill spindle, the circular saw shaft, and the milling machine. A bearing temperature of 50°C is typical of a regular lathe spindle and wood-cutting machine spindle. The higher bearing temperature of 60°C is found in heavier-duty machinery, such as an axle box of train locomotives. A higher temperature range is typical of machines subjected to load combined with severe vibrations. The bearing temperature of motors, of vibratory screens, or impact mills is 70°C; and in vibratory road roller bearings, the higher temperature of 80°C has been measured.

Much higher bearing temperatures are found in machines where there is an external heat source that is conducted into the bearing. Examples are rolls for paper drying, turbocompressors, injection molding machines for plastics, and bearings of large electric motors, where considerable heat is conducted from the motor armature. In such cases, air cooling or water cooling is used in the bearing housing for reducing the bearing temperature. Also, fast oil circulation can help to remove the heat from the bearing.

13.3.1 Estimation of Bearing Temperature

The following derivation is useful where there is already previous experience with a similar machine. In such cases, the temperature rise can be predicted whenever there are modifications in the machine operation, such as an increase in speed or load.

The friction power loss, q, of a bearing is calculated from the frictional torque T_f $[N\text{-}m]$ and the shaft angular speed ω [rad/s]:

$$q = T_f\omega \qquad [W] \tag{13-9}$$

The angular speed can be written as a function of the speed $N[RPM]$:

$$\omega = \frac{2\pi N}{60} \tag{13-10}$$

Under steady-state conditions there is heat balance, and the same amount of heat that is generated by friction, q, must be transferred to the environment. The heat transferred from the bearing is calculated from the difference between the bearing temperature, T_b, and the ambient temperature, T_a, from the size of the heat-transmitting areas A_B $[m^2]$ and the total heat transfer coefficient $U_t\,[W/m^2\text{-}C]$:

$$q = U_t A_B (T_b - T_a) \qquad [W] \tag{13-11}$$

In the case of no oil circulation, all the heat is transferred through the bearing surfaces (in contact with the shaft and housing). Equating the two equations gives

$$T_b - T_a = \frac{\pi N T_f}{30 U_t A_B} \tag{13-12}$$

According to Eq. (13-12), the temperature rise, $T_b - T_a$, is proportional to the speed N and the friction torque, T_f, while all the other terms can form one constant k, which is a function of the heat transfer coefficients and the geometry and material of the bearing and housing:

$$\Delta T = T_b - T_a = k\,N\,T_f \tag{13-13}$$

The friction torque T_f is

$$T_f = f\,R\,F \tag{13-14}$$

where f is the friction coefficient, R is the rolling contact radius, and F is the bearing load. The temperature rise, in Eq. (13–13), can be expressed as

$$\Delta T = (T_b - T_a) = K f N F \tag{13-15}$$

where $K = kR$ is a constant. The result is that the temperature rise, $\Delta T = T_b - T_a$, is proportional to the friction coefficient, speed, and bearing load.

Prediction of the bearing temperature can be obtained by determining the steady-state temperature in a test run and calculating the coefficient K. If the friction coefficient is assumed to be constant, then Eq. (13-15) will allow estimation with sufficient accuracy of the steady-state temperature rise of this bearing for other operating conditions, under various speeds and loads. A better temperature estimation can be obtained if additional data is used concerning the function of the friction coefficient, f, versus speed and load.

In the case of oil circulation lubrication, the oil also carries away heat. This can be considered in the calculation if the lubricant flow rate and inlet and outlet temperatures of the bearing oil are measured.

The bearing temperature can then be calculated by equating

$$q = q_1 + q_2 \qquad [W] \tag{13-16}$$

where q_1 is the heat transferred by conduction according to Eq. (13–11) and q_2 is the heat transferred by convection via the oil circulation.

13.3.2 Operating Temperature of the Oil

For selecting an appropriate lubricant, it is important to estimate the operating temperature of the oil in the bearing. It is possible to estimate the operating oil temperature by measuring the temperature of the bearing housing. If the machine is only in design stages, it is possible to estimate the housing temperature by comparing it to the housing temperature of similar machines. During the operation of standard bearings that are properly designed, the operating temperature of the oil is usually in the range of 3°–11°C above that of the bearing housing. It is relatively simple to measure the housing temperature in an operating machine and to estimate the oil temperature. Knowledge of the oil temperature is important for optimal selection of lubricant, oil replacement, and fatigue life calculations.

Tapered and spherical roller bearings result in higher operating temperatures than do ball bearings or cylindrical roller bearings under similar operating conditions. The reason is the higher friction coefficient in tapered and spherical roller bearings.

13.3.3 Temperature Difference Between Rings

During operation, the shaft temperature is generally higher than the housing temperature. The heat is removed from the outer ring through the housing much faster than from the inner ring through the shaft. There is no good heat transfer through the small contact area between the rolling elements and rings (theoretical point or line contact). Therefore, heat from the inner ring is conducted through the shaft, and heat from the outer ring is conducted through the housing. In general, heat conduction through the shaft is not as effective as through the housing. The outer ring and housing have good heat transfer, because they are in direct contact with the larger body of the machine. In comparison, the inner ring and shaft have more resistance to heat transfer, because the cross-sectional area of the shaft is small in comparison to that of the housing as well as to its smaller surface area, which has lower heat convection relative to the whole machine.

If there is no external source of heat outside the bearing, the operating temperature of the shaft is always higher than that of the housing. For medium-speed operation of standard bearings, if the housing is not cooled, the temperatures of the inner ring are in the range of 5°–10°C higher than that of the outer ring. If the housing is cooled by air flow, the temperature of the inner ring can increase to 15°–20°C higher than that of the outer ring. An example of air cooling of the housing is in motor vehicles, where there is air cooling whenever the car is in motion. It is possible to reduce the temperature difference by means of adequate oil circulation, which assists in the convection heat transfer between the rings.

A higher temperature difference can develop in very high-speed bearings. The temperature difference depends on several factors, such as speed, load, and type of bearing and shape of the housing. This temperature difference can result in additional thermal stresses in the bearing.

13.4 ROLLING BEARING LUBRICATION

13.4.1 Objectives of Lubrication

Various types of grease, oils, and, in certain cases, solid lubricants are used for the lubrication of rolling bearings. Most bearings are lubricated with grease because it provides effective lubrication and does not require expensive supply systems (grease can operate with very simple sealing). In most applications, rolling-element bearings operate successfully with a very thin layer of oil or grease. However, for high-speed applications, such as turbines, oil lubrication is important for removing the heat from the bearing or for formation of an EHD fluid film.

The first objective of liquid lubrication is the formation of a thin elastohydrodynamic lubrication film at the rolling contacts between the rolling elements and the raceways. Under appropriate conditions of load, viscosity, and bearing speed, this film can completely separate the surfaces of rolling elements and raceways, resulting in considerable improvement in bearing life.

The second objective of lubrication is to minimize friction and wear in applications where there is no full EHD film. Experience has indicated that if proper lubrication is provided, rolling bearings operate successfully for a long time under mixed lubrication conditions. In practice, ideal conditions of complete separation are not always maintained. If the height of the surface asperities is larger than the elastohydrodynamic lubrication film, contact of surface asperities will take place, and there is a mixed friction (hydrodynamic combined with direct contact friction).

In addition to pure rolling, there is also a certain amount of sliding contact between the rolling elements and the raceways as well as between the rolling elements and the cage. At the sliding surfaces of a rolling bearing, such as the roller and lip in a roller bearing and at the guiding surface of the cage, a very thin lubricant film can be formed, resulting in mixed friction under favorable conditions. Any sliding contact in the bearing requires lubrication to reduce friction and wear.

The third objective of lubrication (applies to fluid lubricants) is to cool the bearing and reduce the maximum temperature at the contact of the rolling elements and the raceways. For effective cooling, sufficient lubricant circulation should be provided to remove the heat from the bearing. The most effective cooling is achieved by circulating the oil through an external heat exchanger. But even without elaborate circulation, a simple oil sump system can enhance the heat transfer from the bearing by convection. Solid lubricants or greases are not effective in cooling; therefore, they are restricted to relatively low-speed applications.

Additional objectives of lubrication are damping of vibrations, corrosion protection, and removal of dust and wear debris from the raceways via liquid lubricant. A full EHD fluid film plays an important role as a damper. A full EHD fluid film acts as noncontact support of the shaft that effectively isolates vibrations. The fluid film can be helpful in reducing noise and vibrations in a machine.

Lubricants for rolling bearings include liquid lubricants (mineral and synthetic oils), greases, and solid lubricants. The most common liquid lubricants are petroleum-based mineral oils with a long list of additives to improve the lubrication performance. Also, synthetic lubricants are widely used, such as ester, polyglycol, and silicone fluoride. Greases are commonly applied in relatively low-speed applications, where continuous flow for cooling is not essential for successful operation. The most important advantages of grease are that it seals the bearing from dust and provides effective protection from corrosion. To minimize maintenance, sealed bearings are widely used, where the bearing is filled with grease and sealed for the life of the bearing. The grease serves as a matrix that retains the oil. The oil is slowly released from the grease during operation.

In addition to grease, oil-saturated solids, such as oil-saturated polymer, are used successfully for similar applications of sealed bearings. The saturated solid fills the entire bearing cavity and effectively seals the bearing from contaminants. The advantage of oil-saturated polymers over grease is that grease can be filled only into half the bearing internal space in order to avoid churning. In comparison, oil-saturated solid lubricants are available that can fill the complete cavity without causing churning. The oil is released from oil-saturated solid lubricants in a similar way to grease.

Rolling bearings successfully operate in a wide range of environmental conditions. In certain high-temperature applications, liquid oils or greases cannot be applied (they oxidize and deteriorate from the heat) and only solid lubricants can be used. Examples of solid lubricants are PTFE, graphite and molybdenum disulfide (MoS_2). Solid lubricants are effective in reducing friction and wear, but obviously they cannot assist in heat removal as liquid lubricants.

In summary: Lubrication of rolling bearings has several important functions: to form a fluid film, to reduce sliding friction and wear, to transfer heat away from the bearing, to damp vibrations, and to protect the finished surfaces from corrosion. Greases and oils are mostly used. Grease packed sealing is commonly used to protect against the penetration of abrasive particles into the bearing. Reduction of friction and wear by lubrication is obtained in several ways. First, a thin fluid film at high pressure can separate the rolling contacts by forming elastohydrodynamic lubrication. Second, lubrication reduces friction of the sliding contacts that do not involve rolling, such as between the cage and the rolling elements or between the rolling elements and the guiding surfaces. Also, the contacts between the rolling elements and the raceways are not pure rolling, and there is always a certain amount of sliding. Solid lubricants are also effective in reducing sliding friction.

13.4.2 Elastohydrodynamic Lubrication

In Chapter 12, the elastohydrodynamic (EHD) lubrication equations were discussed. EHD theory is concerned with the formation of a thin fluid film at high pressure at the contact area of a rolling element and a raceway under rolling conditions. Both the roller and the raceway surfaces are deformed under the load. In a similar way to fluid film in plain bearings, the oil that is adhering to the surfaces is drawn into a thin clearance formed between the rolling surfaces. An important effect is that the viscosity of the oil rises under high pressure; in turn, a load-carrying fluid film is formed at high rolling speed. The clearance thickness, h_0, is nearly constant along the fluid film, and it is reducing only near the outlet side (Fig. 12-20).

Under high loads, the EHD pressure distribution is similar to the pressure distribution according to the Hertz equations, because the influence of the elastic deformations dominates the pressure distribution. But at high speeds, the hydrodynamic effect prevails.

In Chapter 12, the calculation of the film thickness was quite complex. For many standard applications, engineers often resort to a simplified method based on charts. The simplified approach also considers the effect of the elastohydrodynamic lubrication in improving the fatigue life of the bearing. Even if the EHD fluid film does not separate completely the rolling surfaces (mixed EHD lubrication), the lubrication improves the performance, and longer fatigue life will be obtained. In this chapter, the use of charts is demonstrated for finding the effect of lubrication in improving the fatigue life of a bearing.

13.4.3 Selection of Liquid Lubricants

The best performance of a rolling bearing is under operating conditions where the elastohydrodynamic minimum film thickness, $h_{\min}$, is thicker than the surface asperities, R_s. The required viscosity of the lubricant, μ, for this purpose can be solved for from the EHD equations (see Chapter 12). However, for many standard applications, designers determine the viscosity by a simpler practical method. It is based on an empirical chart, where the required viscosity is determined according to the bearing speed and diameter.

For rolling bearings, the decision concerning the oil viscosity is a compromise between the requirement of low viscous friction (low viscosity) and the requirement for adequate EHD film thickness (high viscosity). The friction of a rolling bearing consists of two components. The first component is the rolling friction, which results from deformation at the contacts between a rolling element and a raceway. The second friction component is viscous resistance of the lubricant to the motion of the rolling elements. The first component of rolling friction is a function of the elastic modulus, geometry, and bearing load. The second component of viscous friction increases with lubricant viscosity, quantity of oil in the bearing, and bearing speed. The viscous component increases with speed, so it becomes a dominant factor in high–speed machinery.

It is possible to minimize the viscous resistance by applying a very small quantity of oil, just sufficient to form a thin layer over the contact surface. In addition, using low-viscosity oil can reduce the viscous resistance. However, minimum lubricant viscosity must be maintained to ensure elastohydrodynamic lubrication with adequate fluid film thickness.

For lubricant selection, a knowledge of the operating bearing temperature is required. One must keep in mind that the lubricant viscosity decreases with temperature. In applications where the bearing temperature is expected to rise significantly, lubricant of higher initial viscosity should be selected. It is possible to reduce the bearing operating temperature via oil circulation for removing the heat and cooling the bearing. The final bearing temperature rise, above the ambient temperature, is affected by many factors, such as speed and load. A simplified method for estimating the bearing temperature was discussed earlier. For predicting the operating temperature, this method relies mostly on experience with similar machinery for determining the heat transfer coefficients.

For bearings that do not dissipate heat from outside the bearing and that operate at moderate speeds and under average loads, it is possible to estimate the oil temperature by measuring the housing temperature. During operation, the temperature of the oil is usually in the range of 3°–11°C above that of the bearing housing. This simple temperature estimation is widely used for lubricant selection.

In order to simplify the selection of oil viscosity, charts based on bearing speed and bearing average diameter are used. Figure 13-1 is used for determining the minimum oil viscosity for lubrication of rolling-element bearings as a function of bearing size and speed.

The ordinate on the left side shows the kinematic viscosity in metric units, mm^2/s (cSt). The ordinate on the right side shows the viscosity in Saybolt universal seconds (SUS). The abscissa is the pitch diameter, d_m, in mm, which is the average of internal bore, d, and outside bearing diameter, D.

$$d_m = \frac{d + D}{2} \tag{13-17}$$

The diagonal straight lines in Fig. 13-1 are for the various bearing speed N in *RPM* (revolutions per minute). The dotted lines show examples of determining the required lubricant viscosity.

Example Problem 13-1

Calculation of Minimum Viscosity

A rolling bearing has a bore diameter $d = 45$ mm and an outside diameter $D = 85$ mm. The bearing rotates at 2000 RPM. Find the required minimum viscosity of the lubricant.

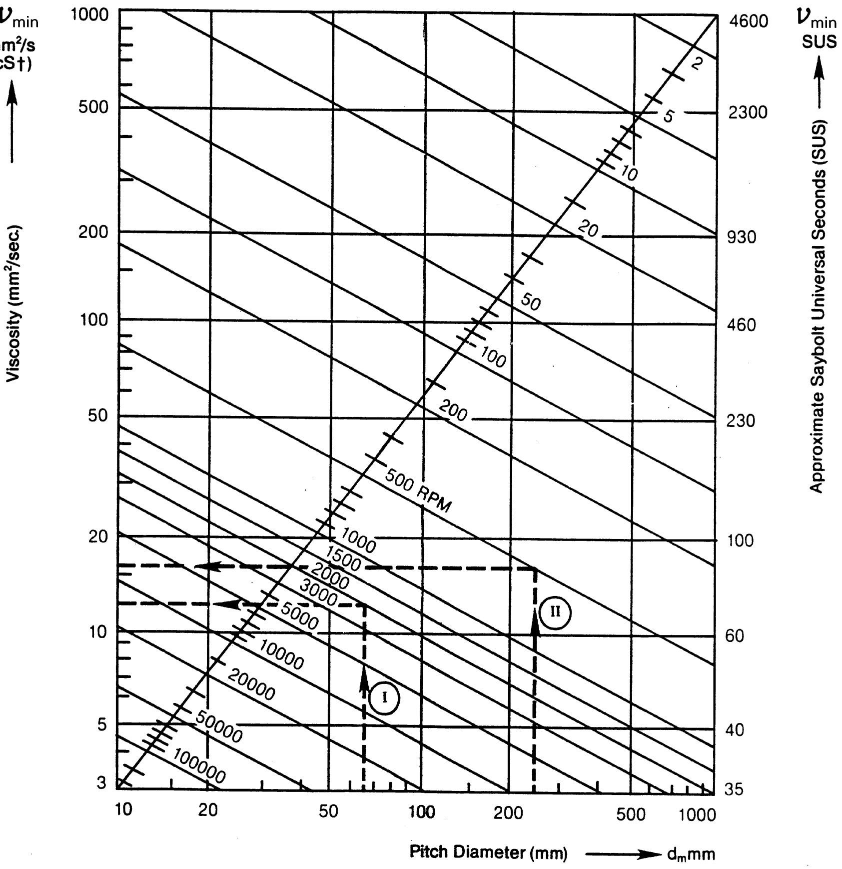


FIG. 13-1 Requirement for minimum lubricant viscosity in rolling bearings (from SKF, 1992, with permission).

Solution

The pitch diameter according to Eq. (13-17) is

$$d_m = \frac{45 + 85}{2} = 65 \text{ mm}$$

Line I in Fig. 13-1 shows the intersection of $d_m = 65$ with the diagonal straight line of 2000 RPM. The horizontal dotted line indicates a minimum viscosity required of 13 cSt (mm^2/s).

Based on the required viscosity, the oil grade should be selected. The oil viscosity decreases with temperature, and the relation between the oil grade and

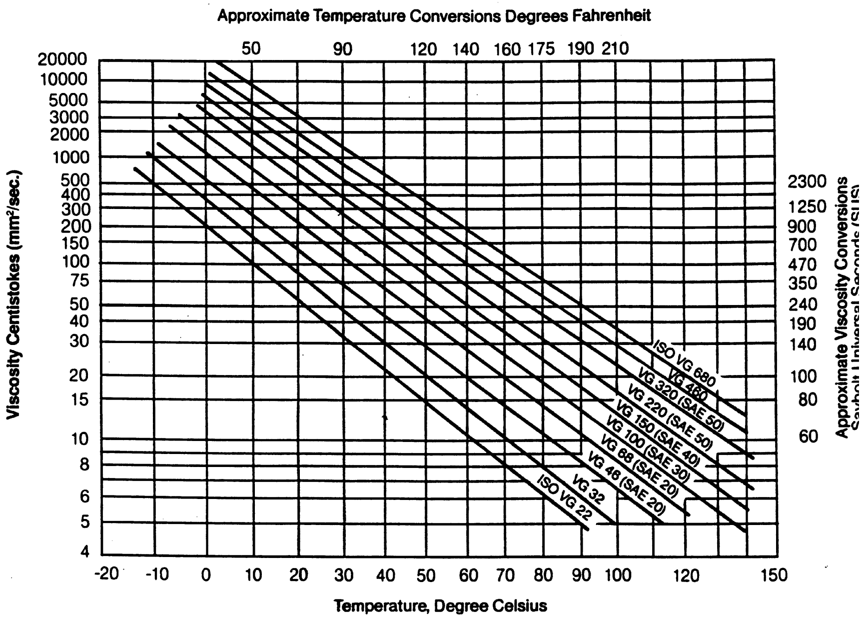


FIG. 13-2 Viscosity–temperature charts (from SKF, 1992, with permission).

its viscosity depends on the oil temperature. In Fig. 13-2, viscosity–temperature charts for several rolling bearing oil grades are presented. Estimation of the oil temperature inside the operating bearing is required before one can select the oil grade according to Fig. 13-2.

It is preferable to estimate the temperature with an error on the high side. This would result in higher viscosity, which can ensure a full EHD fluid film at the rolling contact, although the friction resistance can be slightly higher. If a lubricant with higher-than-required viscosity is selected, an improvement in bearing life can be expected. However, since a higher viscosity raises the bearing operating temperature, there is a limit to the improvement that can be obtained in this manner.

The improvement in the bearing fatigue life due to higher lubricant viscosity (above the minimum required viscosity) is shown in Fig. 13-3. The life adjustment factor a_3 (sec. 13.2.4) is a function of the viscosity ratio, κ, defined as

$$\kappa = \frac{\nu}{\nu_{\min}} \tag{13-18}$$

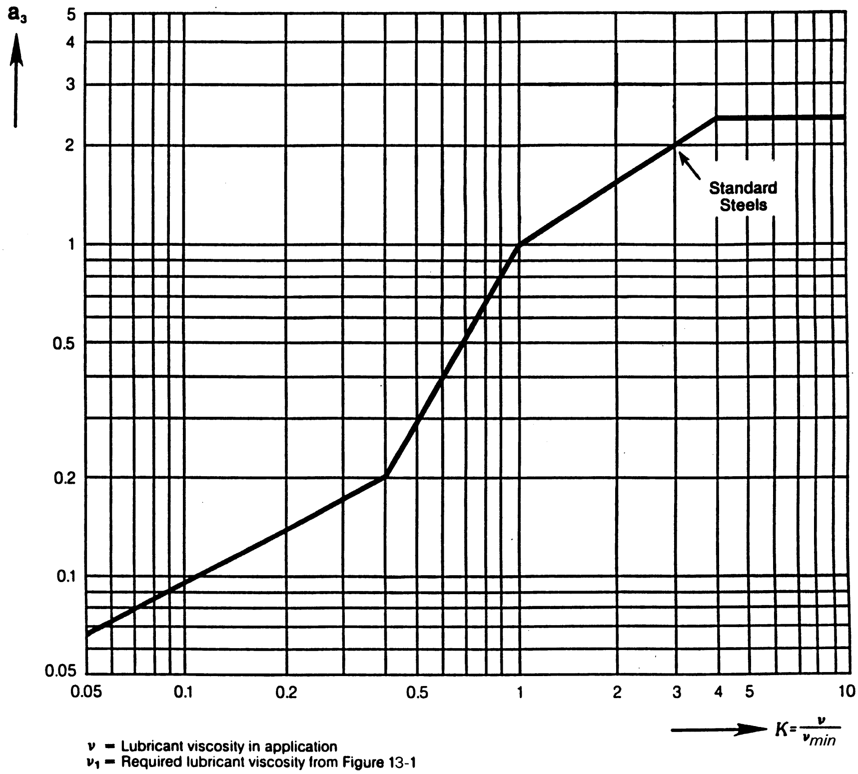


FIG. 13-3 Fatigue life adjustment factor for lubrication (from SKF, 1992, with permission).

Here, ν is the actual viscosity of the lubricant (at the operating temperature) and $\nu_{\min}$ is the minimum required lubricant viscosity from Fig. 13-1.

According to Fig. 13-3, the life adjustment factor a_3 is an increasing function of the viscosity ratio κ. This means that there is an improvement in fatigue-life due to improvement in EHD lubrication at higher viscosity. However, there is a limit to this improvement. For ν higher than 4, Fig. 13-3 indicates that there is no additional improvement in fatigue life from using higher-viscosity oil. This is because higher viscosity has the adverse effect of higher viscous friction, which in turn results in higher bearing operating temperature.

In conclusion, there is a limit on the benefits obtained from increasing oil viscosity. Moreover, oils with excessively high viscosity introduce a higher operating temperature and in turn a higher thermal expansion of the inner ring.

This results in extra rolling contact stresses, which counteract any other benefits obtained from using high-viscosity oil.

The fatigue life adjustment factor a_3 in Fig. 13-3 is often used as $a_{23} = a_2 a_3$. This is because experience indicated that there is no significant improvement in fatigue life due to better bearing steel if there is inadequate lubrication.

Example Problem 13-2

Calculation of Adjusted Fatigue Life

Find the life adjustment factor and adjusted fatigue life of a deep-groove ball bearing. The bearing operates in a gearbox supporting a 25-mm shaft. The bearing is designed for 90% reliability. The shaft speed is 3600 RPM, and the gearbox is designed to transmit a maximum power of 10 kW. The lubricant is SAE 20 oil, and the maximum expected surrounding (ambient) temperature is 30°C. One helical gear is mounted on the shaft at equal distance from both bearings. The rolling bearing data is from the manufacturer's catalog:

Designation bearing:	No. 61805
Bore diameter:	$d = 25$ mm
Outside diameter:	$D = 37$ mm
Dynamic load rating:	$C = 4360$ N
Static load rating:	$C_0 = 2600$ N

The gear data is

Helix angle $\psi = 30^\circ$
Pressure angle (in a cross section normal to the gear) $\phi = 20^\circ$
Diameter of pitch circle $= 5$ in.

Solution

Calculation of Radial and Thrust Forces Acting on Bearing: Given:

Power transmitted by gear:	$\dot{E} = 10\,\text{kW} = 10^4\,\text{N-m/s}$
Rotational speed of shaft:	$N = 3600$ RPM
Helix angle:	$\psi = 30^\circ$
Pressure angle:	$\phi = 20^\circ$
Pitch circle diameter of gear:	$d_p = 5\,\text{in.} = 0.127\,\text{m}$

The angular velocity of the shaft, ω, is

$$\omega = \frac{2\pi N}{60} = \frac{2\pi 3600}{60} = 377\ \text{rad/s}$$

Torque produced by the gear is

$$T = \frac{F_t d_p}{2}$$

Substituting this into the power equation, $\dot{E} = T\omega$, yields

$$\dot{E} = \frac{F_t d_p}{2}\ \omega$$

Solving for the tangential force, F_t, results in

$$F_t = \frac{2\dot{E}}{d_p\omega} = \frac{2 \times 10{,}000 \text{ N-m/s}}{0.127 \text{ m} \times 377 \text{ rad/s}} = 418 \text{ N}$$

Once the tangential component of the force is solved, the radial force F_r, and the thrust load (axial force), F_a, can be calculated, as follows:

$$F_a = F_t\ \tan\psi$$

$$F_a = 417 \text{ N} \times \tan 30^\circ = 241 \text{ N}$$

$$F_r = F_t\ \tan\phi$$

$$F_r = 418 \text{ N} \times \tan 20^\circ$$

$$F_r = 152 \text{ N}$$

The force components F_t and F_r are both in the direction normal to the shaft centerline. The bearing force reacting to these two gear force components, W_r, is the radial force component of the bearing. The gear is in the center, and the bearing radial force is divided between the two bearings. The resultant, W_r, for each bearing is calculated by the equation

$$2W_r = \sqrt{F_t^2 + F_r^2} = \sqrt{418^2 + 152^2} = 445 \text{ N}$$

The resultant force of the gear is supported by the two bearings. It is a radial bearing reaction force, because it is acting in the direction normal to the shaft centerline. Since the helical gear is mounted on the shaft at equal distance from each bearing, each bearing will support half of the radial load:

$$W_r = \frac{445 \text{ N}}{2} = 222.25 \text{ N}$$

However, the thrust load will act on one bearing only. The direction of the thrust load depends on the gear configuration and the direction of rotation. Therefore, each bearing should be designed to support the entire thrust load:

$$F_a = 241 \text{ N}$$

Calculation of Adjusted Fatigue Life of Rolling Bearing. In this example, combined radial and thrust loads are acting on a bearing. In all cases of combined load, it is necessary to determine the equivalent radial load, P, from Eq. (13-7). The radial and thrust load factors X and Y in the following table are available in manufacturers manuals. The values of X and Y differ for different bearings. Table 13-7 includes the factors X and Y of a deep-groove ball bearing.

For $F_a/F_r > e$, the values are

F_a/C_o	e	X	Y
0.025	0.22	0.56	2
0.04	0.24	0.56	1.8
0.07	0.27	0.56	1.6
0.13	0.31	0.56	1.4
0.25	0.37	0.56	1.2
0.5	0.44	0.56	1

The ratio of the axial load, F_a, and the basic static load rating C_0 must be calculated:

$$\frac{F_a}{C_0} = \frac{241.17}{2600} = 0.093$$

Then, by interpolation, the values for e, X, and Y can be determined:

$$e = 0.29 \qquad X = 0.56 \qquad Y = 1.5$$

Also, the ratio of F_a to F_r is

$$\frac{F_a}{F_r} = \frac{241}{222} = 1.09 \quad 1.09 > e$$

Therefore

$$P = XF_r + YF_a$$

$$P = (0.56)(222.2) + (1.5)(241.17)$$

$$P = 486 \text{ N}$$

Using the bearing life equation, the bearing life is determined from the equation

$$L_{10} = \left(\frac{C}{P}\right)^{p} \times 10^{6}$$

$$L_{10} = \left(\frac{4360}{486}\right)^{3} \times 10^{6}$$

$$L_{10} = 722 \times 10^{6} \text{ (revolutions)} = \frac{722 \times 10^{6} \text{ rev}}{3600 \text{ rev/min} \times 60 \text{ min/hr}} = 3343 \text{ hr}$$

This is the fatigue life without adjustment for lubrication. Following is the selection of the minimum required viscosity and the adjustment for the bearing fatigue life when operating with lubricant SAE 20. There is improvement in the fatigue life when the lubricant is of higher viscosity than the minimum required viscosity.

Selection of Oil. The selection of an appropriate oil is an important part of bearing design. The most important property is the oil's viscosity, which is inversely related to temperature. The minimum required viscosity is determined according to the size and rotational speed of the bearing. The bearing size is determined by taking the average of the inner (bearing bore) and outer diameters of the bearing.

The pitch diameter of the bearing is

$$d_m = \frac{d + D}{2}$$

$$d_m = \frac{25 + 37}{2}$$

$$d_m = 31 \text{ mm}$$

From Fig 13-1, at a speed of 3600 RPM and a pitch diameter of 31 mm, the minimum required viscosity is 14 mm^2/s.

As discussed earlier, the temperature of the oil of an operating bearing is usually 3°–11°C above the housing temperature. In this problem, the maximum expected surrounding (ambient) temperature is 30°C, and it is assumed that 5°C should be added for the maximum operating oil temperature. From Fig. 13-2 (viscosity–temperature charts), the viscosity of SAE 20 (VG 46) oil at 35°C is approximately 52 mm^2/s. The viscosity ratio, κ, is

$$\kappa = \frac{\nu}{\nu_{\min}} = \frac{52}{14} = 3.7$$

For 90% reliability, the life adjustment factor a_1 is given a value of 1. The life adjustment factor a_2 for material is also given a value of 1 (standard material).

Based on the viscosity ratio, the operating conditions factor, a_3, can be obtained using Fig. 13-3: $a_3 = 2.2$.

The adjusted rating life is:

$$L_{10a} = a_1 a_2 a_3 (L_{10}), \qquad \text{where } a_1 a_2 = 1$$

$$L_{10a} = 2.2 \times 3343 \text{ hr} = 7354 \text{ hr}$$

Discussion. The fatigue life of an industrial gearbox must be at least five years. If we assume operation of eight hours per day, the minimum fatigue life must be for 14,400 hours. In this case, the tested bearing has an adjusted life of only 7354 hr. The conclusion is that the bearing tested in this example is not a suitable selection for use in an industrial gearbox. The adjusted life is much shorter than required. A more appropriate bearing, therefore, should be selected, of higher dynamic load rating C, and the foregoing procedure should be repeated to verify that the selection is adequate.

13.5 BEARING PRECISION

Manufacturing tolerances specify that the actual dimensions of a bearing be within specified limits. For precision applications, such as precise machine tools and precision instruments, ultrahigh-precision rolling bearings are available with very narrow tolerances. In high-speed machinery, it is important to reduce vibrations, and high-precision bearings are often used. In precision applications and high-speed machinery, it is essential that the center of a rotating shaft remain at the same place, with minimal radial displacement during rotation. During rotation under steady load, any variable eccentricity between the shaft center and the center of rotation is referred to as *radial run-out*. At the same time, any axial displacement of the shaft during its rotation is referred to as *axial run-out*.

If the shaft is precise and centered, the radial and axial run-outs depend on manufacturing tolerances of the rolling-element bearing. Radial run-out depends on errors such as eccentricity between the inside and outside diameters of the rings, deviation from roundness of the races, and deviations in the actual diameters of the rolling elements. For running precision, it is necessary to distinguish between run-out of the inner ring and that of the outer ring, which are not necessarily equal.

There are many applications where different levels of precision of run-out and dimensions are required, such as in machine tools of various precision levels. Of course, higher precision involves higher cost, and engineers must not specify higher precision than really required. The Annular Bearing Engineering Committee (ABEC) introduced five precision grades (ABEC 1, 3, 5, 7, and 9). Each precision grade has an increasing grade of smaller tolerance range of all bearing dimensions. ABEC 1 is the standard bearing and has the lowest cost; it has about

80% of the bearing market share. Bearings of ABEC 3 and 5 precision have very low market share. Bearings of ABEC 7 and 9 precision are for ultraprecision applications. The American Bearing Manufactures Association (ABMA) has adopted this standard for bearing tolerances, ANSI/ABMA-20, 1996, which is accepted as the international standard.

The most important characteristics of precise bearings are the inner ring and outer ring run-outs. However, tolerances of all dimensions are more precise, such as inside and outside diameters, and width. All bearing manufacturers produce standard bearings that conform to these standard dimensions and tolerances.

13.5.1 Inner Ring Run-Out

The inner ring run-out of a rolling bearing is measured by holding the outer ring stationary by means of a fixture and turning the inner ring under steady load. The radial inner ring run-out is measured via an indicator normal to the inner ring surface (inner ring bore). The axial inner ring run-out is measured by an indicator in contact with the face of the inner ring, in a direction normal to the face of the inner ring (parallel to the bearing centerline). In both cases, the run-out is the difference between the maximum and minimum indicator readings.

In machine tools where the shaft is turning, such as in a lathe, the inner ring radial run-out is measured by turning a very precise shaft between two centers, under steady load. A precise dial indicator is fixed normal to the shaft surface. The shaft rotates slowly, and the radial run-out is the difference between the maximum and minimum indicator readings. The axial run-out is measured by an indicator normal to the face of the shaft.

13.5.2 Outer Ring Run-Out

In a similar way, an indicator measures the outer ring run-out. But in that case, the inner ring is constrained by a fixture and the run-out is measured when the outer ring is rotating. In the two cases, a small load, or gravity, is applied to cancel the internal clearance. In this way, the clearance does not affect the run-out measurement, because the run-out depends only on the precision of the bearing parts. For example, an eccentricity between the bore of the inner ring and its raceway will result in a constant inner-ring radial run-out but will not contribute to any outer ring radial run-out.

In precision applications, such as machine tools, there is a requirement for bearings with very low levels of run-out. In addition, there is a requirement for low run-out for high-speed rotors, where radial run-out would result in imbalance and excessive vibrations.

For machine tools, it is important to understand the effect of various types of run-out on the precision of the workpiece. Also, it is necessary to distinguish between a bearing where the inner ring is rotating, such as an electric motor, and a bearing where the outer ring is rotating, such as in car wheels. In the case of an electric motor, the inner ring radial run-out will cause radial run-out of the rotor centerline. If, instead, there is only outer ring radial run-out, there would be no influence on the rotor, because the rotor continues to operate with a new, steady center of rotation (although not concentric with the bearing outer ring).

The opposite applies to a car wheel, where only the outer ring radial run-out is causing run-out of the wheel, while the radial inner ring run-out does not affect the running of the car wheel. In machine tools the precision is measured by the axial and radial run-out of a spindle. However, it is necessary to distinguish between machinery where the workpiece is rotating and where the cutting tool is rotating.

It is necessary to distinguish between steady and time-variable run-out. Steady radial run-out is where the spindle axis has a constant run-out (resulting from eccentricity between the inner ring bore and inner ring raceway). If the workpiece is turning, a steady radial run-out does not result in machining errors, because the workpiece forms its own, new center of rotation, and it will not result in a deviation from roundness. But the workpiece must not be reset during machining, because the center would be relocated. An example is a lathe where the bearing inner ring is rotating together with the spindle and workpiece while the outer ring is stationary. In this configuration, if the spindle has a constant radial displacement, the cutting tool will form a round shape with a new center of rotation without any deviation from roundness. However, any deviation from roundness of the two races, in the form of waviness or elliptical shape, will result in a similar deviation from roundness in the workpiece.

In contrast to a lathe, in a milling machine the cutting tool is rotating while the workpiece is stationary. In this case, operating the rotating tool with a constant eccentricity can result in manufacturing errors. This is evident when trying to machine a planar surface: A wavy surface would be produced, with the wave level depending on the cutting tool run-out (running eccentricity). Such manufacturing error can be minimized by a very slow feed rate of the workpiece.

The load affects the concentric running of a shaft, due to elastic deformations in the contact between rolling elements and raceways. During operation, radial and axial run-outs in rolling bearings are caused not only by deviations from the ideal dimensions, but also by elastic deformation in the bearing—whenever there are rotating forces on the bearing. Most of the elastic deformation is at the contact between the rolling elements and the raceways. Roller-element bearings, such as cylindrical roller bearings, have less elastic deformation than ball bearings.

By designing an adjustable arrangement of two opposing angular ball bearings or tapered bearings, it is possible to eliminate clearance and introduce preload in the bearings (negative clearance). In this way, the bearings are stiffened and the run-out due to elastic deformation or clearance is significantly reduced.

13.6 INTERNAL CLEARANCE OF ROLLING BEARINGS

Rolling bearings are manufactured with internal clearance. The internal clearance is between the rolling elements and the inner and outer raceways. In the absence of clearance, the rolling elements will fit precisely into the space between the raceways of the outer and inner rings. However, in practice this space is always a little larger than the diameter of the rolling elements, resulting in a small clearance. The radial and axial clearances are measured by the displacements in the radial and axial directions that one ring can have relative to the other ring.

The purpose of the clearance is to prevent excessive rolling contact stresses due to uneven thermal expansion of the inner and outer rings. In addition, the clearance prevents excessive rolling contact stresses due to tight-fit assembly of the rings into their seats. During operation, the temperature of the inner ring is usually higher than that of the outer ring, resulting in uneven thermal expansion. In addition, for most bearings the inner and outer rings are tightly fitted into their seats. Tight fit involves elastic deformation of the rings that can cause *negative clearance* (bearing preload), which results in undesired extra contact stresses between the raceways and the rolling elements. The extra stresses can be prevented if the bearing is manufactured with sufficient internal clearance.

In the case of negative clearance, the uneven thermal expansion and elastic deformation due to tight-fit mounting are combined with the bearing load to cause excessive rolling contact stresses. It can produce a chain reaction where the high stresses result in higher friction and additional thermal expansion, which in turn can eventually lead to bearing seizure. Therefore, in most cases bearing manufacturers provide internal clearance to prevent bearing seizure due to excessive contact stresses.

Catalogues of rolling-element bearings specify several standard classes of bearing clearance. This specification of internal bearing clearance is based on the ABMA standard. The specification is from tight, ABMA Class 2, to extra-loose clearance, ABMA Class 5.

Standard bearings have five classes of precision. Table 13-9 shows the classes for increasing levels of internal clearance: C2, Normal, C3, C4, and C5. C2 has the lowest clearance, while C5 has the highest clearance. It should be noted that the normal class is between C2 and C3. The clearance in each class increases with bearing size. In addition, there is a tolerance range for the clearance in each class and bearing size.

After the selection of bearing type and size has been completed, the selection of appropriate internal clearance is the most important design decision. Appropriate internal clearance is important for successful bearing operation. Internal clearance can be measured by displacement of the inner ring relative to the outer ring. This displacement can be divided into radial and axial components. The clearance is selected according to the bearing type and diameter as well as the level of precision required in operation.

TABLE 13-9 Classes of Radial Clearance in Deep Groove Ball Bearing

Normal bore diameter d (mm)		Radial clearance μm									
		C2		Normal		C3		C4		C5	
Over	incl.	min.	max.	min.	max.	min.	max.	min.	max.	min.	max.
d < 10		0	7	2	13	8	23	14	29	20	37
10	18	0	9	3	18	11	25	18	33	25	45
18	24	0	10	5	20	13	28	20	36	28	48
24	30	1	11	5	20	13	28	23	41	30	53
30	40	1	11	6	20	15	33	28	46	40	64
40	50	1	11	6	23	18	36	30	51	45	73
50	65	1	15	8	28	23	43	38	61	55	90
65	80	1	15	10	30	25	51	46	71	65	105
80	100	1	18	12	36	30	58	53	84	75	120
100	120	2	20	15	41	36	66	61	97	90	140
120	140	2	23	18	48	41	81	71	114	105	160
140	160	2	23	18	53	46	91	81	130	120	180
160	180	2	25	20	61	53	102	91	147	135	200
180	200	2	30	25	71	63	117	107	163	150	230
200	225	2	35	25	85	75	140	125	195	175	265
225	250	2	40	30	95	85	160	145	225	205	300
250	280	2	45	35	105	90	170	155	245	225	340
280	315	2	55	40	115	100	190	175	270	245	370
315	335	3	60	45	125	110	210	195	300	275	410
355	400	3	70	55	145	130	240	225	340	315	460
400	450	3	80	60	170	150	270	250	380	350	510
450	500	3	90	70	190	170	300	280	420	390	570
500	550	10	100	80	210	190	330	310	470	440	630
560	630	10	110	90	230	210	360	340	520	490	690
630	710	20	130	110	260	240	400	380	570	540	760
710	800	20	140	120	290	270	450	430	630	600	840

The operating clearance is usually lower than the bearing clearance before its assembly. The clearance is reduced by tight fit and thermal expansion during operation. Much care should be taken during design to secure an appropriate operating clearance. Example Problem 13-3 presents a calculation for predicting the operating clearance.

Low operating clearance is very important in precision machines and high-speed machines. Operating clearance affects the concentric running of a shaft, it causes radial and axial run-out, and reduction of precision. Therefore, elimination of clearance is particularly important for precision machinery, such as machine tools and measuring machines, and high-speed machines, such as turbines, where radial clearance can cause imbalance and vibrations.

Moreover, a small amount of preload (small negative bearing clearance) is desirable in order to stiffen the support of the shaft. This is done in order to reduce the level of vibrations, particularly in high-speed machines or high-precision machines, such as machine tools. In such cases, much care is required in the design, to secure that the preload is not excessive during operation. In certain applications, it is desirable to run the bearing as close as possible to clearance-free operation. Experiments have indicated that clearance-free operation reduces the bearing noise. Clearance-free operation requires accurate design that includes calculations of the housing and shaft tolerances.

In the following sections, it is shown that a widely used method for eliminating the clearance and providing an accurate preload is an adjustable bearing arrangement. In this arrangement, two angular contact ball bearings or tapered roller bearings are mounted in opposite directions on one shaft. This arrangement is designed to allow, during mounting, for one ring to be forced to slide in its seat, in the axial direction, to adjust the bearing clearance or even provide preload inside the bearing.

An adjustable arrangement is not appropriate in all applications. In order to reduce the operating clearance, the designer can select high-precision bearings manufactured with relatively low clearances. Another method often used in precision machines is to incorporate a tapered bore or tapered housing. It is possible to design a tapered-bore bearing, which reduces radial clearance by either expanding elastically the inner ring with a tapered shaft or press-fitting the outer ring with a tapered housing bore.

In many cases, a floating bearing is required to prevent thermal stresses due to thermal elongation of the shaft (thermal expansion in the axial direction). If, at the same time, precise concentric shaft running is required, the designer should not specify a sliding fit (radial clearance) between the bore of the inner ring and the shaft or outer ring and housing. Instead, it is possible to use a cylindrical roller bearing, which can be installed with interference fit to both the shaft and housing and still allow small axial displacements.

13.7 VIBRATIONS AND NOISE IN ROLLING BEARINGS

Theoretically, vibrations can be generated in rolling bearings even if the bearing is manufactured with great precision and its geometry does not significantly deviate from the ideal dimensions. Under external load, the rotation of a rolling element induces periodic cycles of variable elastic deformation, which result in audible noise. In practice, however, vibrations resulting from inaccuracy in dimensions generate most of the noise. Deviation from roundness of the races or rolling elements results in vibrations and noise. Dimensional inaccuracy results in imbalances in the bearing that induce noise.

The noise level of rolling bearings increases with the inaccuracy of the dimensions of the rolling elements and races. It is well known that there are always small deviations from the ideal geometry. Even the most precise manufacturing processes can only reduce dimensional errors, not eliminate them completely. The level of noise increases with the magnitude of the deviation from the nominal dimensions, particularly deviation from roundness of rolling elements or races or uneven diameters of the rolling elements, which can vary within a certain tolerance, depending on the specified tolerances. In addition, the waviness of the surface finish of the raceways and rolling-element surfaces can result in audible noise. In order to reduce the noise, it is important to minimize dimensional errors as well as the magnitude of the surface waviness, which is specified as deviation from roundness. Additional parameters that play a role in the noise level and vibrations include: elastic deformations at the contacts, design and material of the cage, and clearances between the rolling elements and the cage.

A steady radial run-out (resulting from eccentricity of the inner and outer diameters of the rotating ring) generates a vibrational frequency equal to that of the shaft frequency. This frequency is usually too low to cause audible noise, but deviations from roundness, in the form of a few oscillations over the circumference of the race, can generate audible noise. One must keep in mind that a machine is a dynamic system, and vibrations, which originate in the rolling bearing, can excite vibrations and audible noise in other parts of the machine. In the same way, vibrations originating in other parts of the machine can excite audible noise in the rolling bearing. In particular, a high level of vibrations is expected whenever the exciting frequency is close to one of the natural frequencies in the machine.

High-frequency vibrations generate audible noise as well as ultrasonic sound waves. Ultrasonic sound waves can be measured and used for predictive maintenance. The ultrasonic measurements are analyzed and the results used to predict the condition of rolling bearings while the machine is running. A significant change in the level or frequency of ultrasonic sound is an indication of possible damage on the surface of the races or rolling elements. Experiments have indicated that when the bearing runs as close as possible to clearance free, the operating conditions are optimal for minimum noise and vibrations. This can be achieved by proper tight-fit mounting to eliminate bearing clearance; however, thermal expansion must be considered, to avoid excessive preload.

Noise generated by roller bearings is a concern in many applications, particularly in office or hospital machines. Moreover, there is an increasing interest in improving the manufacturing environment and reducing the noise level in all machinery. Bearing manufacturers supply low-noise bearings. These bearings are of high precision and pass quality control tests. The tests include a plot of the actual roundness of rings in polar coordinates, where deviations from theoretical roundness are magnified. In addition, samples are tested in operation, the level of noise is measured and recorded, and the correlation between certain dimensional deviations and noise level can be established. In this way, better low-noise bearings are developed.

In additional to manufacturing precision, low-noise bearings must be mounted properly. Manufacturing errors in the form of deviations from roundness can be transferred by elastic deformation of the thin rings from the housing and shaft seats to the races. Therefore, for low-noise applications, high precision is required for the housing and shaft seats. Also, minimal misalignment is required.

There are many parameters that influence vibrations in bearings, including bearing clearance. To minimize noise, it is desirable to have a clearance-free operating bearing. On the one hand, large clearance generates a characteristic hollow noise. On the other hand, a bearing that is excessively press-fitted (negative clearance) induces another characteristic high-pitch noise. For optimum results, an effort must be made to operate the bearing as closely as possible to a clearance-free condition. But in practice this is difficult to achieve, because the operating clearance is affected by several factors and there are variable conditions during operation. The design engineer must take into account the reduction of clearance after installation due to the interference fits at the bearing seats, and thermal expansion must be considered (usually the operating temperature of the inner ring is higher than that of the outer ring). Example Problem 13-3 covers a radial clearance calculation during operation.

Another method to achieve clearance-free running is to include adjustable arrangements of two angular ball bearings or tapered rolling bearings in opposition. By axial movement of the inner or outer ring, either through the use of a nut, which is tightened once the machine has reached thermal equilibrium, or with the use of spring washers, it is possible to adjust the clearance. The operating temperatures of the inner and outer rings are not always known, and an adjustable arrangement that is adjusted during machine assembly does not ensure clearance-free operation. Therefore, the best results can be achieved by designs that allow for clearance adjustment while the bearing is running and after thermal steady state has been reached.

In addition to precise geometry, noise reduction is often achieved by using special greases with better damping characteristics for noise and vibrations. Grease manufacturers recommend certain greases that have been tested and proved to reduce noise more effectively than other types.

13.8 SHAFT AND HOUSING FITS

For successful bearing operation, care must be taken to specify tolerances of the appropriate fits between the bearing bore and the shaft seat as well as the outer diameter and the housing seat. It is important that the rotating shaft always be tightly fitted into the bearing bore, because a loose fit will damage the bearing bore as well as the shaft seat. The type of fit, such as tight fit or loose fit, depends on the application. Since the tolerances for the bore and outside bearing diameter are standardized, the required fits are obtained by selecting the proper tolerances for the shaft diameter and the housing bore. The fits and tolerances are selected based on the ISO standard, which contains a very large selection of shaft and housing tolerances for any desirable fit, from a tight fit to a loose fit.

For the housing and shaft, the ISO standard provides for various degrees of tightness: very tight, moderately tight, sliding, and loose fit. According to the ISO standard, a letter and a number are used for specifying tolerances; capital letters are used for housing bores, and small letters for shaft diameters. The letter specifies the location of the *tolerance zone* (range between minimum and maximum dimensions) relative to the *nominal dimension*. In fact, the letter determines the degree of clearance or tightness of the housing or shaft in relation to the outside diameter or bore of the bearing. At the same time, the number specifies the size of the *tolerance zone*.

A demonstration of the tolerance zone of the housing and shaft is shown in Fig. 13-4. The tolerance zone of the housing and shaft is relative to the bearing tolerance, for various tolerance grades, from the loosest, G7, to the tightest, P7. Table 13-10 lists the tolerances of the shaft for various nominal diameters; Table 13-11 lists the tolerances of the housing for various nominal diameters. Table 13-12 provides recommendations for shaft tolerances for various applications; Table 13-13 gives similar recommendations for housing tolerances.

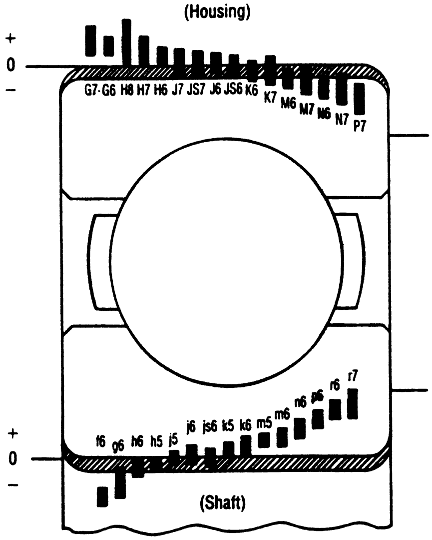
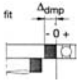
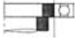
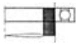
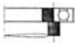
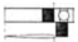
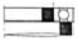
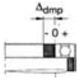
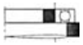
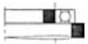
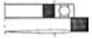
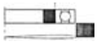


FIG. 13-4 Illustration of tolerance grades (from SKF, 1992, with permission).

TABLE 13-10 Tolerances of the Shaft (from FAG (1999) with permission of FAG and Handel AG)

		Dimensions in mm																									
Nominal shaft dimension	over	**3**		**6**		**10**		**18**		**30**		**50**		**65**		**80**		**100**		**120**		**140**		**160**		**180**	
	to	**6**		**10**		**18**		**30**		**50**		**65**		**80**		**100**		**120**		**140**		**160**		**180**		**200**	
		Tolerance in microns (0.001 mm) (normal tolerance)																									
Bearing bore diameter deviation		0		0		0		0		0		0		0		0		0		0		0		0		0	
	Δ_{dmp}	−8		−8		−8		−10		−12		−15		−15		−20		−20		−25		−25		−25		−30	
Diagram of fit Shaft		Shaft tolerance, interference or clearance in microns (0.001 mm)																									
f6			**2**		**5**		**8**		**10**		**13**		**15**		**15**		**16**		**16**		**18**		**18**		**18**		**20**
		−10	**8**	−13	**11**	−16	**15**	−20	**17**	−25	**22**	−30	**26**	−30	**26**	−36	**30**	−36	**30**	−43	**34**	−43	**34**	−43	**34**	−50	**40**
		−18	**18**	−22	**22**	−27	**27**	−33	**33**	−41	**41**	−49	**49**	−49	**49**	−58	**58**	−58	**58**	−68	**68**	−68	**68**	−68	**68**	−79	**79**
g5			**4**		**3**		**2**		**3**		**3**		**5**		**5**		**8**		**8**		**11**		**11**		**11**		**15**
		−4	**0**	−5	**2**	−6	**3**	−7	**3**	−9	**5**	−10	**4**	−10	**4**	−12	**4**	−12	**4**	−14	**3**	−14	**3**	−14	**3**	−15	**2**
		−9	**9**	−11	**11**	−14	**14**	−16	**16**	−20	**20**	−23	**23**	−23	**23**	−27	**27**	−27	**27**	−32	**32**	−32	**32**	−32	**32**	−35	**35**
g6			**4**		**3**		**2**		**3**		**3**		**5**		**5**		**8**		**8**		**11**		**11**		**11**		**15**
		−4	**1**	−5	**3**	−6	**4**	−7	**5**	−9	**6**	−10	**6**	−10	**6**	−12	**6**	−12	**6**	−14	**6**	−14	**6**	−14	**6**	−15	**5**
		−12	**12**	−14	**14**	−17	**17**	−20	**20**	−25	**25**	−29	**29**	−29	**29**	−34	**34**	−34	**34**	−39	**39**	−39	**39**	−39	**39**	−44	**44**
h5			**8**		**8**		**8**		**10**		**12**		**15**		**15**		**20**		**20**		**25**		**25**		**25**		**30**
		0	**4**	0	**3**	0	**3**	0	**4**	0	**4**	0	**6**	0	**6**	0	**8**	0	**8**	0	**11**	0	**11**	0	**11**	0	**13**
		−5	**5**	−6	**6**	−8	**8**	−9	**9**	−11	**11**	−13	**13**	−13	**13**	−15	**15**	−15	**15**	−18	**18**	−18	**18**	−18	**18**	−20	**20**
h6			**8**		**8**		**8**		**10**		**12**		**15**		**15**		**20**		**20**		**25**		**25**		**25**		**30**
		0	**3**	0	**2**	0	**2**	0	**2**	0	**3**	0	**4**	0	**4**	0	**6**	0	**6**	0	**8**	0	**8**	0	**8**	0	**10**
		−8	**8**	−9	**9**	−11	**11**	−13	**13**	−16	**16**	−19	**19**	−19	**19**	−22	**22**	−22	**22**	−25	**25**	−25	**25**	−25	**25**	−29	**29**
j5			**11**		**12**		**13**		**15**		**18**		**21**		**21**		**26**		**26**		**32**		**32**		**32**		**37**
		+3	**7**	+4	**7**	+5	**8**	+5	**9**	+6	**10**	+6	**12**	+6	**12**	+6	**14**	+6	**14**	+7	**18**	+7	**18**	+7	**18**	+7	**20**
		−2	**2**	−2	**2**	−3	**3**	−4	**4**	−5	**5**	−7	**7**	−7	**7**	−9	**9**	−9	**9**	−11	**11**	−11	**11**	−11	**11**	−13	**13**
j6			**14**		**15**		**16**		**19**		**23**		**27**		**27**		**33**		**33**		**39**		**39**		**39**		**46**
		+6	**8**	+7	**9**	+8	**10**	+9	**11**	+11	**14**	+12	**16**	+12	**16**	+13	**19**	+13	**19**	+14	**22**	+14	**22**	+14	**22**	+16	**26**
		−2	2	−2	2	−3	3	−4	4	−5	5	−7	7	−7	7	−9	9	−9	9	−11	11	−11	11	−11	11	−13	13
js5			**11**		**11**		**12**		**15**		**18**		**22**		**22**		**28**		**28**		**34**		**34**		**34**		**40**
		+2.5	**6**	+3	**6**	+4	**6**	+4.5	**9**	+5.5	**10**	+6.5	**13**	+6.5	**13**	+7.5	**16**	+7.5	**16**	+9	**20**	+9	**20**	+9	**20**	+10	**23**
		−2.5	3	−3	3	−4	4	−4.5	5	−5.5	6	−6.5	7	−6.5	7	−7.5	8	−7.5	8	−9	9	−9	9	−9	9	−10	10
js6			**12**		**13**		**14**		**17**		**20**		**25**		**25**		**31**		**31**		**38**		**38**		**38**		**45**
		+4	**7**	+4.5	**7**	+5.5	**8**	+6.5	**9**	+8	**11**	+9.5	**13**	+9.5	**13**	+11	**17**	+11	**17**	+12.5	**21**	+12.5	**21**	+12.5	**21**	+14.5	**25**
		−4	4	−4.5	5	−5.5	6	−6.5	7	−8	8	−9.5	10	−9.5	10	−11	11	−11	11	−12.5	13	−12.5	13	−12.5	13	−14.5	15
k5			**14**		**15**		**17**		**21**		**25**		**30**		**30**		**38**		**38**		**46**		**46**		**46**		**54**
		+6	**9**	+7	**10**	+9	**12**	+11	**15**	+13	**17**	+15	**21**	+15	**21**	+18	**26**	+18	**26**	+21	**32**	+21	**32**	+21	**32**	+24	**37**
		+1	**1**	+1	**1**	+1	**1**	+2	**2**	+2	**2**	+2	**2**	+2	**2**	+3	**3**	+3	**3**	+3	**3**	+3	**3**	+3	**3**	+4	**4**

k6 		**17**		**18**		**20**		**25**		**30**		**36**		**36**		**45**		**45**		**53**		**53**		**53**		**63**
	+9	**11**	+10	**12**	+12	**14**	+15	**17**	+18	**21**	+21	**25**	+21	**25**	+25	**31**	+25	**31**	+28	**36**	+28	**36**	+28	**36**	+33	**43**
	+1	**1**	+1	**1**	+1	**1**	+2	**2**	+2	**2**	+2	**2**	+2	**2**	+3	**3**	+3	**3**	+3	**3**	+3	**3**	+3	**3**	+4	**4**
m5 		**17**		**20**		**23**		**27**		**32**		**39**		**39**		**48**		**48**		**58**		**58**		**58**		**67**
	+9	**13**	+12	**15**	+15	**18**	+17	**21**	+20	**24**	+24	**30**	+24	**30**	+28	**36**	+28	**36**	+33	**44**	+33	**44**	+33	**44**	+37	**50**
	+4	**4**	+6	**6**	+7	**7**	+8	**8**	+9	**9**	+11	**11**	+11	**11**	+13	**13**	+13	**13**	+15	**15**	+15	**15**	+15	**15**	+17	**17**
m6 		**20**		**23**		**26**		**31**		**37**		**45**		**45**		**55**		**55**		**65**		**65**		**65**		**76**
	+12	**15**	+15	**17**	+18	**20**	+21	**23**	+25	**27**	+30	**34**	+30	**34**	+35	**42**	+35	**42**	+40	**48**	+40	**48**	+40	**48**	+46	**56**
	+4	**4**	+6	**6**	+7	**7**	+8	**8**	+9	**9**	+11	**11**	+11	**11**	+13	**13**	+13	**13**	+15	**15**	+15	**15**	+15	**15**	+17	**17**
n5 		**21**		**24**		**28**		**34**		**40**		**48**		**48**		**58**		**58**		**70**		**70**		**70**		**81**
	+13	**17**	+16	**19**	+20	**23**	+24	**28**	+28	**32**	+33	**39**	+33	**39**	+38	**46**	+38	**46**	+45	**56**	+45	**56**	+45	**56**	+51	**64**
	+8	**8**	+10	**10**	+12	**12**	+15	**15**	+17	**17**	+20	**20**	+20	**20**	+23	**23**	+23	**23**	+27	**27**	+27	**27**	+27	**27**	+31	**31**
n6 		**24**		**27**		**31**		**38**		**45**		**54**		**54**		**65**	**65**	**65**		**77**		**77**		**77**		**90**
	+16	**19**	+19	**21**	+23	**25**	+28	**30**	+33	**36**	+39	**43**	+39	**43**	+45	**51**	+45	**51**	+52	**60**	+52	**60**	+52	**60**	+60	**70**
	+8	**8**	+10	**10**	+12	**12**	+15	**15**	+17	**17**	+20	**20**	+20	**20**	+23	**23**	+23	**23**	+27	**27**	+27	**27**	+27	**27**	+31	**31**
p6 		**28**		**32**		**37**		**45**		**54**		**66**		**66**		**79**		**79**		**93**		**93**		**93**		**109**
	+20	**23**	+24	**26**	+29	**31**	+35	**37**	+42	**45**	+51	**55**	+51	**55**	+59	**65**	+59	**65**	+68	**76**	+68	**76**	+68	**76**	+79	**89**
	+12	**12**	+15	**15**	+18	**18**	+22	**22**	+26	**26**	+32	**32**	+32	**32**	+37	**37**	+37	**37**	+43	**43**	+43	**43**	+43	**43**	+50	**50**
p7 		**32**		**38**		**44**		**53**		**63**		**77**		**77**		**92**		**92**		**108**		**108**		**108**		**126**
	+24	**25**	+30	**30**	+36	**35**	+43	**43**	+51	**51**	+62	**62**	+62	**62**	+72	**73**	+72	**73**	+83	**87**	+83	**87**	+83	**87**	+96	**101**
	+12	**12**	+15	**15**	+18	**18**	+22	**22**	+26	**26**	+32	**32**	+32	**32**	+37	**37**	+37	**37**	+43	**43**	+43	**43**	+43	**43**	+50	**50**
r6 		**31**		**36**		**42**		**51**		**62**		**75**		**77**		**93**		**96**		**113**		**115**		**118**		**136**
	+23	**25**	+28	**30**	+34	**35**	+41	**44**	+50	**53**	+60	**64**	+62	**66**	+73	**79**	+76	**82**	+88	**97**	+90	**99**	+93	**102**	+106	**116**
	+15	**15**	+19	**19**	+23	**23**	+28	**28**	+34	**34**	+41	**41**	+43	**43**	+51	**51**	+54	**54**	+63	**63**	+65	**65**	+68	**68**	+77	**77**
r7 		**35**		**42**		**49**		**59**		**71**		**86**		**88**		**106**		**109**		**128**		**130**		**133**		**153**
	+27	**28**	+34	**34**	+41	**40**	+49	**49**	+59	**59**	+71	**71**	+73	**73**	+86	**87**	+89	**90**	+103	**107**	+105	**109**	+108	**112**	+123	**128**
	+15	**15**	+19	**19**	+23	**23**	+28	**28**	+34	**34**	+41	**41**	+43	**43**	+51	**51**	+54	**54**	+63	**63**	+65	**65**	+68	**68**	+77	**77**

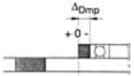
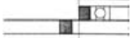
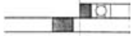
Example: Shaft dia 40 j5

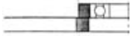
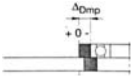
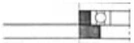
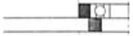
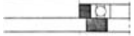
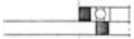
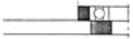
Maximum material	+6	**18**	Interference or clearance when upper shaft deviations coincide with lower bore deviations
		10	Probable interference or clearance
Minimum material	−5	5	Interference or clearance when lower shaft deviations coincide with upper bore deviations

---

Numbers in boldface print identify interference.
Standard-type numbers in right column identify clearance.

TABLE 13-11 Tolerances of the Housing (from FAG (1999) with permission of FAG OEM and Handel AG)

					Dimensions in mm						Dimensions in mm																		
Nominal housing bore		over	6		10		18		30		50		80		120		150		180		250		315		400		500		
		to	10		18		30		50		80		120		150		180		250		315		400		500		630		
			Tolerance in microns (0.001mm) (normal tolerance)												Tolerance in microns (0.001 mm) (normal tolerance)														
Bearing outside diameter deviation			0		0		0		0		0		0		0		0		0		0		0		0		0		
		Δ_{dmp}	−8		−8		−9		−11		−13		−15		−18		−25		−30		−35		−40		−45		−50		
Diagram of fit Housing		Δ_{Dmp} +0−	Housing tolerance, interference or clearance in microns (0.001 mm)														Housing tolerance, interference or clearance in microns (0.001 mm)												
E8				**25**		**32**		**40**		**50**		**60**		**72**		**85**		**85**		**100**		**110**		**125**		**135**		**145**	
			+47	35	+59	44	+73	54	+89	67	+106	79	+126	85	+148	112	+148	114	+172	134	+191	149	+214	168	+232	182	+255	199	+
			+25	55	+32	67	+40	82	+50	100	+60	119	+72	141	+85	166	+85	173	+100	202	+110	226	+125	254	+135	277	+145	305	+
F7				**13**		**16**		**20**		**25**		**30**		**36**		**43**		**43**		**50**		**56**		**62**		**68**		**76**	
			+28	21	+34	25	+41	30	+50	37	+60	44	+71	53	+83	62	+83	64	+96	75	+108	85	+119	94	+131	104	+146	116	+
			+13	36	+16	42	+20	50	+25	61	+30	73	+36	86	+43	101	+43	108	+50	126	+56	143	+62	159	+68	176	+76	196	+
G6				**5**		**6**		**7**		**9**		**10**		**12**		**14**		**14**		**15**		**17**		**18**		**20**		**22**	
			+14	11	+17	12	+20	14	+25	18	+29	21	+34	24	+39	28	+39	31	+44	35	+49	39	+54	43	+60	48	+66	54	+
			+5	**22**	+6	**25**	+7	**29**	+9	**36**	+10	**42**	+12	**49**	+14	**57**	+14	**64**	+15	**74**	+17	**84**	+18	**94**	+20	**105**	+22	**116**	+
G7				**5**		**6**		**7**		**9**		**10**		**12**		**14**		**14**		**15**		**17**		**18**		**20**		**22**	
			+20	**13**	+24	**15**	+28	**17**	+34	**21**	+40	**24**	+47	**29**	+54	**33**	+54	**36**	+61	**40**	+69	**46**	+75	**50**	+83	**56**	+92	**62**	+
			+5	**28**	+6	**32**	+7	**37**	+9	**45**	+10	**53**	+12	**62**	+14	**72**	+14	**79**	+15	**91**	+17	**104**	+18	**115**	+20	**128**	+22	**142**	+
H6				**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**	
			+9	**6**	+11	**6**	+13	**7**	+16	**9**	+19	**11**	+22	**12**	+25	**14**	+25	**17**	+29	**20**	+32	**22**	+36	**25**	+40	**28**	+44	**32**	+
			0	**17**	0	**19**	0	**22**	0	**27**	0	**32**	0	**37**	0	**43**	0	**50**	0	**59**	0	**67**	0	**76**	0	**85**	0	**94**	
H7				**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**	
			+15	**8**	+18	**9**	+21	**10**	+25	**12**	+30	**14**	+35	**17**	+40	**19**	+40	**22**	+46	**25**	+52	**29**	+57	**32**	+63	36	+70	**40**	
			0	**23**	0	**26**	0	**30**	0	**36**	0	**43**	0	**50**	0	**58**	0	**65**	0	**76**	0	**87**	0	**97**	0	**108**	0	**120**	0
H8				**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**		**0**	
			+22	**10**	+27	**12**	+33	**14**	+39	**17**	+46	**20**	+54	**23**	+63	**27**	+63	**29**	+72	**34**	+81	**39**	+89	**43**	+97	**47**	+110	**54**	+
			0	**30**	0	**35**	0	**42**	0	**50**	0	**59**	0	**69**	0	**81**	0	**88**	0	**102**	0	**116**	0	**129**	0	**142**	0	**160**	0
J6				**4**		**5**		**5**		**6**		**6**		**6**		**7**		**7**		**7**		**7**		**7**		**7**			
			+5	**2**	+6	**1**	+8	**2**	+10	**3**	+13	**5**	+16	**6**	+18	**7**	+18	**10**	+22	**13**	+25	**15**	+29	**18**	+33	**21**			
			−4	**13**	−5	**14**	−5	**17**	−6	**21**	−6	**26**	−6	**31**	−7	**36**	−7	**43**	−7	**52**	−7	**60**	−7	**69**	−7	**78**			
J7				**7**		**8**		**9**		**11**		**12**		**13**		**14**		**14**		**16**		**16**		**18**		**20**			
			+8	**1**	+10	**1**	+12	**1**	+14	**1**	+18	**2**	+22	**4**	+26	**5**	+26	**8**	+30	**9**	+36	**13**	+39	**14**	+43	**16**			
			−7	**16**	−8	**18**	−9	**21**	−11	**25**	−12	**31**	−13	**37**	−14	**44**	−14	**51**	−16	**60**	−16	**71**	−18	**79**	−20	**88**			
JS6				**4.5**		**5.5**		**6.5**		**8**		**9.5**		**11**		**12.5**		**12.5**		**14.5**		**16**		**18**		**20**		**22**	
			+4.5	**2**	+5.5	**1**	+6.5	**0**	+8	**1**	+9.5	**0**	+11	**1**	+12.5	**1**	+12.5	**3**	+14.5	**5**	+16	**7**	+18	**6**	+20	**8**	+22	**10**	

	−4.5	**12.5**	−5.5	**13.5**	−6.5	**15.5**	−8	**19**	−9.5	**22.5**	−11	**26**	−12.5	**30.5**	−12.5	**37.5**	−14.5	**44.5**	−16	**51**	−18	**58**	−20	**65**	−22	**72**	
JS7 		**7.5**		**9**		**10.5**		**12.5**		**15**		**17.5**		**20**		**20**		**23**		**26**		**28.5**		**31.5**		**35**	
	+7.5	**1**	+9	**0**	+10.5	**1**	+12.5	**1**	+15	**1**	17.5	**1**	20	**1**	20	**1**	23	**2**	+26	**3**	+28.5	**3**	31.5	**4**	+35	**5**	
	−7.5	**15.5**	−9	**17**	−10.5	**19.5**	−12.5	**23.5**	−15	**28**	−17.5	**32.5**	−20	**38**	−20	**45**	−23	**53**	−26	**61**	−28.5	**68.5**	−31.5	**76.5**	−35	**85**	
K6 		**7**		**9**		**11**		**13**		**15**		**18**		**21**		**21**		**24**		**27**		**29**		**32**		**44**	
	+2	**1**	+2	**3**	+2	**4**	+3	**4**	+4	**4**	+4	**6**	+4	**7**	+4	**4**	+5	**4**	+5	**5**	+7	**4**	+8	**4**	0	**12**	
	−7	**10**	−9	**10**	−11	**11**	−13	**14**	−15	**17**	−18	**19**	−21	**22**	−21	**29**	−24	**35**	−27	**40**	−29	**47**	−32	**53**	−44	**50**	
K7 		**10**		**12**		**15**		**18**		**21**		**25**		**28**		**28**		**33**		**36**		**40**		**45**		**70**	
	+5	**2**	+6	**3**	+6	**5**	+7	**6**	+9	**7**	+10	**8**	+12	**9**	+12	**6**	+13	**8**	+16	**7**	+17	**8**	+18	**9**	0	**30**	
	−10	**13**	12	**14**	−15	**15**	−18	**18**	−21	**22**	−25	**25**	−28	**30**	−28	**37**	−33	**43**	−36	**51**	−40	**57**	−45	**63**	−70	**50**	
M6 		**12**		**15**		**17**		**20**		**24**		**28**		**33**		**33**		**37**		**41**		**46**		**50**		**70**	
	−3	**6**	−4	**9**	−4	**10**	−4	**11**	−5	**13**	−6	**16**	−8	**19**	−8	**16**	−8	**17**	−9	**19**	−10	**21**	−10	**22**	−26	**38**	−30
	−12	**5**	−15	**4**	−17	**5**	−20	**7**	−24	**8**	−28	**9**	−33	**10**	−33	**17**	−37	**22**	−41	**26**	−46	**30**	−50	**35**	−70	**24**	−80
M7 		**15**		**18**		**21**		**25**		**30**		**35**		**40**		**40**		**46**		**52**		**57**		**63**		**96**	
	0	**7**	0	**9**	0	**11**	0	**13**	0	**16**	0	**18**	0	**21**	0	**18**	0	**21**	0	**23**	0	**25**	0	**27**	−26	**56**	−30
	−15	**8**	−18	**8**	−21	**9**	−25	**11**	−30	**13**	−35	**15**	−40	**18**	−40	**25**	−46	**30**	−52	**35**	−57	**40**	−63	**45**	−96	**24**	−11
N6 		**16**		**20**		**24**		**28**		**33**		**38**		**45**		**45**		**51**		**57**		**62**		**67**		**88**	
	−7	**10**	−9	**14**	−11	**17**	−12	**19**	−14	**22**	−16	**26**	−20	**31**	−20	**28**	−22	**31**	−25	**35**	−26	**37**	−27	**39**	−44	**56**	−50
	−16	**1**	−20	**1**	−24	**2**	−28	**1**	−33	**1**	−38	**1**	45	**2**	−45	**5**	−51	**8**	−57	**10**	−62	**14**	−67	**18**	−88	**6**	−10
N7 		**19**		**23**		**28**		**33**		**39**		**45**		**52**		**52**		**60**		**66**		**73**		**80**		**114**	
	−4	**11**	−5	**14**	−7	**18**	−8	**21**	−9	**25**	−10	**28**	−12	**33**	−12	**30**	−14	**35**	−14	**37**	−16	**41**	−17	**44**	−44	**74**	−50
	−19	**4**	−23	**3**	−28	**2**	−33	**3**	−39	**4**	−45	**5**	−52	**6**	−52	**13**	−60	**16**	−66	**21**	−73	**24**	−80	**28**	−114	**6**	−13
P6 		**21**		**26**		**31**		**37**		**45**		**52**		**61**		**61**		**70**		**79**		**87**		**95**		**122**	
	−12	**15**	−15	**20**	−18	**24**	−21	**28**	−26	**34**	−30	**40**	−36	**47**	−36	**44**	−41	**50**	−47	**57**	−51	**62**	−55	**67**	−122	**28**	−13
	−21	**4**	−26	**7**	−31	**9**	−37	**10**	−45	**13**	−52	**15**	−61	**18**	−61	**11**	−70	**11**	−79	**12**	−87	**11**	−95	**10**	−122	**28**	−13
P7 		**24**		**29**		**35**		**42**		**51**		**59**		**68**		**68**		**79**		**88**		**98**		**108**		**148**	
	−9	**16**	−11	**20**	−14	**25**	−17	**30**	−21	**37**	−24	**42**	−28	**49**	−28	**46**	−33	**54**	−36	**59**	−41	**66**	−45	**72**	−78	**108**	−88
	−24	**1**	−29	**3**	−35	**5**	−42	**6**	−51	**8**	−59	**9**	−68	**10**	−68	**3**	−79	**3**	−88	**1**	−98	**1**	−108	**0**	−148	**28**	−16

Example: Housing bore dia 100 M7

Minimum material	0	**35**	Interference or clearance when upper outside diameter deviations of ring coincide with lower housing bore deviations
		18	Probable interference or clearance
Maximum material	−35	15	Interference or clearance when lower outside diameter deviations of ring coincide with upper housing bore deviations

---

Numbers in **boldface print** identify interference.
Standard-type numbers in right column identify clearance.

TABLE 13-12 Recommendations for Shaft Tolerances Selection of Solid Steel Shaft Tolerance Classification for Metric Radial Ball and Roller Bearings of Tolerance Classes ABEC-1, RBEC-1 (Except Tapered Roller Bearings) (From SKF, 1992, with permission)

Conditions	Examples	Shaft diameter, mm			
		ball bearings[1]	Cylindrical roller bearing	Spherical roller bearings	Tolerance symbol
Rotating inner ring load or direction of loading indeterminate					
Light loads	Conveyors, lightly loaded gearbox bearings	(18) to 100	≤ 40	—	j6
		(100) to 140	(40) to 100	—	k6
Normal loads	Bearing applications generally electric motors turbines, pumps internal combustion engines, gearing woodworking machines	≤ 18	—	—	j5
		(18) to 100	≤ 40	≤ 40	k5 (k6)[2]
		(100) to 140	(40) to 100	(40) to 65	m5 (m6)[2]
		(140) to 200	(100) to 140	(65) to 100	m6
		(200) to 280	(140) to 200	(100) to 140	n6
		—	(200) to 400	(140) to 280	p6
		—	—	(280) to 500	r6
		—	—	> 500	r7
Heavy loads	Axleboxes for heavy railway vehicles, traction motors, rolling mills	—	(50) to 140	(50) to 100	n6[3]
		—	(140) to 200	(100) to 140	p6[3]
		—	> 200	> 140	r6[3]

High demands on running accuracy with light loads	Machine tools	≤ 18	—	—	h5[4]
		(18) to 100	≤ 40	—	j5[4]
		(100) to 200	(40) to 140	—	k5[4]
		—	(140) to 200	—	m5[4]
Stationary inner ring load					
Easy axial displacement of inner ring on shaft desirable	Wheels on non-rotating axles	all	all	all	g6
Easy axial displacement of inner ring on shaft unnecessary	Tension pulleys, rope sheaves	all	all	all	h6
Axial loads only					
	Bearing applications of all kinds	all	all	all	j6

TABLE 13-13 Recommendations for Housing Tolerances (From SKF, 1992, with permission)

Conditions	Examples	Tolerence symbol	Displacement of outer ring
SOLID HOUSINGS			
Rotating outer ring load			
Heavy loads on bearings in thin-walled housings, heavy shock loads	Roller bearing wheel hubs, big-end bearings	P7	Cannot be displaced
Normal loads and heavy loads	Ball bearing wheel hubs, big-end bearings, crane travelling wheels	N7	Cannot be displaced
Light and variable loads	Conveyor rollers, rope sheaves, belt tension pulleys	M7	Cannot be displaced
Direction of load indeterminate			
Heavy shock loads	Electric traction motors	M7	Cannot be displaced
Normal loads and heavy loads axial displacement of outer ring unnecessary	Electric motors, pumps, crankshaft bearings	K7	Cannot be displaced as a rule
Accurate or silent running			
	Roller bearings for machine tool work spindles	K6[1]	Cannot be displaced as a rule
	Ball bearings for grinding spindles, small electric motors	J6[2]	Can be displaced
	Small electric motors	H6	Can easily be displaced

SPLIT OR SOLID HOUSING			
Direction of load indeterminate			
Light loads and normal loads axial displacement of outer ring desirable	Medium-sized electrical machines, pumps, crankshaft bearings	J7	Can be normally displaced
Stationary outer ring load			
Loads of all kinds	Railway axleboxes	H7[3]	Can easily be displaced
Light loads and normal loads with simple working conditions	General engineering	H8	Can easily be displaced
Heat condition through shaft	Drying cylinders, large electrical machines with spherical roller bearings	G7[4]	Can easily be displaced

For a standard rolling bearing, the tolerance zones of the outside and bore diameters are below the nominal diameter. The tolerance zone has two boundaries. One boundary is the nominal dimension, and the second boundary is of lower diameter. The lower boundary, which determines the tolerance zone, depends on the bearing precision and size. For example, for a normal bearing of outside diameter $D = 100\,\text{mm}$, the tolerance zone is $+0$ to $-18\,\mu\text{m}$. This means that the actual outside bearing diameter can be within the tolerance zone of the nominal 100 mm and 18 μm lower than the nominal dimension. In drawings, dimensions with a tolerance are specified in several ways, for example, $100^{+0,-18}$. For a bearing bore diameter $d = 60\,\text{mm}$ of a normal bearing, the tolerance is $+0$, $-15\,\mu\text{m}$. In this case, the actual bore diameter can be between the nominal 60 mm and 15 μm lower than the nominal dimension, $60^{+0,-15}$. In addition, there are various precision classes, from class 2 to class 6, where class 2 is of the highest precision. For comparison with the previous example of a normal precision class, the dimension of class 2 of the outside diameter is $D = 100^{+0,-5}$; for the bore diameter it is $d = 60^{+0,-2.5}$.

As a rule, the rotating ring of a rolling-element bearing is always tightly fitted in its seat. In most machines, the rotating ring is the inner ring, such as in a centrifugal pump, where the bearing bore is mounted by a tight fit on the shaft. In that case, the outer ring can be mounted in the housing with tight fit, or it can be mounted with a loose fit to allow for free thermal expansion of the shaft. However, if the outer ring is rotating, such as in a grinding wheel, the outer ring should be mounted with a tight fit, while the inner ring can be mounted on a stationary shaft with tight or loose fit. Tight fit of the rotating ring is essential for preventing sliding between the ring and its seat during start-up and stopping, when the rotating ring is subjected to high angular acceleration and tends to slide. Sliding of the ring will result in severe wear of the seat, and eventually the ring will be completely loose in its seat.

In the case of a rotating force, such as centrifugal forces in an unbalanced spindle of a lathe, it is important that the two rings be tightly fitted. Otherwise, the bearing will freely swing inside the free clearance, resulting in an excessive level of vibrations. Usually two or more bearings are used to support a shaft, and only the bearings at one end of the shaft can have a completely tight fit of the two rings in their seats. The radial bearing on the other end of the shaft must have one ring with a loose fit. This is essential to allow the ring to *float* on the shaft or inside the housing seat in order to prevent thermal stresses during operation due to thermal expansion of the shaft length relative to the machine.

In many designs, the bearing is located between a shoulder on the shaft and a standard locknut and lock washer, for preventing any axial bearing displacement (Fig. 13-5). Precision machining of the housing and shaft seats is required in order to prevent the bending of the bearing relative to the shaft. The shoulders on the shaft must form a plane normal to the shaft centerline, the threads on the

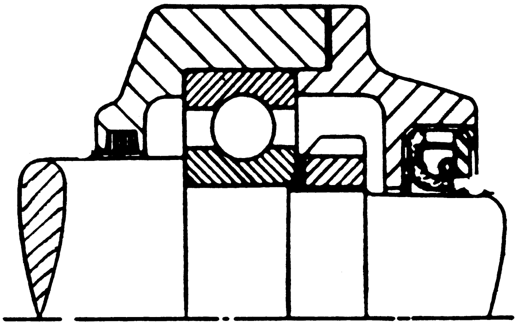


FIG. 13-5 Locating a bearing by a locknut.

locknut and shaft must be precisely cut, with very small run-out tolerance, and the washers must have parallel surfaces in order to secure uniform parallel contact with the inner ring face. Many mass-produced small machines and appliances that operate under light conditions rely only on a tight fit for locating the bearings, in order to reduce the cost of production.

13.9 STRESS AND DEFORMATION DUE TO TIGHT FITS

Tight-fit mounting is where there is an interference (negative clearance) between the housing seat and the outer ring or between the bearing bore and the shaft. A hydraulic or mechanical press is used for bearing mounting. For larger bearings, a temperature difference is used for mounting. The bearing is heated and fitted on the shaft, or the housing is heated for fitting in the bearing.

Tight-fitting involves elastic deformation. Tight-fitting shrinks the outer ring and expands the inner ring. After tight-fit mounting of the inner ring on a shaft, the bore of the inner rings slightly increases in diameter, while the shaft diameter slightly decreases. This results in tangential tensile stresses around the ring, while tight-fit mounting of the outer ring into the housing seat results in compression stresses. When the bearing housing and shaft are made of the same material (steel), the stress equation is simplified. In addition, in order to simplify the equation, it is assumed that the bearing rings have a rectangular cross section.

If the diameter interference of the inner ring on the shaft is Δd, the equation for the tensile stress in the inner ring is

$$\sigma_t = \frac{1}{2} E \left(1 + \frac{d_i^2}{d_o^2}\right) \frac{\Delta d}{d_i} \tag{13-19}$$

where

d_i = ID (inside diameter) of inner ring
d_o = OD (outside diameter) of inner ring
E = modulus of elasticity
Δd = diameter interference (negative clearance)

In a similar way, if the diameter interference of the outer ring inside the housing seat is ΔD, the equation for the compression stresses in the outer ring is

$$\sigma_t = \frac{1}{2} E (1 + \frac{D_i^2}{D_o^2}) \frac{\Delta D}{D_o} \tag{13-20}$$

where

D_i = ID of outer ring
D_o = the OD of outer ring
ΔD = diameter interference of outer ring

For the two rings, there are compression stresses in the radial direction. At the interference boundary, the compression stress is in the form of pressure between the rings and the seats. The equation for the pressure between the inner ring and the shaft (for a full shaft) is

$$p_{(\mathrm{shaft})} = \frac{1}{2} E \left(1 + \frac{d_i^2}{d_o^2} \right) \frac{\Delta d}{d_i} \tag{13-21}$$

In a similar way, the equation for the pressure between the outer ring and the housing is

$$p_{(\mathrm{housing})} = \frac{1}{2} E \left(1 + \frac{D_i^2}{D_o^2} \right) \frac{\Delta D}{D_o} \tag{13-22}$$

The pressure keeps the rings tight in place, and the friction prevents any sliding in the axial direction or due to the rotation of the ring. The axial load required to pull out the fitted ring or to displace it in the axial direction is

$$F_a = f \pi d L p \tag{13-23a}$$

where f is the static friction coefficient. In steel-on-steel bearings, the range of the static friction coefficient is 0.1–0.25. In a similar way, the equation for the maximum torque that can be transmitted through the tight fit by friction (without key) is

$$T_{\max} = f \pi L p \frac{d^2}{2} \tag{13-23b}$$

13.9.1 Radial Clearance Reduction Due to Interference Fit

Interference-fit mounting of the inner or outer ring results in elastic deformation and, in turn, in a reduction of the radial clearance of the bearing. The reduction of radial clearance, Δ_s, due to tight-fit mounting of interference Δd with the shaft is

$$\Delta_s = \frac{d_i}{d_o}\,\Delta d \tag{13-24a}$$

In a similar way, the reduction in radial clearance due to interference with the housing seat is

$$\Delta_h = \frac{D_i}{D_o}\,\Delta d \tag{13-24b}$$

13.9.2 Reduction of Surface Roughness by Tight Fit

The actual interference is reduced by a reduction of roughness (surface smoothing) of tight-fit mating surfaces. Roughness reduction is equivalent to interference loss. For the calculation of the stresses and radial clearance reduction by interference fit, the surface smoothing should be considered.

The greater the surface roughness of the mating parts, the greater the resulting smoothing effect, which will result in interference loss. According to DIN 7190 standard, about 60% of the roughness depth, R_s, is expected to be smoothed (reduction of the outside diameter and increase of the inside diameter) when parts are mated by a tight-fit assembly.

In rolling bearing mounting, the smoothing of the hardened fine-finish surfaces of the rolling bearing rings can be neglected in comparison to the smoothing of the softer surfaces of the shaft and housing. Table 13-14 can be

TABLE 13-14 Surface Roughness for Various Machining Qualities

	Roughnes of surfaces, R_s	
	μm	μin
Ultrafine grinding	0.8	32
Fine grinding	2	79
Ultrafine turning	4	158
Fine turning	6	236

used as a guide for determining the roughness, R_s, according to the quality of machining (Eschmann et al., 1985).

The smoothing effect is neglected for precision-ground and hardened bearing rings, because the roughness R_s is very small. However, there is interference loss to the part fitted to the bearing, such as a shaft or housing. Since 60% of the roughness depth, R_s, is smoothed, the reduction in diameter, ΔD_s, by smoothing is estimated to be

$$\Delta D_s = 1.2R_s \tag{13-25}$$

Here,

$\Delta D_s =$ reduction in diameter due to smoothing (interference loss)
$R_s =$ surface roughness (maximum peak to valley height)

In addition to interference loss due to smoothing, losses due to uneven thermal expansion occur. When the outer ring and housing or inner ring and shaft are made from different materials, operating temperatures will alter the original interference. Usually, the bearing housing is made of a lighter material than the bearing outer ring (higher thermal expansion coefficient), resulting in interference loss at operating temperatures higher than the ambient temperature. Interference loss due to thermal expansion can be calculated as follows:

$$\Delta D_t = D(\alpha_o - \alpha_i)(T_o - T_a) \tag{13-26}$$

Here,

$\Delta D_t =$ interference loss due to thermal expansion
$D =$ bearing OD
$\alpha_o =$ coefficient of expansion of outside metal
$\alpha_i =$ coefficient of expansion of inside metal
$T_o =$ operating temperature
$T_a =$ ambient temperature

On the housing side, the effective interference after interference reduction due to surface smoothing and thermal expansion of dissimilar materials is

$$u = \Delta D_{(\text{machining interference})} - \Delta D_s - \Delta D_t \tag{13-27}$$

where

$u =$ effective interference
$\Delta D =$ machining interference
$\Delta D_s =$ diameter reduction due to smoothing
$\Delta D_t =$ interference loss due to thermal expansion

13.9.3 Bearing Radial Clearance During Operation

Bearings are manufactured with a larger radial clearance than required for operation. The original manufactured radial clearance is reduced by tight-fit mounting and later by uneven thermal expansion of the rings during operation. The design engineer should estimate the radial clearance during operation. In many cases, the radical clearance becomes interference, and the design engineer should conduct calculations to ensure that the interference is not excessive. The interference results in extra rolling contact pressure, which can reduce the fatigue life of the bearing. However, small interference is desirable for many applications, because it increases the bearing stiffness.

The purpose of the following section is to demonstrate the calculation of the final bearing clearance (or interference). This calculation is not completely accurate, because it involves estimation of the temperature difference between the inner and outer rings.

13.9.4 Effects of Temperature Difference Between Rings

During operation, there is uneven temperature distribution in the bearing. In Sec. 13.3.3, it was mentioned that for average operation speed the temperature of the inner ring is 5°–10°C higher than that of the outer ring (if the housing is cooled by air flow, the difference increases to 15°–20°C). The temperature difference causes the inner ring to expand more than the outer ring, resulting in a reduction of the bearing radial clearance. The radial clearance reduction can be estimated by the equation

$$\Delta D_{td} = \frac{\Delta T\ \alpha(d + D)}{2} \tag{13-28}$$

Here,

ΔD_{td} = diameter clearance reduction due to temperature difference between inner and outer rings

ΔT = temperature difference between inner and outer rings

α = coefficient of linear thermal expansion

d = bearing bore diameter

D = bearing OD

Example Problem 13-3

Calculation of Operating Clearance

Find the operating clearance (or interference) for a standard deep-groove ball bearing No. 6306 that is fitted on a shaft and inside housing as shown in Fig. 13-6. During operation, the temperature of the inner ring as well as of the shaft is 10°C higher than that of the outer ring and housing. The dimensions and tolerances of the inner ring and shaft are:

Bore diameter: $d = 30\,\text{mm}\ (-10,\ +0)\,\mu\text{m}$
Shaft diameter: $d_s = 30\,\text{mm}\ (+15,\ +2)\,\mu\text{m}$ k6
OD of inner ring: $d_1 = 38.2\,\text{mm}$

The dimensions and tolerances of outer ring and housing seat are:

OD of outer ring: $D = 72\,\text{mm}\ (+0,\ -11)\,\mu\text{m}$
ID of outer ring: $D_1 = 59.9\,\text{mm}$
ID of housing seat: $D_H = 72\,\text{mm}\ (-15,\ +4)\,\mu\text{m}$ K6
Shaft finish: fine grinding
Housing finish: fine grinding
Radial clearance before mounting: C5 Group, 40–50 μm
Coefficient of linear expansion of steel: $\alpha = 0.000011$ [1/K]

Consider surface smoothing, elastic deformation, and thermal expansion while calculating the operating radial clearance.

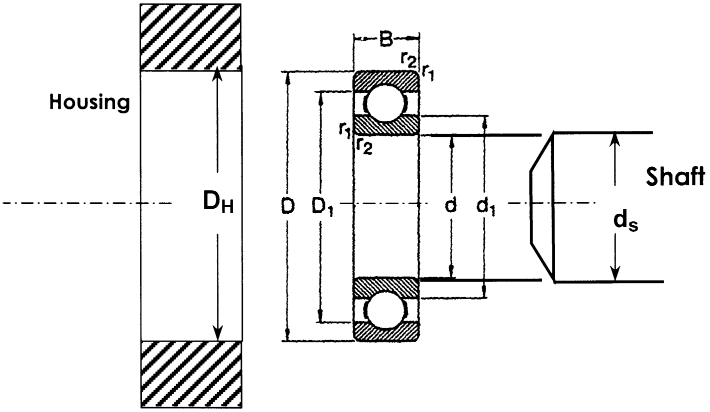


FIG. 13-6 Dimensions and tolerances of rolling bearing, shaft, and housing.

Solution

In most cases, the machining process of the rings, shaft, and housing seat will stop not too far after reaching the desired tolerance. There is high probability that the actual dimension will be near one-third of the tolerance zone, measured from the tolerance boundary close to the surface where the machining started. Common engineering practice is to take two-thirds of the tolerance range and then add to that the lowest tolerance (Eschmann et al., 1985). The result should be a value close to the side on which the machining is started. Therefore,

Shaft interference :

Bearing bore: $(-10+0)\times 2/3+0=-7\,\mu\text{m}$

Shaft: $(15-2)\times 2/3+2=+11\,\mu\text{m}$

Total theoretical interference fit is: $11+7=+18\,\mu\text{m}$

Housing interference :

OD of outer ring: $(0+11)\times 2/3-11=-4\,\mu\text{m}$

ID of housing seat: $(-15-4)\times 2/3+4=-9\,\mu\text{m}$

Total theoretical interference fit: $9-4=+5\,\mu\text{m}$

The bearing is made of hardened steel and is precision ground, so smoothing of the bearing inner and outer rings can be neglected. The R_s value for a finely ground surface is obtained from Table 13-5:

Smoothing to finely ground shaft: $\Delta D_s=1.2R_s=1.2(2)=2.4\,\mu\text{m}$
Smoothing to finely ground housing: $\Delta D_s=1.2R_s=1.2(2)=2.4\,\mu\text{m}$

In this example, the shaft and housing are both made of steel, so there is no change in interference due to different thermal expansion of two materials.

The effective interference becomes:

$$u=\text{theoretical interference}-\Delta D_s$$

Inner ring: $u_i=18-2.4=15.6\,\mu\text{m}$

Outer ring: $u_o=5-2.4=2.6\,\mu\text{m}$

The radial clearance reduction due to tight-fit installation of the rolling bearing is also considered. The reduction in clearance due to interference with the shaft is (Eq. 13-24)

$$\Delta_s=\frac{d}{d_1}u_i=\frac{30\,\text{mm}}{38.2\,\text{mm}}\times 15.6\,\mu\text{m}=12.25\,\mu\text{m}$$

The reduction in clearance due to interference with the housing is

$$\Delta_H=\frac{D_1}{D}u_0=\frac{59.9\,\text{mm}}{72\,\text{mm}}\times 2.6\,\mu\text{m}=2.16\,\mu\text{m}$$

The total radial clearance reduction due to installation is therefore

$$\Delta_s + \Delta_H = 12.25\,\mu\text{m} + 2.16\,\mu\text{m} = 14.4\,\mu\text{m}$$

Finally, as stated in the problem, there is a temperature difference of $\Delta T = 10^\circ\text{C}$ between the inner and outer rings. This is due to the more rapid heat transfer away from the housing than from the shaft. In turn the shaft and inner ring will have a higher operating temperature than the outer ring. This will result in higher thermal expansion of the inner ring, which will further reduce the radial clearance. The thermal clearance reduction is

$$\Delta D_{\text{th}} = \frac{\Delta T\ \alpha(d + D)}{2}$$

$$\Delta D_{\text{th}} = 10(\text{K}) \times 0.000011\ (1/\text{K}) \times \frac{(30 + 72)\,\text{mm}}{2} \times \frac{1000\,\text{m}}{\text{mm}}$$
$$= 5.6 \times 10^{-3}\,\text{m} = 5.6\,\mu\text{m}$$

In summary, the expected radial running clearance of this bearing will be:

Radial clearance before mounting:	40–50 μm
Radial clearance reduction due to mounting:	−14.4 μm
Radial clearance reduction by thermal expansion:	−5.6 μm
Expected radial clearance during operation	20–30 μm

13.10 BEARING MOUNTING ARRANGEMENTS

An important part of bearing design is the mounting arrangement, which requires careful consideration. For an appropriate design, the following aspects should be considered.

The shaft should be able to have free thermal expansion in the axial direction, due to its temperature rise during operation. This is essential for preventing extra thermal stresses.

The mounting arrangement should allow easy mounting and dismounting of the bearings. The designer must keep in mind that rolling bearings need maintenance and replacement.

The shaft and bearings are part of a dynamic system that should be designed to have sufficient rigidity to minimize vibrations and for improvement of running precision. For improved rigidity, the mounting arrangement is often designed for elimination of any clearance by preloading the bearings.

Bearing arrangements should ensure that the bearings are located in their place while supporting the radial and axial forces.

It was discussed earlier that during operation, if the housing has no cooling arrangement, the temperatures of the shaft and inner ring could be 5°–10°C higher than that of the outer ring. If the housing is cooled by air flow, the temperatures of the inner ring can increase to 15°–20°C higher than that of the outer ring. During operation, the temperature difference between the shaft and the machine frame is higher than between the rings. This results in a thermal elongation of the shaft relative to the machine frame that can cause extra stresses at the rolling contact of the bearings. In addition, due to manufacturing tolerances, the distances between the shaft seats and the housing seats are not equal. The extra stresses caused by thermal elongation and manufacturing tolerances can be very high if the shaft is long and there is a large distance between the supporting bearings.

This problem can be prevented by appropriate design of the bearing arrangement. The design must provide one bearing with a loose fit so that it will have the freedom to float in the axial direction (*floating bearing*). In most cases, the loose fit of the floating bearing is at the outer ring, which is fitted in the housing seat.

A floating bearing allows free axial elongation of the shaft. The common design is referred to as a *locating/floating* or *fixed-end/free-end* bearing arrangement. In this design, one bearing is the *locating* bearing, which is fixed in the axial direction to the housing and shaft and can support thrust (axial) as well as radial loads. On the other side of the shaft, the second bearing is *floating*, in the sense that it can slide freely, relative to its seat, in the axial direction. The floating bearing can support only radial loads, and only the locating bearing supports the entire thrust load on the shaft. In shafts supported by two or more bearings, only one bearing is designed as a locating bearing, while all the rest are floating bearings. This is essential in order to prevent extremely high thermal stresses in the bearings.

An example of a *locating/floating* bearing arrangement is shown in Fig. 13-7. Additional practical examples are presented in Sec. 13-12. The bearing on the left side of the shaft is fixed in the axial direction and can support thrust forces in the two directions as well as radial force. The bearing on the right end of the shaft can float in the axial direction and can support only radial force. Axial floating of the bearing is achieved by providing the housing with a loose fit (a clearance between the housing seat and the bearing outer ring). In certain applications, two angular contact ball bearings or tapered roller bearings that are symmetrically arranged and preloaded are used as locating bearings (see Sec. 13.11). This design provides for an accurate rigid location of the shaft.

In principle, axial floating of the shaft is also possible by means of a loose fit between the shaft and the bearing bore. However, for a rotating shaft and stationary housing, the clearance must be on the housing side, to prevent wear of the shaft surface during starting and stopping of the machine.

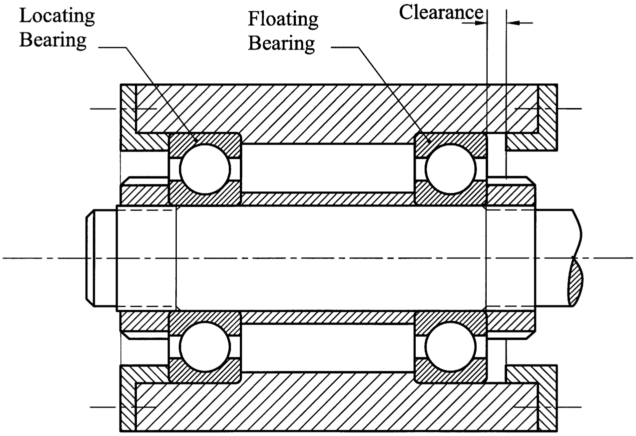


FIG. 13-7 Locating/floating bearing arrangement.

During starting and stopping, the shaft has a high angular acceleration. In turn, the moment of inertia (of the inner ring and balls of the bearing) causes inertial torque resistance (in the direction opposite to that of the shaft angular acceleration). In many cases, the inertial torque is higher than the friction and the shaft would slide during the start-up inside the bearing bore. The relative sliding can cause severe wear of the shaft surface. This undesired effect can be prevented by means of a tight fit (interference fit) of the shaft inside the bearing inner ring bore. Shafts seats are often rebuilt due to the wear during starting and stopping. Therefore, for a rotating shaft, the clearance (loose fit) of the floating arrangement is always on the housing seat, while a tight fit is on the shaft side.

In certain machines, the shaft is stationary and the outer ring rotates, e.g., a grinding machine or a centrifuge. In such applications, a tight fit must be provided on the rotating side and a sliding fit at the seat of the stationary shaft.

Compensation for shaft elongation can also be achieved by using a cylindrical roller bearing. Certain cylindrical roller bearings are designed to operate as floating bearings by allowing the roller-and-cage assembly to shift in the axial direction on the raceway of the outer ring. For this purpose, the bearing rings are designed without ribs (often named *lipless bearing rings*). All other bearing types, such as a deep-groove ball bearing or a spherical roller bearing, can operate as floating bearings only if one bearing ring has a loose fit in its seat.

13.10.1 Tandem Arrangement

Two angular contact ball bearings can be used in series for heavy unidirectional thrust loads. Precise dimensions and high quality surface finish are required to secure load sharing of the two bearings. The arrangement of two or more angular contact bearings, adjacent to each other in the same direction, is referred to as *tandem arrangement*. This arrangement is used to increase the thrust load carrying capacity as well as the radial load capacity. Tandem arrangement is often used in spindles of machine tools, where high axial stiffness and high thrust load capacity are required; examples are shown in Sec. 13.12. Bearing manufacturers provide a combination of two angular contact ball bearings that are designed and made for tandem arrangement.

13.10.2 Bearing Seat Precision

For a locating bearing, the inner and outer rings are tightly fitted into their seats. But a floating bearing has one ring that is fitted tightly, while the other ring has a loose fit to allow free axial sliding. For a floating bearing, if the shaft is rotating, only the inner ring must be mounted by interference fit. If the outer ring is rotating, only the outer ring is mounted by interference fit. The reason for a tight fit of the rotating ring is to avoid sliding and wear during start-up and stopping.

In interference fit (tight fit) there is elastic deformation of the ring that reduces the internal clearance of the bearing. Therefore, it is important to select the recommended standard fit for a proper internal radial clearance after the bearing mounting. The bearings are manufactured with internal clearance to provide for this elastic deformation and for thermal expansion of the shaft and inner ring during operation.

In the case of a tight fit, the bearing can be mounted by application of heat or cold-mounted by pressing the face of the ring that is tightly fitted (in order to prevent bearing damage, never apply force through the rolling elements). In many cases, such as a bore diameter over 70 mm, it is easier to mount via temperature difference. This can be obtained by heating one part, or heating and cooling, respectively, the two parts. An additional simple method for tight-fit mounting is the use of tapered-bore bearings combined with tapered seats. The bearing is tightened in the axial direction by a locknut. A tapered adapter sleeve is another convenient method for tight-fit mounting.

For the shaft and housing seats, precision and good surface finish are required. In fact, the precision and surface finish of the seats should be similar to those of the rolling bearing in contact with the seat. Whenever possible, a ground finish of the bearing seats on the shaft and housing is preferred. Only in exceptional cases of low speed and load—if cost saving is critical—are rougher shaft and housing seats used. In such cases, rougher ball bearings can be used as well, in order to reduce the cost in low-cost machines.

A common locating arrangement is where the ring is tightly fixed between a shaft shoulder and a locknut, as shown in Fig. 13-5. Precision of the shaft shoulder seat is required because many rolling bearings are so narrow that they are not aligned accurately by the length of the seat on the shaft. The final accurate alignment is by the shaft shoulder and nut. Precision of the seats and the nuts is particularly important for medium-and high-speed applications. The shoulder plane should be perpendicular to the shaft centerline (squareness). In the same way, locknut precision is required. A standard locknut should have precise thread having maximum face run-out within 0.05 mm (0.002 in.). Precision nuts with much lower face run-out are used for precision or high-speed applications.

Quality inspection of shaft seats and shoulders for axial and radial run-out is required for medium and high speeds. Rotating the shaft between centers, with a dial indicator placed against the seat or shoulder, is the standard inspection. Proper manufacturing practice is to grind the seats for the inner ring and shaft shoulder together, in one clamping of the shaft, and the same applies to the housing. One clamping ensures that the two surfaces are perpendicular.

The recommended height of a shaft shoulder is one-half of the inner ring face. Were the shoulder too low, it would result in a plastic deformation of the shoulder due to excessive pressure, particularly under high thrust load. On the other hand, the shaft shoulder should not be too high (more than half of the inner ring face), to allow disassembly and removal of the bearing from the shaft. A puller placed against the inner ring surface is usually used for removing the bearing.

Careful design of the corners of the shaft shoulder and bearing seat is necessary. The corner radius of the seat must be less than that of the ring. In many designs, the corner has an undercut or a shaft fillet to secure a proper fit to the bearing ring. However, an undercut weakens the shaft and causes stress concentration at the corner. Whenever weakening of the shaft is not desired, a fillet can be used. Standard fillet sizes for each particular bearing are available and are listed in bearing catalogues. In many cases, a small taper is provided on the bearing seat edge to provide a guide to assist in mounting the bearing.

13.11 ADJUSTABLE BEARING ARRANGEMENT

The bearing clearance allows a free radial or axial displacement of the inner ring relative to the outer ring. The objective of an adjustable arrangement of angular contact ball bearings or tapered roller bearings is to eliminate this undesired clearance. In addition, by using an adjustable arrangement it is possible to preload the bearing (negative clearance). Preload means that there are compression stresses and elastic deformation at the contacts of the rolling elements and the raceways before the bearing is in operation.

Bearing preload is important for many applications requiring high system rigidity. By preloading the bearing, the stiffness of the machine increases; namely, there is a reduction in the elastic deformation under external load. Bearing preload causes extra stresses at the contacts between the rolling elements and the raceways, which can reduce the fatigue life of the bearing. Therefore, the preload must be precisely adjusted, because excessive contact stresses will have an adverse effect on bearing life.

In an adjustable arrangement, angular ball bearings or tapered bearings are mounted in pairs against each other on one shaft and are preloaded. Deep-groove ball bearings are used as well for adjustable arrangements, because they act like angular contact ball bearings with a small contact angle. The arrangement is designed to allow, during mounting, for one ring to slide in its seat, in the axial direction, for adjusting the bearing clearance or even provide preload inside the bearing. This is done by tightening the inner ring by means of a nut on the shaft or via an alternative design for tightening the outer ring of the bearing in the axial direction. Examples of adjustable arrangements are shown in Figs. 13-8 and 13-9.

It was discussed earlier that by a tight fit of the bearing rings in their seats, the radial clearance can be eliminated and the bearing can be preloaded. However, better control and precision of the preload can be achieved via an adjustable arrangement using angular contact ball bearings or tapered roller bearings. Preload by tight fit of the bearings in their seats is not always precise. This is due to machining tolerances of the seats and bearing rings. However, in an

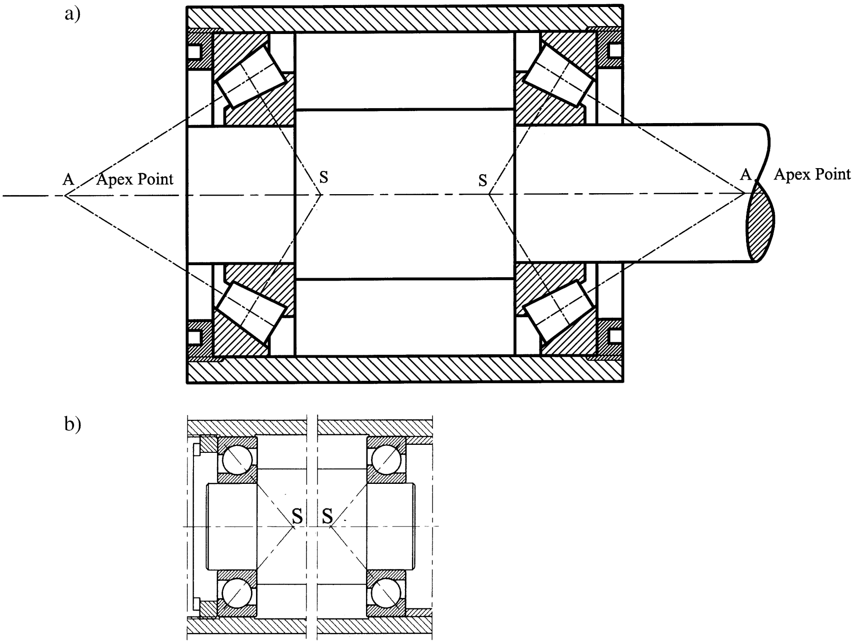


FIG. 13-8 (a) Adjustable arrangement, apex points outside the two bearings. (b) Similar adjustable arrangement for angular contact bearings.

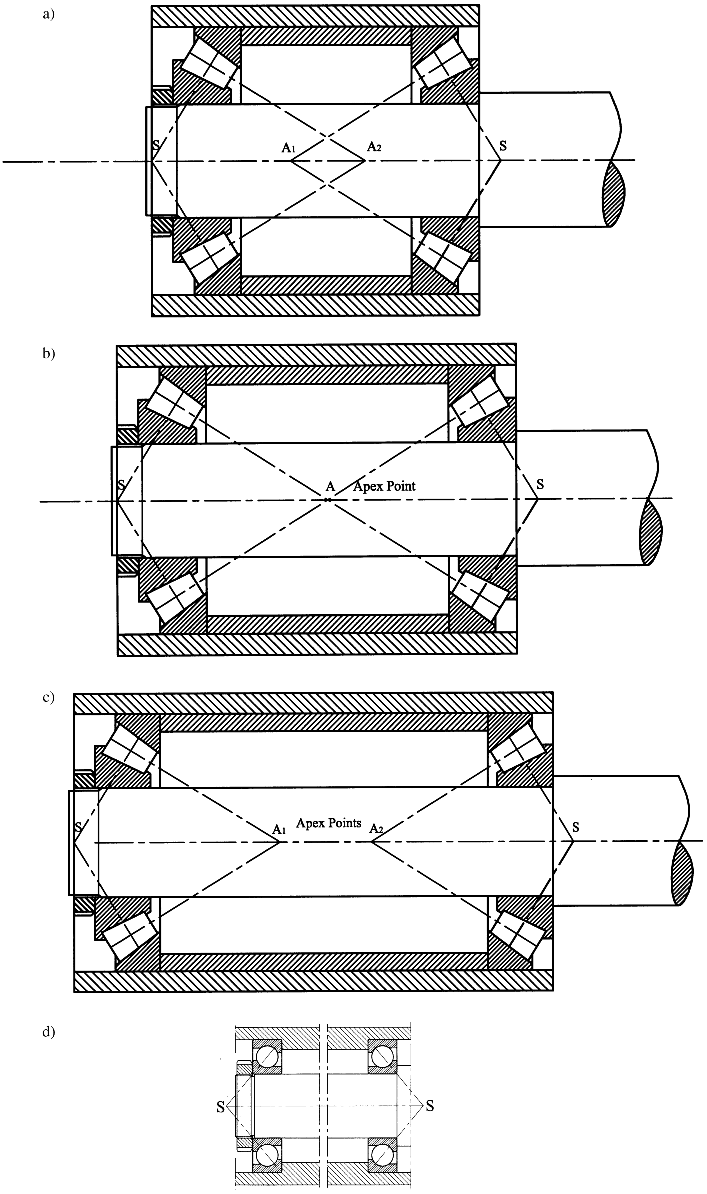


FIG. 13-9 (a) Adjustable arrangement, apex points between the bearings overlap. (b) Adjustable arrangement, apex points coincide between the two bearings. (c) Adjustable arrangement, apex points apart between the bearings. (d) Similar adjustable arrangements for angular contact bearings.

adjustable arrangement, the preload is independent of machining tolerances. Nevertheless, thermal expansion of the shaft during operation must be taken into consideration when the adjustment is performed during assembly, when the machine is cold. If the operation temperatures of the shaft and machine are known from previous experience, the thermal expansion can be calculated, and precise adjustment to the desired tightness during bearing operation is possible.

13.11.1 Thermal Effects

Whenever the operation temperatures are unknown, it is possible to reduce the thermal stresses by having the adjustable pair of bearings close to each other (a short shaft length between the two bearings). A better alternative is to design an adjustable bearing arrangement that can be adjusted after the machine is assembled and run. In such cases, the adjustment is performed after the machine has been operating for some time and thermal equilibrium has been reached.

In a tapered bearing, the rolling elements and races have a conical shape, with a line contact between them. In order to have a rolling motion, all the contact lines of the tapered rollers and raceways must meet at a common point on the axis, referred to as the *apex point*. Similarly for angular contact ball bearings, the lines of contact angle meet at the apex point on the bearing axis. There are two types of adjustable bearing arrangement, depending on the location of the apex points. The first type is where the two apex points, A, are outside the space between the two bearings (see Fig 13-8a and 8b). The second type is where the apex points are between the two bearings (see Fig. 13-9a, 9b, 9c, 9d). The designer should consider the level of thermal expansion in order to choose between these arrangements.

It was discussed in Sec. 13.3.3 that the temperature of the inner ring is higher than that of the outer ring. For the same reason, the shaft temperature is higher than the housing temperature. In turn, the shaft is thermally expanding in the axial direction more than the distance between the two outer bearing seats in the housing. The thermal expansion of the shaft relative to the housing seats is proportional to the distance between the two bearings. The diameters of the shaft and inner ring will also expand thermally more than the outer ring and housing diameters.

13.11.1.1 Apex Points Outside the Two Bearings

This bearing arrangement is often referred to as *X arrangement*, because the lines in the direction normal to the contact lines, intersecting at point S, form an X shape. These lines are the directions of the forces acting on the rolling element. In angular contact ball bearings (Fig. 13-8b), these lines form the contact angle.

The temperature rise of the shaft relative to that of the housing increases the length and diameter of the shaft as well as the diameter of the cone (inner ring) of the bearings. The first type of an adjustable bearing arrangement is shown in Fig 13-8. In this arrangement, the apex points, *A*, are outside the two bearings. As shown in Fig. 13-8, tightening a threaded ring on the housing side does the adjustment. In this way, the bearing cup (outer ring) is shifted in the axial direction and, thus, the clearance in the two bearings can be adjusted. In this arrangement, a temperature rise will always result in a tighter bearing clearance.

In the bearing arrangement of Figs. 13-8a and 13-8b, if the bearings are preloaded when the machine is cold, the temperature rise results in a higher bearing preload and rolling contact stresses. If some bearing clearance is left after the adjustment, the clearance will be reduced due to the thermal expansion. As discussed earlier, the advantage of this arrangement type is that it can be designed to allow a final adjustment during operation, after the machine has reached a steady thermal equilibrium.

13.11.1.2 Apex points between the two bearings

This arrangement is often referred to as *O arrangement*, because the lines in the direction normal to the contact lines, intersecting at point S, form an O shape. These lines are the directions of the forces acting on the rolling element. In angular contact ball bearings (see Fig. 13-9d), these lines form the contact angle.

In general, arrangement of apex points between the two bearings (*O arrangement*) is preferred whenever a strong axial guidance is required. This means that the shaft is supported more rigidly than the adjustable arrangement with apex points outside the bearings. The direction of the rolling-elements reaction force resists better any rotational vibrations of the shaft around an axis perpendicular to the shaft centerline.

The effect of a temperature rise may be different in the second arrangement type, which is shown in Figs. 13-9a, 13-9b, and 13-9c, where the apex points are between the bearings. As shown in these figures, tightening a nut on the shaft side does the adjustment. The bearing cone (inner ring) is shifted in the axial direction, and the clearance in the two bearings is adjusted.

During operation, the temperature rise increases the shaft length and at the same time increases the diameter of the inner ring (cone). In the second arrangement type (Figs. 13-9), a thermal expansion of the shaft length has a loosening effect on the two bearings; however, at the same time, the thermal expansion of the cone diameter has a tightening effect. The combined effect depends on the ratio of the shaft length to the cone diameter. The combined thermal effects can be determined by the location of the apex points. This arrangement can be divided into three cases:

1. For a short distance between the two bearings, the roller cone apex points overlap, as shown in Fig. 13-9a. In this case, the thermal expansion of the cone (inner ring) diameter has a larger effect than

the axial expansion of the shaft. The combined effect is that the thermal expansion increases the preload. This combined effect should be considered and the bearings should be adjusted with a reduced preload in comparison to the desired preload during operation.

2. The two apex points coincide, as shown in Fig. 13-9b. In this case, the axial and the radial thermal expansions compensate each other without any significant thermal effect on the clearance.
3. The distance between the bearings is large and the roller cone apices do not overlap, as shown in Fig. 13-9c. In this case, the cone (inner ring) thermal expansion is less than that of the shaft. In turn, the combined thermal effect is to increase the clearances of the two bearings (or reduce the preload). This should be considered, and the bearings are usually adjusted tighter with more preload in comparison to the desired preload during operation.

The selection of the adjustment arrangement type depends on several factors. The second type, where the roller cone apices are between the bearings, has more rigidity to keep the shaft centerline in place. In addition, it can be designed so that the thermal expansion is compensated. The first type, where the roller cone apices are outside the bearings, is often selected in order to allow a fine adjustment during operation. This is possible to do only if the threaded ring (or other adjustment design) is accessible for adjustment during the operation of the machine.

13.11.2 Inner and Outer Ring Fits

The inner or outer ring that is adjusted should move freely by means of a slightly loose fit, while the other ring is mounted with a tight fit. As with other rolling bearings, the inner ring (cone) should always be mounted with a tight fit when the cone rotates. Similarly, the outer ring (cup) should be mounted with a tight fit when it rotates. For a rotating shaft, this requirement usually favors the first type of adjustable bearing arrangement, where the apex points are outside the two bearings. However, the second type is often used for rotating shafts as well.

If the housing rotates, as in a nondriven car wheel, the cup is tightly fitted. If the bearing is subjected to severe loads, shocks, or frequent direction reversals, such as in construction equipment, both cup and cone must be tightly fitted.

For high-speed applications, such as turbines and high-speed machine tools, an adjustable arrangement of angular contact ball bearings is preferred, because tapered rolling bearings have higher friction. In a similar way to the intersection of cone apices, in angular contact bearings the arrangement type is determined by the intersection of the lines normal to the angular bearing contact lines.

Manufactured pairs of angular contact ball bearings or tapered bearings are available. The bearings are paired in a first-or second-type arrangement. Angular contact ball bearings of these designs are accurately finished and can be selected with a low axial clearance, a zero clearance, or a light preload.

13.11.3 Designs for Reduction of Thermal Effects on Bearings Preload

It is important that the bearings in an adjustable arrangement will operate with the desired precise preload force. However, the operating temperature can fluctuate resulting in a variable preload force. Excessive preload can reduce the fatigue life of the bearing, and if the preload is reduced, the bearings' stiffness may be too low. Engineers are always looking for new designs for mitigating the thermal effect, so that a precise preload will be sustained in the bearings.

It is possible to design a preloaded bearing arrangement where springs provide the thrust force. The spring force is not as sensitive to the thermal elongation in comparison to the rigid shaft in the common adjustable bearing arrangement. The advantage is that the spring force is constant, and the preload force does not increase by the temperature rise during bearing operation. Examples of designs where springs provide the preload force are shown in Sec. 13.12.

Additional example for reducing the effect of the temperature on the level of the preload force is by using two spacer sleeves between the two bearings for the outer and inner rings of the adjustable arrangement. The two spacer sleeves have only a small contact area with the rings and housing. The spacer for the inner rings has an air clearance with the shaft, and the spacer for the outer rings has an air clearance with the housing. It results that the two long spacer sleeves are nearly thermally isolated, and have approximately equal temperature during operation. In this way, the axial elongation of the two long spacer sleeves is equal without any significant effect on the preload. An example of a design where two long spacer sleeves are used for a NC Lathe spindle bearing arrangement is shown in Sec. 13.12.

13.11.4 Machine Tool Spindles

The two most important requirements for machine tool spindle bearings are

(a) high precision (very low bearing run-out)
(b) high rigidity (very low elastic deformation under load).

High precision spindle bearings are manufactured with very low tolerances, which are tested for very small radial and axial run-out. In addition, the bearing seats must have similar precision, and very good surface finish. The requirement of high stiffness can be achieved by using relatively large bearings that are precisely preloaded. For precision machining, the cutting forces should result in very small elastic deformation. Therefore, the system of spindle and bearings must be rigid. For this purpose, machine tool designs entail large diameter spindle and large bearings, in comparison to other machines with similar forces. Moreover, to ensure rigidity of the system, the bearings must be preloaded, in order to minimize the elastic deformation at the contacts of rolling elements and races.

The requirement for high stiffness results in large bearings relative to other machines with similar forces; therefore, the fatigue life is usually not a problem in machine tool spindle bearings. Spindle bearings usually do not fail by fatigue, but can wear out, and it is important to have clean lubricant to reduce wear.

In most cases, machine tools are fitted with angular contact ball bearings to support the high thrust load. The requirement for high axial stiffness under heavy thrust cutting forces is achieved in many cases by arrangement of two or more angular contact ball bearings in *tandem arrangement* (see Sec. 13.10.1). The bearings are preloaded by adjustable arrangement, and care must be taken to ensure that the preload will remain constant and will not vary due to variable bearing temperature. Examples of bearing arrangements for machine tool spindles are presented in Sec. 13.12.

13.12 EXAMPLES OF BEARING ARRANGEMENTS IN MACHINERY

13.12.1 Vertical-Pump Motor (Fig. 13-10a)

Design data
Power: 160 kW
Speed: 3000 RPM
Thrust force: 14 kN (Total of weight of rotor and impeller, pump thrust force and spring force).

Tolerances
cylindrical roller bearing shaft m5; housing M6
deep grove ball bearing: shaft k5; housing H6
angular contact bearing: shaft k5; housing E8

Lubrication: Grease lubrication with time period of 1000 hours between lubrications.

Design: This is a locating/floating bearing arrangement. The two bearings at the top form the locating side, whereas the lower cylindrical roller bearing is a floating bearing. In the locating top bearings, preload is

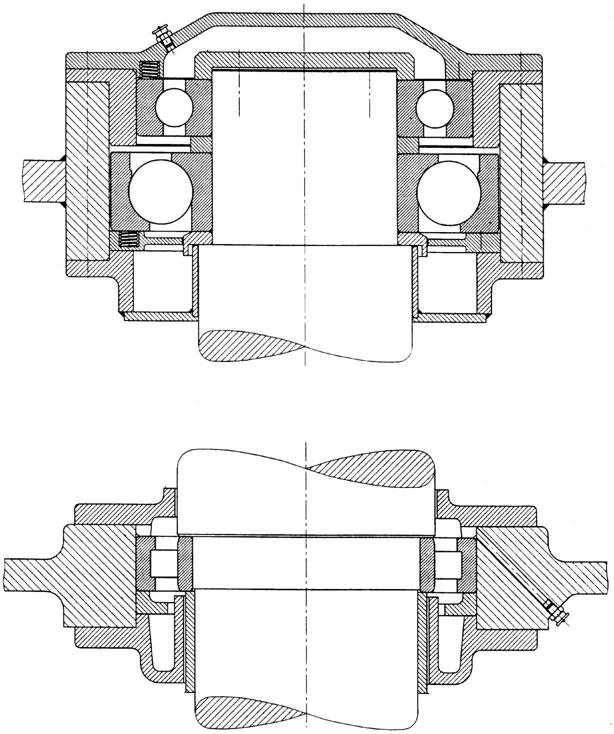


FIG. 13-10a Vertical pump motor. (From FAG, 1998, with permission of FAG and Handel AG.)

done via springs. The springs ensure a constant predetermined load (see Sec. 13.11.3). The angular contact bearing carries the thrust load, and the deep grove bearing carries any possible radial load (and the small spring axial preload). A clearance fit relieves the angular contact bearing from any radial load, which can reduce its fatigue life. The lower cylindrical roller bearing carries only radial load.

13.12.2 NC-Lathe Spindle

Figure 13-10b shows a bearing arrangement for a spindle of *numerically controlled* (NC) lathe. The bearings have adjustable arrangement and are lightly preloaded. The adjustable arrangement is of the type of apex points between the two bearings (often referred to as an *O arrangement*). As the speed is relatively high, the spindle is fitted with angular contact ball bearings (lower friction than tapered bearings). However, in order to support the high thrust load and provide

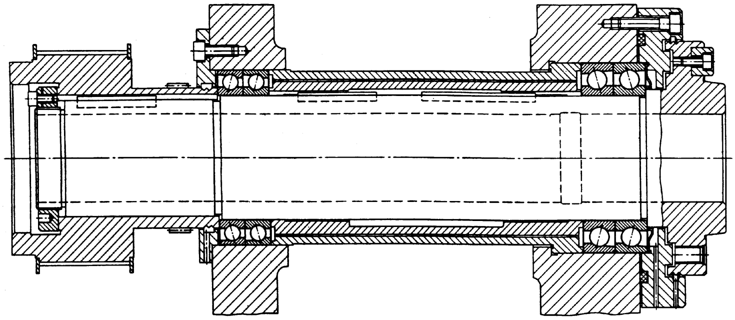


FIG. 13-10b NC-lathe main spindle. (From FAG, 1998, with permission of FAG OEM and Handel AG).

the required rigidity, two angular contact ball bearings in tandem arrangement are fitted at each side.

For mitigating the effect of the temperature rise on the preload level, the design includes two spacer sleeves of approximately equal temperature between the two bearings for axial support of the outer and inner rings of the adjustable arrangement (see Sec. 13.11.3).

Design data
Power: 27 kW
speed: 9000 RPM

Lubrication: The bearings are greased and sealed for the bearing life, and 35% of cavity is filled. Sealing is via labyrinth seals.

Tolerances: High precision spindle bearings are used. The bearings have tight fit on the shaft seat (shaft seat tolerance $+5/-5\ \mu m$), and sliding fit on the housing seats, (housing seat tolerance $+2/+10\ \mu m$).

13.12.3 Bore Grinding Spindle (Fig. 13-10c)

Design data
Power: 1.3 kW
Speed: 16,000 RPM

Lubrication: The bearings are greased and sealed for the bearing life. Sealing is by labyrinth seals.

Design: High rigidity is required. Angular contact ball bearings are used of 15° contact angle for high radial stiffness, and each side has a tandem arrangement for axial stiffness. Bearings have adjustable arrangement and are lightly preloaded by a coil spring.

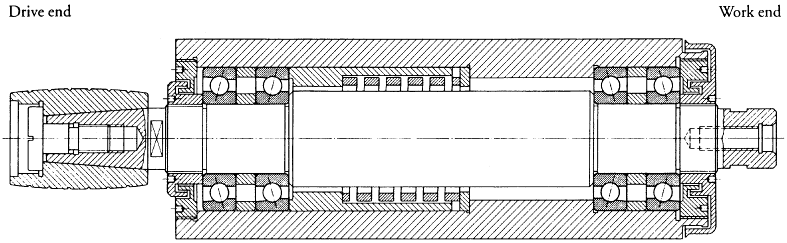


FIG. 13-10c Bore-grinding spindle. (From FAG, 1998, with permission of FAG OEM and Handel AG).

13.12.4 Rough-turning lathe (Fig. 13-10d)

Design data
Power: 75 kW
Speed: 300–3600 RPM

Machining tolerances
seats for the outer ring G6
seats for the inner ring js5

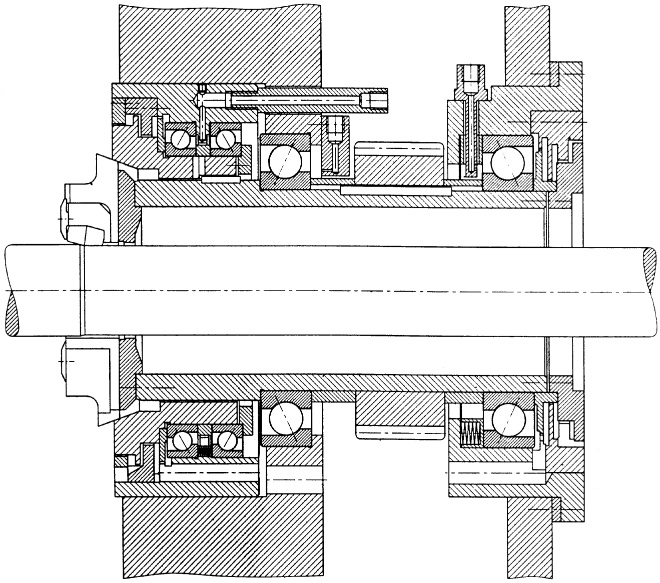


FIG. 13-10d Rough-turning lathe (from FAG, 1998, with permission of FAG OEM and Handel AG).

Lubrication: Oil injection lubrication. A well designed, non-contact labyrinth seal prevents oil leaks and protects the bearings from any penetration of cutting fluid and metal chips.

Design: The bearings have adjustable arrangement and lightly preloaded by springs.

13.12.5 Gearbox (Fig 13-10*e*)

Design data
Power: 135 kW
Speed: 1000 RPM

Tolerances: shaft m5, housing H6

Lubrication: Splash oil from the gears. Shaft seals are fitted at the shaft openings.

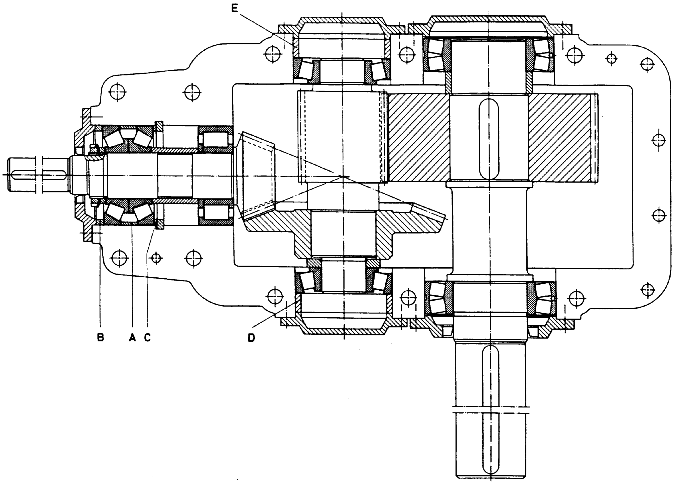


FIG. 13-10*e* Gearbox (from FAG, 1998, with permission of FAG OEM and Handel AG).

13.12.6 Worm Gear Transmission (Fig. 13-10f)

Design data
Power: 3.7 kW
Speed: 1500 RPM

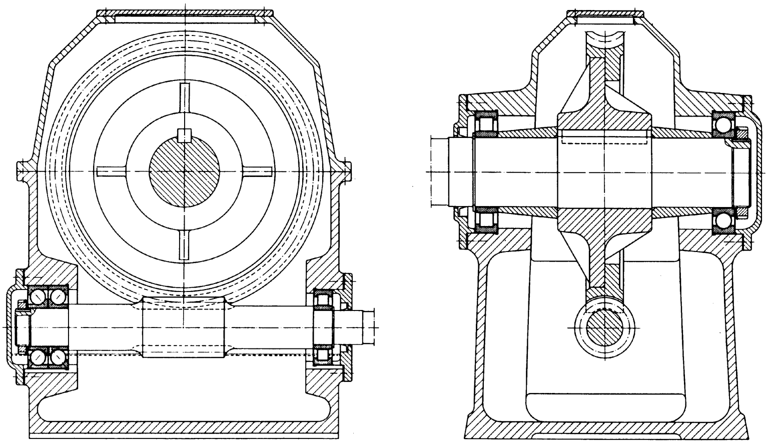


FIG. 13-10f Worm gear (From FAG 1998, with permission of FAG OEM and Handel AG).

Tolerances
Angular contact ball bearing: shaft j5; housing J6
Cylindrical roller bearing: shaft k5; housing J6
Deep groove ball bearing: shaft k5; housing K6

Lubrication: Oil. Contact sealing rings at the shaft opening prevent oil from escaping and protect from contamination.

Design: The two shafts have a locating/floating bearing arrangement.

13.12.7 Passenger Car Differential Gear (Fig. 13-10g)

Design Data
Torque: 160 N-m
Speed: 3000 RPM

Tolerances
Pinion shaft: m6 (larger size bearing)
h6 (smaller size bearing)
housing P7
Crown wheel: hollow shaft r6
housing H6

Lubrication: Gear oil

Design: The turn shafts have adjustable bearing arrangement.

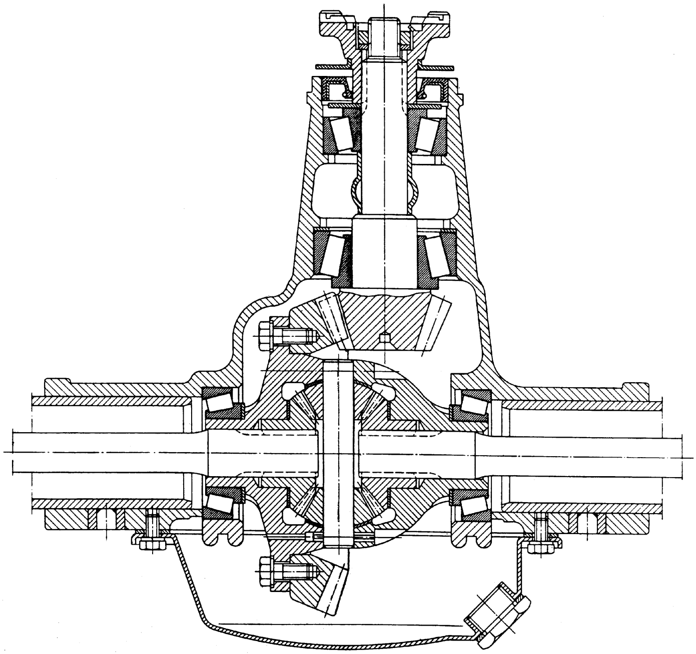


FIG. 13-10g Passenger car differential gear (with permission of FAG OEM and Handel AG).

13.12.8 Guide Roll for Paper Mill (Fig. 13-10h)

Design Data
Speed: 750 RPM
Roll weight: 80 kN
Paper pull force: 9 kN
Bearing load: 44.5 N
Bearing temperature: 105°C

Tolerances: housing G7, inner ring fitted to a tapered shaft

Lubrication: oil circulation

Sealing: double noncontact seal, as shown in Fig 13-10h. The double noncontact seals prevent oil from leaking out.

Design: Special bearings durable to the high operation temperature of the dryer are required. Bearing manufacturers offer high-temperature bear-

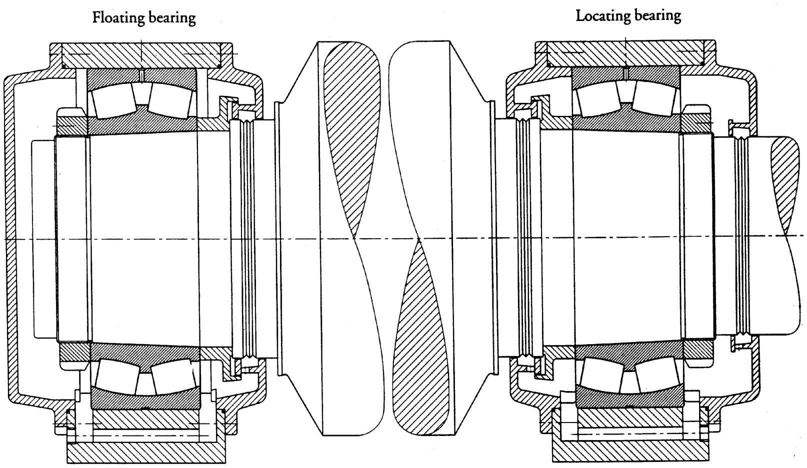


FIG. 13-10h Guide roll for paper mill (from FAG, 1998, with permission of FAG OEM and Handel AG).

ings, which passed special heat treatment, and are dimensionally stable up to 200°C.

Operating clearance is required for preventing thermal stresses, due to the large temperature rise during operation. Also, locating/floating arrangement must be included in this design of relatively high operating temperature. Self-aligning bearings are used to compensate for any misaligning due to thermal distortion.

13.12.9 Centrifugal pump (Fig. 13-10i)

Design Data
Power: 44 kW
Speed: 1450 RPM
Radial load: 6 kN
Thrust force: 7.7 kN

Lubrication: Oil bath lubrication, the oil level should be no higher than the center of the lowest rolling element.

Sealing: Contact seals are used on the two sides. At the impeller side, a noncontact labyrinth seal provides extra sealing protection.

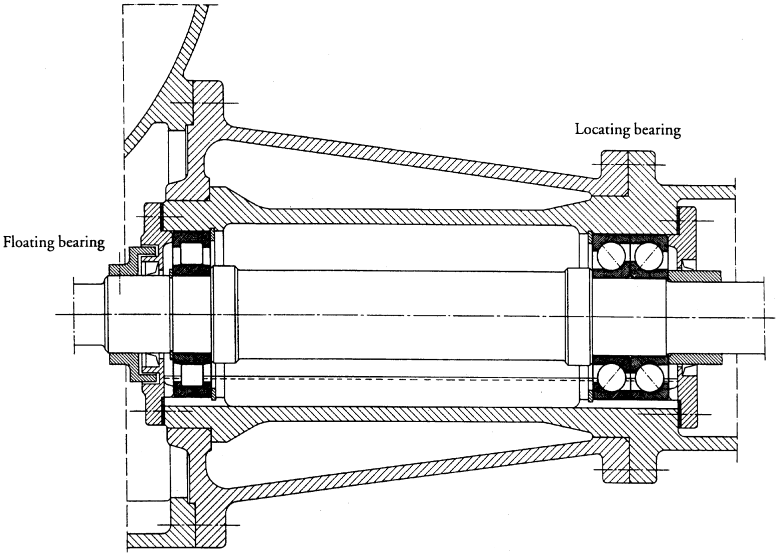


FIG. 13-10i Centrifugal pump (from FAG, 1998, with permission of FAG OEM and Handel AG).

13.12.10 Support Roller of a Rotary Kiln (Fig. 13-10j)

Design Data
Radial load: 1200 kN
Thrust load: 700 kN
Speed: 5 RPM

Tolerances
Shaft n6
Housing H7

Lubrication and Sealing: Grease lubrication with lithium soap base grease. At the roller side, the bearings are sealed with felt strips and grease packed labyrinths.

Design: The bearings are under very high load, and are exposed to a severe dusty environment. Lithium soap base grease is used for bearing lubrication and for sealing. These rollers support a large rotary kiln, which is used in cement manufacturing. Self-aligning spherical roller bearings are used. The two bearings are mounted in a floating arrange-

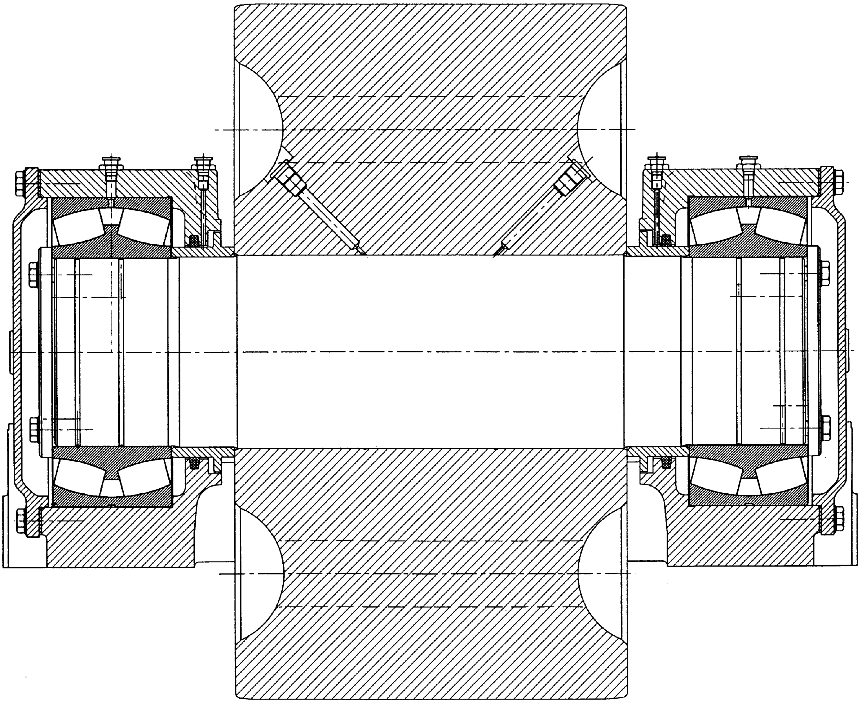


FIG. 13-10j Support roller of a rotary kiln (from FAG, 1998, with permission).

ment (to allow axial adjustment to the kiln). The bearings are mounted into split plummer block housings with a common base.

The grease is fed directly into the bearing through a grease valve and a hole in the outer rings. The grease valve restricts the grease flow and protects the bearing from overfilling. The bearings have double seal of felt strip and grease packed labyrinth. A second grease valve feeds grease directly into the labyrinth seal and prevents penetration of any contamination into the bearings.

The support roller shown in this figure has diameter of 1.6 m and width of 0.8 m. The speed is low, N = 5 RPM and the load on one bearing is high, Fr = 1200 kN. These rollers support the rotary kiln for cement production. The kiln dimensions are 150 m long and 4.4 m diameter. The supports are spaced at 30 m intervals.

13.12.11 Crane Pillar Mounting (Fig. 13-10k)

Design Data
Thrust load: 6200 kN
Radial force: 2800 kN
Speed: 1 RPM

Tolerances: Shaft j6, housing K7

Lubrication and Sealing: Oil bath lubrication with rollers fully immersed in oil.

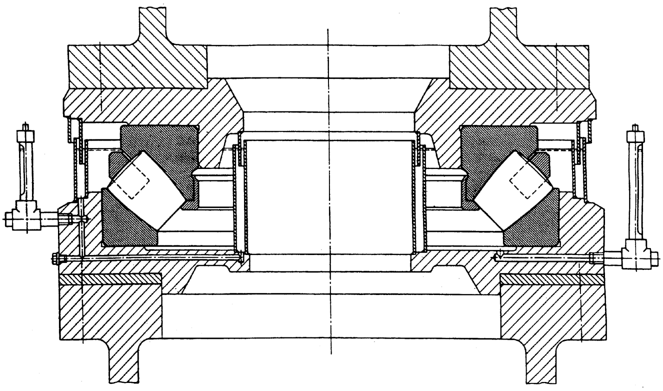


FIG. 13-10k Crane pillar mounting (from FAG, 1998, with permission).

Sealing: Noncontact labyrinth seal as shown in Fig 13-10k. The crane usually operates in severe dust environment. The labyrinth is full of oil, to prevent any penetration of dust from the environment into the bearing.

13.13 SELECTION OF OIL VERSUS GREASE

Greases and oils are widely used for lubrication of rolling-element bearings. In this section, the considerations for selection of oil versus grease are discussed. In addition to considerations directly related to bearing performance, selection depends on economic considerations as well as the ease of maintenance and effective sealing of the bearing.

Whenever possible, greases are preferred by engineers because they are easier to use and involve lower cost. For example, grease lubrication is widely selected for light-and-medium-duty industrial applications, in order to reduce the cost of maintenance. However, at high speeds, considerable amount of heat is generated in the bearings, and greases usually deteriorate at elevated temperatures. In addition, liquid oils improve the heat transfer from the bearing.

Empirical criterion that is widely used by engineers for the selection of oil versus grease is the DN value, which is the product of rolling bearing bore (equal to shaft diameter) in mm and shaft speed in RPM. Rolling bearings operating at DN value above 0.2 million usually require liquid oil, although there are special high-temperature greases that can operate above this limit. Below this limit, both greases and oils can be used. This is an approximate criterion, which considers only the bearing speed for medium loads. In fact, the load, friction coefficient, and heat sources outside the bearing also affect the bearing temperature.

In addition to the DN value, the product of speed and load is used to determine whether the bearing operates under light or heavy-duty conditions. This product is proportional to the bearing temperature rise (see estimation of the temperature rise in Sec. 13.3). The bearing operation temperature must be much lower than the temperature limit specified for the grease.

For low-speed rolling bearings, grease is the most widely used lubricant, because it has several advantages and the maintenance cost is lower. In comparison to oil, grease does not leak out easily through the seals. Prevention of leakage is essential in certain industries such as food, pharmaceuticals and textiles. Tight contact seals on the shafts are undesirable because they introduce additional friction and wear. The advantage of grease is that it can be used in bearing housings with noncontact labyrinth seals. The grease does not leak out, as oil would, and it seals the bearing from abrasive dust particles and a corrosive environment. Rolling bearings are sensitive to penetration of dust, which causes severe erosion, and the bearings must be properly sealed. Section 13.23 presents various types of contact and noncontact seals.

Contact seals are often referred to as *tight seals* or *rubbing seals*. They are tightly fitted on the shaft and are used mostly for oil lubrication. They introduce additional energy losses due to high friction between the seal and the rotating shaft, which raises the bearing temperature. Tightly fitted seals are also undesirable because they wear out and require frequent replacements; they should be avoided wherever possible. Moreover, the shaft wears out due to friction with the seal. In high-speed machines, expensive mechanical seals are often used to replace the regular contact seals. An important advantage of grease lubrication is that noncontact labyrinth seals of low friction and wear can be used effectively. In certain applications, unique designs of noncontact seals are used successfully for oil lubrication (see Sec. 13.12.11).

Grease is particularly effective where the shaft is not horizontal and oils leak easily through the seals. For grease, a relatively simple noncontact labyrinth seal with a small clearance is adequate in most applications.

A very thin layer of grease can be applied on the races to reduce the friction resistance. In such cases, the friction is lower than for oil sump lubrication. Another important advantage of grease is the low cost of maintenance in comparison to oil. Oil requires extra expense to refill and maintain oil levels. In addition, oil can be lost due to leakage, and expensive frequent inspections of oil levels must be conducted in order to prevent machine failure. In comparison, in grease lubrication, there is no need to maintain oil levels, and the addition of lubricant is less frequent. In most cases, grease lubrication results in a lower cost of maintenance.

Economic considerations favor grease lubrication. Oil lubrication systems involve higher initial cost and the long-term bearing maintenance is also more expensive. Therefore, oil is selected only where the selection can be justified based on performance. Oil has several important performance advantages over grease.

1. Unlike grease, oil flows through the bearing and assists in heat transfer from the bearing. This advantage is particularly important in applications of high speed and high temperature.
2. Continuous supply of oil is essential for the formation of an EHD fluid-film. This is very important in high-speed machinery, such as gas turbines.
3. Oil circulation through the bearing has an important function in removing wear debris.
4. Liquid oil is much easier to handle via pumps and tubes, in comparison to grease. In addition, oil is relatively simple to fill and drain; therefore it should be selected particularly when frequent replacements of lubricant are required.
5. In most applications, only a very thin lubrication layer is required. This can be obtained by introducing an accurate slow flow rate of lubricant

(measured in drops per minute) into the bearing. Flow dividers (described in Chapter 10) can be used for feeding at the desired flow rate to each bearing. A precise amount of lubricant at a steady flow rate can be supplied to the bearing and controlled only if the lubricant is oil; this is not feasible with grease.

6. Oil can provide lubrication to all the parts of a machine. An example is a gearbox, where the same oil lubricates the gears as well as the bearings.

As this discussion indicates, grease can be selected for light- and medium-duty applications, whereas oil should be selected for heavy-duty applications in which sufficient flow rates of liquid oil are essential for removing the heat from the bearing and for the formation of a fluid film.

13.14 GREASE LUBRICATION

The compositions and properties of various greases are discussed in Chapter 3. Greases are suspensions of mineral or synthetic oil in soaps, such as sodium, calcium, aluminum, lithium, and barium soaps, as well as other thickeners, such as silica and treated clays. The thickener acts as a sponge that contains and slowly releases small quantities of oil. When the rolling elements roll over the grease, the thickener structure breaks down gradually. Minute quantities of oil release and form a thin lubrication layer on the races and rolling-element surfaces. The lubrication layer is very thin and cannot generate a proper elastohydrodynamic film for separation of the rolling contacts, but it is effective in reducing friction and wear. In addition, the oil layer is too thin to play a role in cooling the bearing or in removing wear debris.

13.14.1 Design of Bearing Housings for Grease Lubrication

The design of the housing and grease supply depends mostly on the temperature, bearing size, load, and speed as well as the environment. The following is a survey of the most common designs.

13.14.1.1 Bearings Packed and Sealed for Life

If the bearing operating temperature is low and its speed and load are not high, the life of the grease can equal or exceed the bearing life. In such cases, using a bearing packed with grease and sealed for life would reduce significantly the maintenance cost. Sealed-for-life small bearings are commonly used under light-duty conditions. Sealed-for-life bearings are also used for occasional operation (not for 24 hours a day), such as in cars, domestic appliances, and pumps for occasional use. Examples are small electric motors for domestic appliances, bearings supporting the drum of a washing machine, and many bearings in passenger cars, such as water pump bearings.

The grease life is sensitive to a temperature rise, and sealed-for-life bearings are not used in machines having a heat source that can raise the bearing temperature. In some applications that involve a moderate temperature rise, such as small electric motors, sealed-for-life bearings with high-temperature grease are used successfully. We have to keep in mind that the life of sealed ball bearings is limited to the lower of bearing life and lubricant life. A method for estimating grease life is presented in Sec. 13.15.

Fig. 13-11 presents an example of the front wheel of a front-wheel-drive car. A double-row angular contact ball bearing is used. Certain cars use angular contact ball bearings or tapered roller bearings that are adjusted. The bearing is packed with grease and sealed on both sides for the life of the bearing.

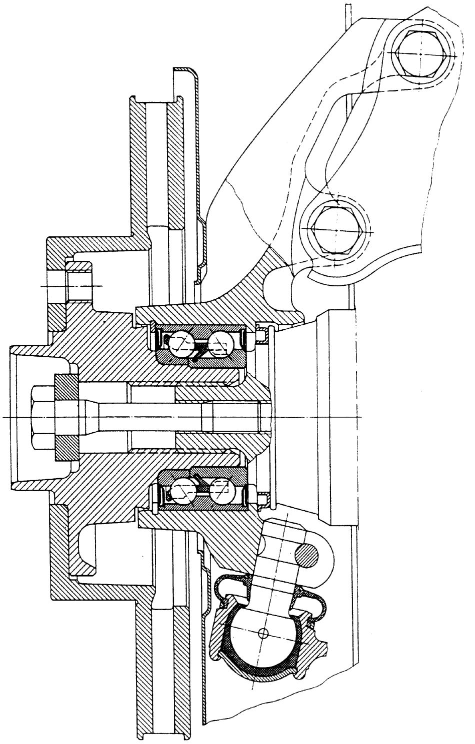


FIG. 13-11 Sealed-for-life bearing in the front wheel of a front-wheel-drive car (from FAG, 1988, with permission).

13.14.1.2 Housing Without Feeding Fittings

For industrial machines that operate for many hours, if the bearings operate at low temperature under light-to-medium duty, the original grease in the housing can last one to two years or even longer. In such cases, the grease can be replaced by new grease only during machine overhauls and the housing is not provided with grease-feeding fittings.

Housing without feeding fittings are used only if there is no heat source from any process outside the bearing and if the bearing operating temperature due to friction is low. This can be applied to light- and medium-duty bearings, namely, where the load and speed are not very high.

The advantage of elimination of any grease fittings is that it prevents overfilling of grease in the housing. The old grease is replaced by new grease only during overhauls, and this can be done manually without using grease guns. Only one-third to one-half the volume of the housing is filled with grease for regular applications. However, to minimize friction in small machines, only a very thin layer of grease is applied on the bearing surfaces, particularly if the drive motor is small and has low power.

Overfilling of grease in the bearing housing results in a high resistance to the motion of the rolling elements and grease overheating, as well as early breakdown of the grease (the grease is overworked). Therefore, the use of high-pressure guns for feeding grease into the housing of rolling bearings is undesirable, particularly for large bearings, because it packs too much grease into the housing and causes bearing overheating. Moreover, feeding under high pressure always results in grease loss.

During the assembly and periodic relubrication, it is very important to keep the bearing and lubricant completely clean from dust or even from old grease. Although less than half the volume of the housing is filled for regular applications, if the bearing is exposed to a severe environment of dust or moisture, the bearing should be fully packed to seal the bearing and prevent its contamination. Grease-feeding fittings are provided for frequent topping-up of grease. In many cases, additional grease fittings feed grease directly to the labyrinth seals (see Sec. 13.12). Fully packed bearings are used only for low- and medium-speed bearings, where the extra friction power loss is not significant.

13.14.1.3 Housings with Feeding Fittings

The common bearing design includes fittings for grease topping-up (adding grease between replacements by grease gun). Although it is desirable not to overfill the housing with grease, this is difficult to avoid. Low-cost maintenance is an important consideration, and in most cases the new grease replaces the old by pushing it out with grease guns. Experience indicates that small, light-duty bearings can operate successfully even when overfilled with grease. Overfilling initially generates extra resistance, but the extra grease is lost over time through the labyrinth seals. The housing is often designed with an outlet hole at the lower side of the housing and noncontact labyrinth seals. The temperature of the overfilled grease rises; after running a few hours (depending on bearing size and grease consistency), the surplus grease escapes through the hole and labyrinth seals. Low-consistency grease is used for this purpose.

If the bearing is exposed to an environment of dust, overfilling prevents contamination of the bearing. Frequent topping-up of grease ensures overfilling, particularly near the seals. The grease fittings must be completely clean before adding grease.

For small and medium-size bearings, it is possible to avoid overfilling and at the same time simplify the grease replacement. This is done via a simple housing design that allows one to force the old grease out completely with the new grease. The design includes a large-diameter drain outlet with a plug, in the side opposite the inlet grease fitting and at the lower side of the housing. This way the grease must pass through the bearing. In order to avoid overfilling, the replacement procedure is as follows: The outlet plug is removed; the shaft is rotating while the new grease is pumped into the housing. The old grease is worked out so that it is easier to replace. The new grease is pumped until it starts to come out of the drain. The shaft rotates for about half an hour to allow the surplus grease to drain out before locking the outlet.

This method is not applicable to large bearings because the pressure of the grease gun is not sufficient to remove all the old grease through the outlet. Also, the bearing might be overfilled, resulting in overheating during operation. Therefore, in large bearings, the grease is replaced manually during overhauls. In addition, large bearings require topping-up of grease at certain intervals, determined according to the temperature and operating conditions. It is important to avoid overfilling during relubrication. The addition of grease is done with grease guns, and it is important to design the housing and fittings to prevent overfilling. These designs involve higher cost and can be justified only for larger bearings.

13.14.2 Design Examples of Bearing Housings

It is important to ensure by appropriate design that during topping-up of grease, the new grease (fed by grease guns) will pass as much as possible through the bearings. The grease is supplied as close as possible to the bearings and discharged through the bearing into the space on the opposite side. In this way, the new grease must pass through the bearing, and the new grease will replace the old grease as much as possible.

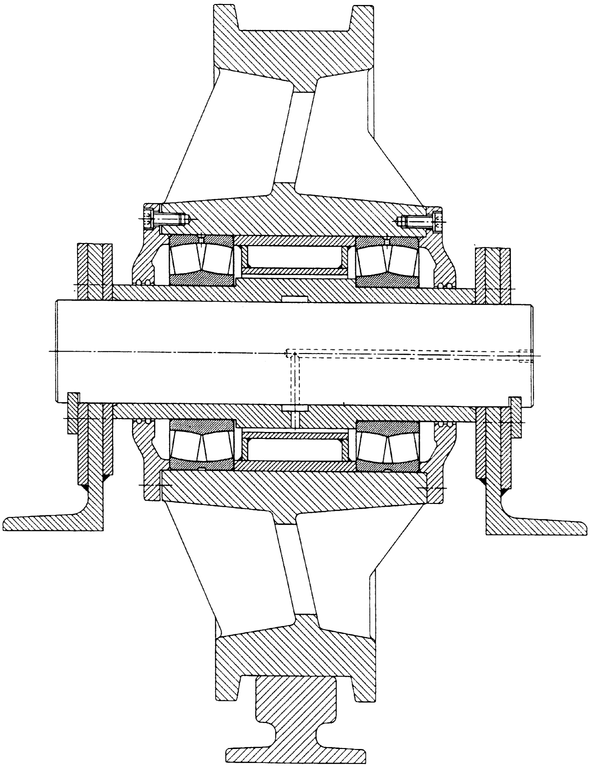


FIG. 13-12 Grease lubrication of crane wheel bearings (from FAG, 1986, with permission of FAG and Handel AG).

13.14.2.1 Crane Wheel Bearing Lubrication

An example of grease lubrication in a crane wheel is shown in Fig. 13-12. The crane wheel runs on a rail. The grease is fed through holes in the stationary shaft between two self-aligning spherical roller bearings. The design limits the grease volume between the two bearings. The grease passes through the two bearings, and the surplus grease is discharged through a double labyrinth seal clearance. Lithium soap base grease is used. The time period between grease replacements is approximately one year.

13.14.2.2 Grease-Quantity Regulators

An example of large bearing housing that is designed for avoiding overfilling during relubrication by grease guns is shown in Fig. 13-13. This design is widely used for large electric motors (SKF, 1992). The grease is fed at the bottom of the housing, near the left side of the outer ring. The design of the housing includes

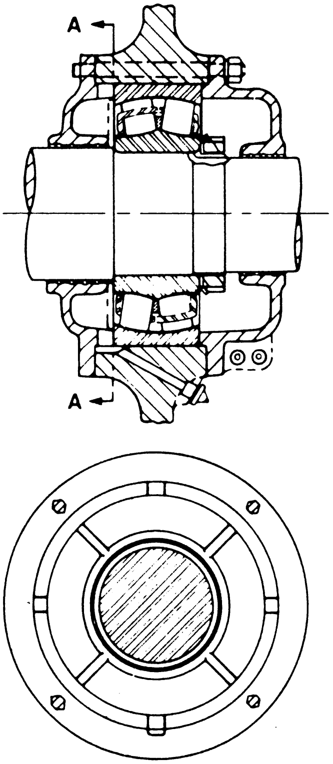


FIG. 13-13 Bearing housing for a large electric motor (from SKF, 1992, with permission).

radial ribs inside the left cover of the housing. They direct the new grease into the bearing without overfilling the space on the left side of the bearing. The old grease escapes through the bearing into the large space at the right side of the bearing. The ribs also keep the grease in place and prevent it from being worked by the rotating shaft during regular operation. In this way, the ribs prevent overheating. In this design, the cover is split to simplify the removal of the old grease during overhauls.

13.14.2.3 Grease Chamber

Another method that prevents overfilling of a bearing is shown in Fig. 13-14. It uses a double-sealed, prelubricated bearing. The concept is that only one side of the bearing housing is full of grease (fed by a grease gun). The advantage of this method is that only a small quantity of grease is gradually released from the

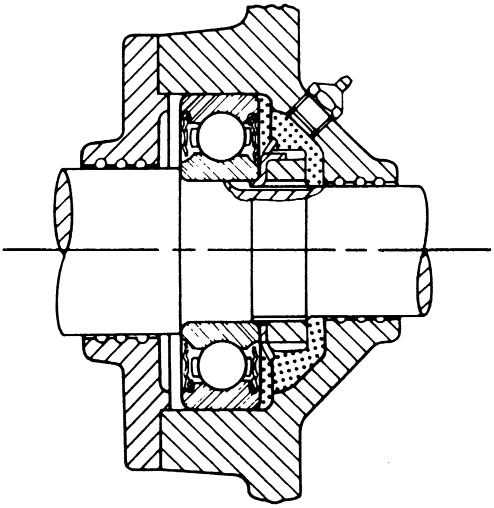


FIG. 13-14 Grease chamber for double-sealed bearings (from SKF, 1992, with permission).

grease packing and penetrates into the bearing. This design of a double-sealed bearing combined with noncontact labyrinth seals protects the bearing from dust.

13.14.2.4 Dust Environment

Small bearings in a dusty environment are fully packed with grease. However, for large bearings, it is important to prevent overfilling with grease, which results in overheating and early failure of the bearing.

An example of a double-shaft hammer mill for crushing large material (FAG, 1986) exposed to a severe dust environment is shown in Fig. 13-15. This example combines a design for a grease-quantity regulating disk that prevents overfilling and a separate arrangement for packing the grease between the labyrinth and felt seals.

13.14.2.5 Regulating Disk

The bearing housing design consists of a regulating disk that rotates together with the shaft. It is mounted at the side opposite the grease inlet side. If the grease quantity in the bearing cavity is too high, the rotating disc shears and softens part of the grease. By centrifugal action, the grease drains through the radial clearance into the volume between the disk and seals, as shown in Fig. 13-15.

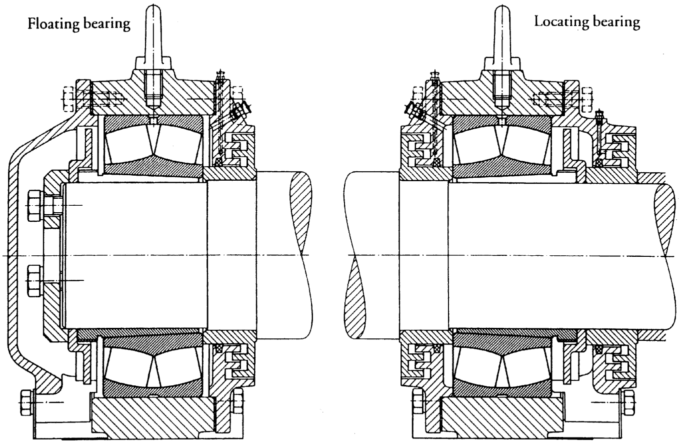


FIG. 13-15 Bearing housing of a double-shaft hammer mill (from FAG, 1986, with permission of FAG and Handel AG).

13.14.2.6 Sealing in a Severe Dust Environment

In Fig. 13-15, grease is fed directly into the bearing through a grease valve and a hole in the outer rings. The bearings have a double seal of felt strip and grease-packed labyrinth. A second grease valve feeds grease between the labyrinth and the felt seal. Frequent relubrication of the grease in the labyrinth seal prevents penetration of abrasive dust particles into the bearing.

13.15 GREASE LIFE

The life of greases and oils is limited due to oxidation. High temperature accelerates the oxidation rate, and the life of greases and oils is very sensitive to a temperature rise. An approximate rule is that grease life is divided by 2 for every 15°C (27°F) temperature rise above 70°C (160°F). In addition to oxidation, the bleeding of the oil from the grease and its evaporation limit the life of the grease at high temperature.

At low operating temperature, the life of the grease is long, and bearings packed with grease and sealed for life are widely used. Adding fresh grease to sealed-for-life bearings is not necessary, because the life of the grease is longer than the bearing life.

If the life of the grease is shorter than the life of the bearing, the grease should be replaced. Since it is difficult to precisely predict the grease life, fresh grease should be added much before the grease loses its effectiveness. The time period between lubrications (also referred to as relubrication intervals) is a function of many operating parameters, such as temperature, grease type, bearing type and size, speed, and grease contamination. The time period, Δt, between grease replacements is determined empirically. It is based on the requirement that less than 1% of the bearings not be effectively lubricated by the end of the period.

In Fig. 13-16, curves are presented of the recommended time period Δt (in hours) as a function of bearing speed N (RPM) and bearing bore diameter d (SKF, 1992). The charts are based on experiments with lithium-based greases at temperatures below 70°C (160°F). For higher temperatures, the time period Δt is divided by two for every 15°C (27°F) of temperature rise above 70°C (160°F). However, the temperature should never exceed the maximum temperature allowed for the grease. In the same way, the time period Δt can be longer at temperatures lower than 70°C (160°F), but Δt should not be more than double that obtained from the charts in Fig. 13-16. Also, one should keep in mind that at very low temperatures, the grease releases less oil.

The time period Δt between grease replacements is a function of the bearing speed N (RPM), and bearing bore diameter d (mm), and bearing type. According to the bearing type, the time period Δt is determined by one of the following scales.

Scale a: is for radial ball bearings.
Scale b: is for cylindrical and needle roller bearings.
Scale c: is for spherical roller bearings, tapered roller bearings, and thrust ball bearings.

Figure 13-16 is valid only for bearings on horizontal shafts. For vertical shafts, only half of Δt from in Fig. 13-16 is applied. The maximum time period between grease replacements, Δt should not exceed 30,000 hours. Bearings subjected to severe operating conditions, such as elevated temperature, high speed, contamination, or humidity, must have more frequent grease replacements. Under severe conditions, the best way to determine the time period between grease replacements is by periodic inspections of the grease.

The following cases require shorter periods between lubrications:

1. Full-complement cylindrical rolling bearing, 0.2 Δt (in scale c)
2. Cylindrical rolling bearing with a cage, 0.3 Δt (in scale c)
3. Cylindrical roller thrust bearing, needle roller thrust bearing, spherical roller thrust bearing. 0.5 Δt (in scale c)

Experience has indicated that large bearings, of bore diameter over $d = 300\,\text{mm}$, need more frequent grease replacements than indicated in Fig. 13-16 (the large bearings are marked by dotted lines). Frequent grease replacements are required if there are high contact stresses, high speed and high temperature. Whenever the time period between grease replacements is short, a continuous grease supply can be provided via a grease pump and a grease valve. For a continuous grease supply, the grease mass per unit of time, G, fed into the

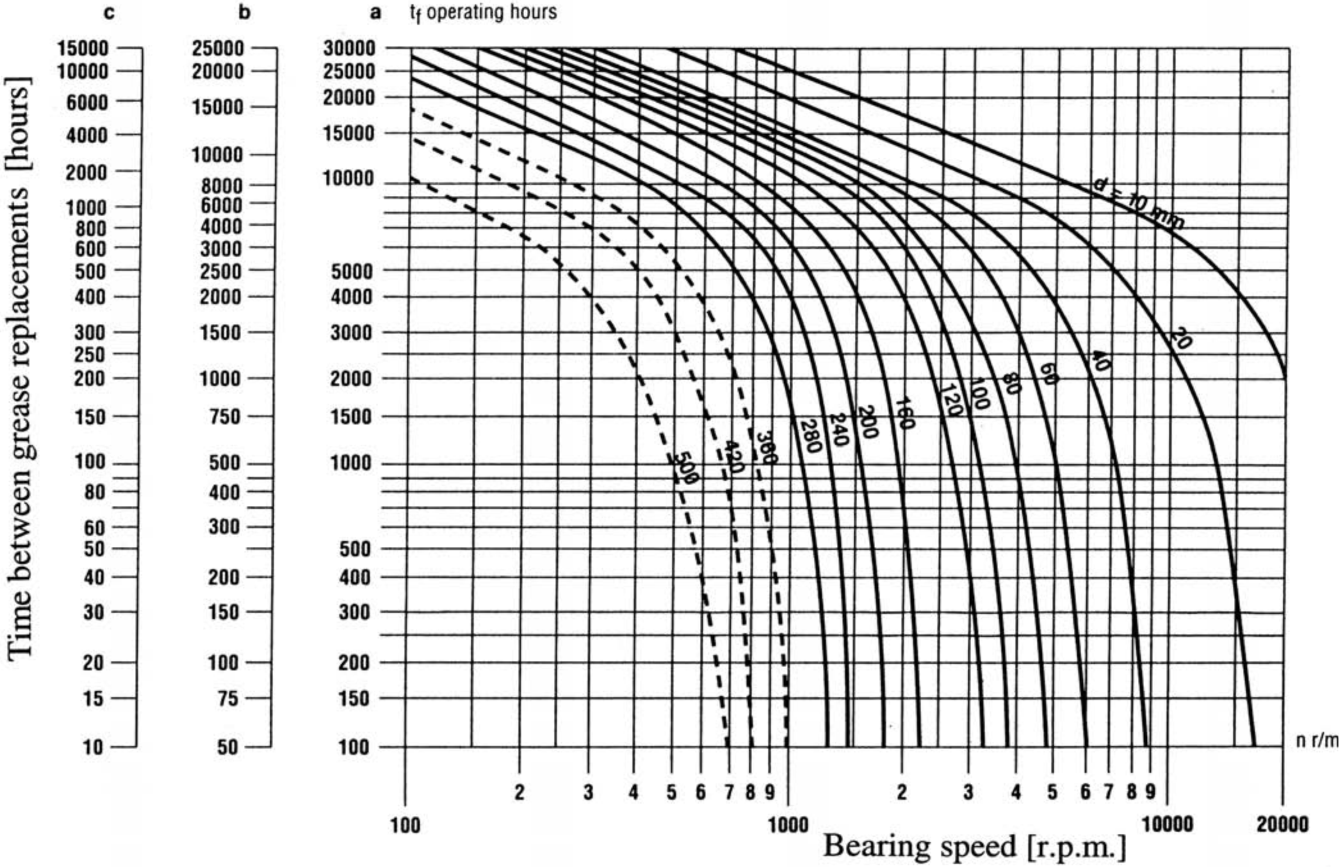


FIG. 13-16 Time between grease replacements, Δt (in hours) (from SKF, 1992, with permission from SKF, USA).

large bearing is determined by an empirical equation (SKF, 1992). The following empirical equation is for regular conditions, without any conduction of external heat into the bearing (the bearing temperature is only due to friction losses):

$$G = (0.3 - 0.5)DL \times 10^{-4} \tag{13-29}$$

Here,

$G =$ continuous mass flow rate supply of grease *(g/h)*
$D =$ bearing OD (mm)
$L =$ bearing width (mm) [for thrust bearings use total height, H]

13.15.1 Topping-Up Intervals

In applications where the grease life is considerably shorter than the bearing life, either complete replacements (relubrication) or more frequent applications of topping-up grease (by grease guns) are required. Topping-up grease is much faster and it is preferred whenever possible. In most cases, during topping-up, the fresh grease replaces only part of the used grease, and more frequent applications are needed in comparison to complete grease replacements. The initial filling and subsequent topping-up and complete replacement of grease (after cleaning at main overhauls) is done as follows (SKF, 1992):

1. If the period between grease replacements, Δt (in hours) is less than 6 months of machine operation, the grease is topped-up at half the recommended Δt from Fig. 13.6. After three periods of topping-up, all grease is replaced by fresh grease.
2. If the period between grease replacements, Δt (in hours) is equivalent to more than 6 months of machine operation, topping-up should be avoided, and all the grease in the housing is replaced with fresh grease after each period.

13.15.2 Topping-Up Quantity

In the topping-up procedure, the grease in the bearing housing is only partially replaced by adding a small quantity of fresh grease after each period. The recommended grease quantity to be added can be obtained from the following empirical equation (SKF, 1992):

$$G_p(g) = 0.005D(\text{mm}) \times L(\text{mm}) \tag{13-30}$$

Here,

$G_p =$ grease mass quantity to be added (grams)
$D =$ bearing OD (mm)
$L =$ total bearing width (mm) [for thrust bearings use total height, H]

13.16 LIQUID LUBRICATION SYSTEMS

Oil lubrication can be provided by several methods. For low and moderate speeds, an oil bath, also called an *oil sump*, is used. For low speeds, the oil level in an oil bath is the center of the lower rolling element. For heavy-duty large bearings cooling is necessary, and the oil is circulated in the oil bath. If the oil level is the center of the lower rolling element, it is referred to as a *wet sump*; if all the oil is drained, it is referred to as a *dry sump*. The level is determined by the height of the outlet. A pump feeds the oil through flow dividers to the bearing housing. The oil can be supplied also by gravitation. The major advantage of circulation lubrication is that it can cool the bearings. Circulation lubrication of many bearings is relatively inexpensive.

An additional method is mist lubrication. In this method, the oil is not recovered. The most important advantage is that the lubrication layer is very thin. It results in low viscous resistance to the motion of the rolling elements. For example, mist lubrication is used for machine tool spindles.

Several examples of the various methods of oil lubrications follow.

13.16.1 Bearing Housing with Oil Sump

Oil lubrication requires a special design of the bearing housing, often referred to as a *pillow block*. Various standard designs of pillow blocks are available from bearing manufacturers. It is possible to select a design based on the optimal oil level and rate of flow of lubricant that is appropriate for each application. For large bearings, a welded housing is less expensive than a cast housing.

An example is the housing of the propeller-ship shaft bearing shown in Fig. 13-17. In this example, the speed is 105 RPM and the shaft diameter is 560 mm. Contact seals protect the bearing from the corrosive seawater. The oil can be fed by circulation lubrication, and the pressure in the housing is kept above ambient pressure to prevent penetration of seawater.

In this arrangement, the fluid level is relatively high, and it can be applied only when the bearing speed is low. In order to minimize the viscous resistance at high speed, the oil level must be lower. For low speeds, the oil level should not be above the center of the lowest rolling element; but this level is too high for high-speed bearings. A drain is always provided for oil replacement.

The oil level is preferably checked when the machine is at rest, when all the oil is drained into the reservoir. There are always oil losses, and a sight-glass gauge is usually provided for checking oil level; oil is added as soon as the oil level is low. This method requires much individual attention to each bearing, and it can be expensive in manufacturing industries where a large number of bearings are maintained.

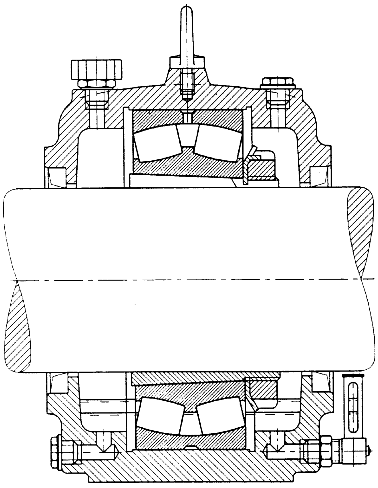


FIG. 13-17 Bearing housing of a ship shaft (from FAG, 1998, with permission of FAG and Handel AG).

13.16.2 Lubrication with Wick Arrangement

A better design for feeding a very low flow rate of oil into the bearing is the wick feed arrangement. A design for a vertical shaft is shown in Fig. 13-18. The wick siphons oil from a reservoir into the bearing. An important advantage is that the wick acts like a filter and supplies only clean oil to the bearing (solid particles are not siphoned). Viscous friction is minimized by this arrangement. The wick continues to deliver oil even when the machine is not operating.

An improved design where oil is fed only during bearing operation is shown in Fig. 13-19. A wick provides lubricant by capillary attraction to a rotating bearing. The bearing is above the fluid level, and the wick must be in contact with the collar for proper function of this arrangement. The oil is thrown off by centrifugal force, and the oil is continually siphoned. This system delivers oil only when required, i.e., when the bearing is rotating. The oil is drained back into the reservoir without losses.

Wick feed has an important advantage where the bearing operates at high speeds, because it can supply a continuous low flow rate of filtered oil to the bearing. With this wick feed system, there is no resistance to the motion of the rolling elements through the oil reservoir. For effective operation, the wicks should be properly maintained; they have to be replaced occasionally. During servicing, the wick should be dried and thoroughly saturated with oil before reinstallation. This prevents absorption of moisture, which would impair the oil-siphon action.

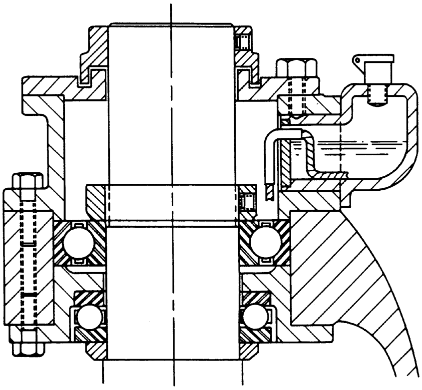


FIG. 13-18 Bearing housing with a wick for oil feeding (from SKF, 1992, with permission).

13.16.3 Oil Circulating Systems

There are several benefits in using oil circulation systems for rolling bearings, where a monitoring pump supplies a low flow rate of oil to each bearing. In certain applications, particularly in hot environments, the oil circulation plays an important role in assisting to transfer heat from the bearing. In addition, a circulating system simplifies maintenance, particularly for large industrial machines with many bearings. For oil circulation, a special design of the housing is used for controlling the oil level.

An example of a bearing housing for oil circulation is shown in Fig. 13-20. The level of the oil in the housing is controlled by the height of the outlet. For a

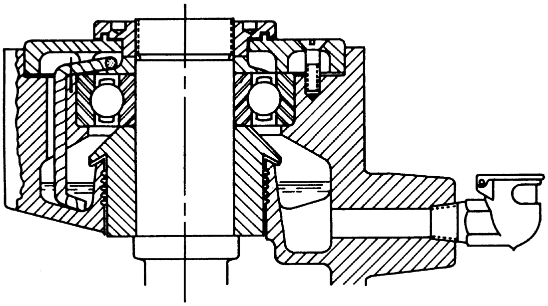


FIG. 13-19 Bearing housing with a wick and centrifugal oil feeding (from SKF, 1992, with permission).

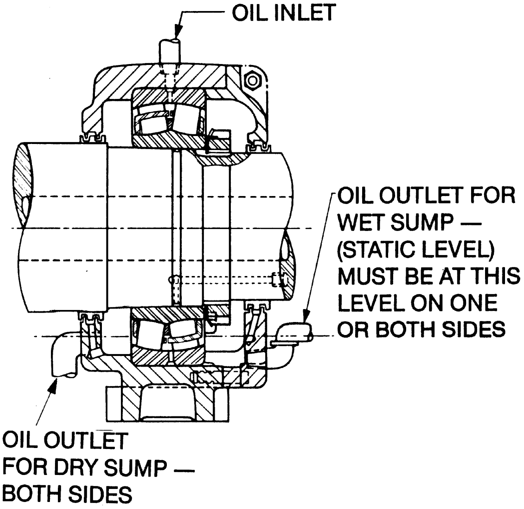


FIG. 13-20 Oil circulation for pulp and paper dryers (from SKF, 1992, with permission from SKF, USA).

wet sump, the oil level at a standstill should not be higher than the center of the lowest ball or roller. A sight-glass gauge is usually provided for easy monitoring.

As mentioned earlier, high-speed bearings require a dry sump, where the oil drains completely after passing through the bearing. In addition, a dry sump is used for bearings operating at high temperature because the lubricant must not be exposed for long to the high temperature (to minimize oxidation). For a dry sump, two outlets are located at the lowest points on both sides of the housing, as shown on the left side of Fig. 13-20.

For applications where bearing failure must be avoided at any cost, oil circulation systems require an automatic monitoring to indicate when oil flow is blocked through any bearing. Safety measures include electrical interlocking of the oil pump motor with the motor that is driving the machine.

13.16.4 Oil Mist Systems

This arrangement entails lubrication by a mixture of air and atomized oil. An atomizer device forms the oil mist. In order to have the required quantity of oil and appropriate viscosity at the bearings' rolling contacts, oil mist system manufacturers provide recommendations for system designs, capacities, and operating temperatures and pressures.

The bearing operating temperature is reduced by this method of lubrication, by means of air cooling. A thin oil layer is formed on the bearing surfaces due to

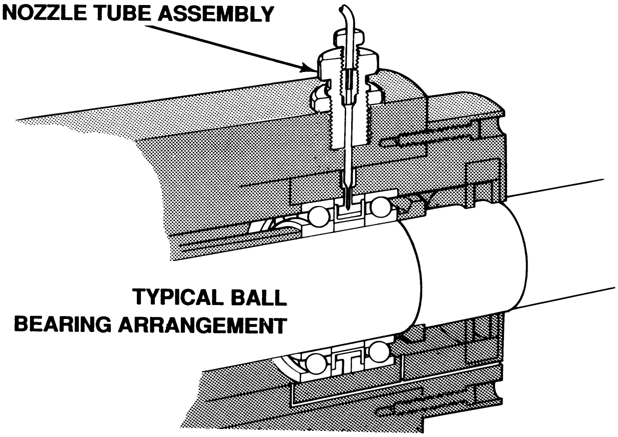


FIG. 13-21 Nozzle assembly of oil mist system. (Reprinted with permission from Lubriquip Inc.)

the air flow, which prevents accumulation of excess oil. The air is supplied under pressure, and it prevents moisture from the environment from penetrating into the bearing. An additional advantage is that oil mist lubrication supplies clean, fresh oil into the bearings (the oil is not recycled). These advantages increase the life expectancy of the bearing. Although the oil in the mist is lost after passing through the bearing, very little lubricant is used, so oil consumption is relatively low. The connection of the nozzle assembly in the bearing housing is shown in Fig. 13-21.

In Fig. 13-22, a mist lubrication system is shown that is widely used for grinding spindles. The air, charged with a mist of oil, is introduced in the housing

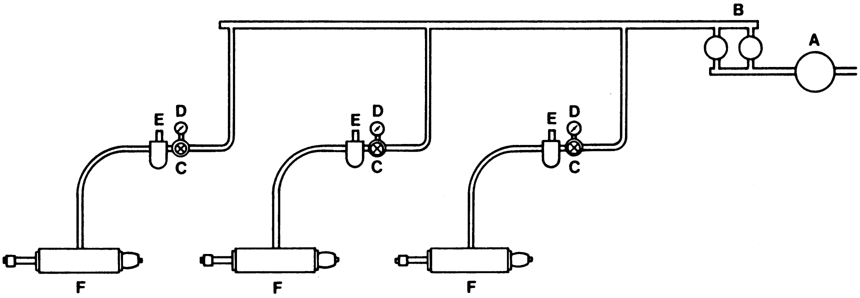


FIG. 13-22 Oil mist system for machine tool spindles (from SKF, 1992, with permission of SKF).

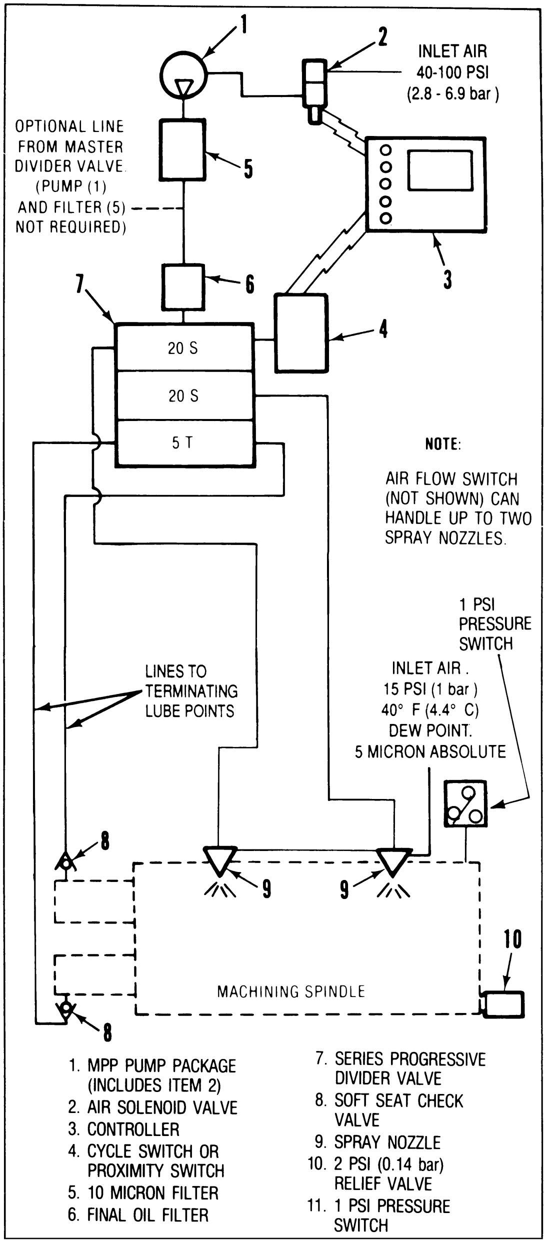


FIG. 13-23 Control of advanced oil mist system with flow dividers (reprinted with permission of Lubriquip).

between the bearings in order to ensure that the air passes through the bearings before escaping from the housing. Air from the supply line passes through a filter, B, then through a pressure reduction valve, D, and then through an atomizer, E, where the oil mist is generated. The air must be sufficiently dry before it is filtered, and a dehumidifier, A, is often used.

Advanced oil mist systems with precise control of the flow rate are often used in machining spindles. The systems include a series of flow dividers and an electronic controller. A schematic layout of a controlled system is shown in Fig. 13-23.

13.16.5 Lubrication of High-Speed Bearings

In bearings operating at very high speeds (high DN value) a considerable amount of heat is generated, and jet lubrication proved to be effective in transferring the heat away from the bearing, see a survey by Zaretzky (1997). Jet lubrication is used for high-speed bearings aircraft engines. Several nozzles are placed around the bearing, and the jet is directed to the rolling elements near the contact with the inner race. The centrifugal forces move the oil through the bearing for cooling and lubrication. Experiments have shown that in small bearings jet lubrication can be used successfully at very high speeds of 3 million DN, and speeds to 2.5 million DN for larger bearing of 120 mm bore diameter.

A more effective method of lubrication for very high-speed bearings is by means of under-race lubrication, see Zaretzky (1997). The lubricant is fed through several holes in the inner race. In addition, the lubricant is used for cooling in clearances (annular passages) between the inner and outer rings and their seats.

13.16.6 Oil Replacement in Circulation Systems

The time period between oil replacements depends on the operating conditions, particularly oil temperature, and the amount of contamination that is penetrating into the oil as well as the quantity of oil in circulation. In most cases, the reason for frequent oil replacements is the oxidation of the oil due to elevated temperatures or the penetration of dust particles into the oil.

If the bearing temperature is below 50°C (120°F) and the bearing is properly sealed from any significant contamination, the life of the oil is long and intervals of one year are adequate. At elevated temperatures, however the oil life is much shorter. For similar operating conditions, if the oil temperature is doubled and reaches 100°C (220°F), the oil life is reduced to only 3 months (a quarter of the time for 50°C (120°F).

In central lubrication systems, the oil is fed from an oil sump through a filter and than passes through the bearing and returns to the oil sump. In order to reduce the oil temperature, the system can include a cooler. There are many variable operating conditions that determine the oil temperature, including the rate of flow of the circulation and the presence of a cooling system, which reduces the oil temperature. Since there are many operating parameters, it is difficult to set rigid rules for the lubrication intervals. It is recommended to test the oil frequently for determining the optimum time period for oil replacement. The tests include measurement of the oxidation level of the oil, the amount of antioxidation additives left in the oil, and the level of contamination by dust particles.

13.17 HIGH-TEMPERATURE APPLICATIONS

In cases where heat is transferred into the bearings from outside sources, cooling of the oil in circulation is necessary to avoid excessive bearing temperatures and premature oxidation of the lubricant. Examples are combustion processes (such as car engines) and steam dryers. In addition, high temperatures reduce the viscosity and effectiveness of the oil. Various methods for controlling the oil temperature are used. In Fig. 13-24, a cooling disc is shown that is mounted on the shaft between the bearing and the heat source. The disc increases the convection area of heat transferred from the shaft (SKF, 1992).

An improved cooling system is shown in Fig. 13-25. It is a design of a pillow block with water-cooling coils. Water-cooled copper coils transfer the heat away from the oil reservoir in the pillow block. It is important to shut off the cooling water whenever the machine is stationary in order to prevent condensation, which generates rust.

Air is also used for cooling bearings. A direct stream of fresh air is usually created through the use of fans, blowers, or air ducts around the bearing that can

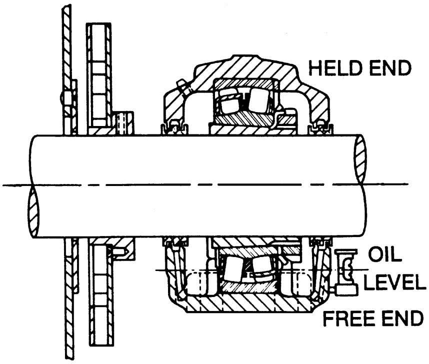


FIG. 13-24 Cooling disc mounted on the shaft (from SKF, 1992, with permission).

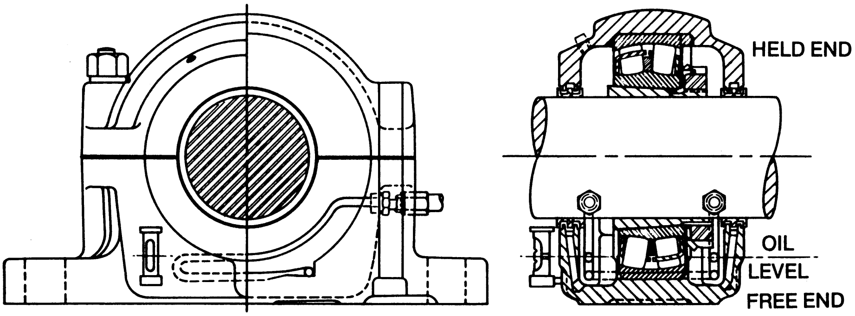


FIG. 13-25 Pillow block with water cooling (from SKF, 1992, with permission).

help in dissipating the heat. An additional method that is widely used is to cool the oil outside the bearing, via a heat exchanger. The circulating oil can pass through a radiator for cooling, such as in car engine oil circulation.

13.17.1 Moisture in Rolling Bearings

Lubricants do not completely protect the bearing against corrosion caused by moisture that penetrates into the bearing. In particular, the combined effect of acids (products of oxidation) and moisture are harmful to the bearing surfaces.

The design of bearing arrangement and lubrication systems must ensure that the bearing is sealed from moisture. Certain lubricants can reduce moisture effects, such as compound oils, which are more water repellent than regular mineral oils. Lithium-based greases are good water repellents and also provide an effective labyrinth seal. In all cases, the lubricant should completely cover the bearing surface to protect it. Nonoperating machines should be set in motion periodically in order to spread the lubricant over the complete bearing surfaces for corrosion protection.

13.18 SPEED LIMIT OF STANDARD BEARINGS

The standard bearing has a much lower speed limit than special steels. Bearing manufacturers recommend a speed limit to their standard bearings. The DN value is widely used for limiting the speed of various rolling bearings. This is defined as the product of bearing bore in mm and shaft speed in RPM.

The friction power loss in a rolling bearing is proportional to the rolling velocity, which is proportional to the bearing temperature rise above ambient temperature. The centrifugal force of the rolling elements is also a function of the DN value. Special steels have been developed for aircraft turbine engines that can operate at very high speeds of 2 million DN. There is continuous search for better materials, such as the introduction of silicon nitride rolling elements, and unique designs (see Chapter 18) to allow a breakthrough past the limit of 2 million DN.

However, for standard bearings, made of SAE 52100 steel, the maximum DN value is quite low, of the order of magnitude of 0.1 million. The reason for limiting the DN value of industrial bearings is in order to limit the temperature rise and, thus, to extend the fatigue life of the bearings.

Bearing manufacturers recommend low limits of the DN values. The speed limits for various bearing types can be obtained from Fig. 13-26. These limits are based on a temperature limit of 82°C (180°F) as measured on the outside bearing diameter. Standard steel at higher temperature starts to lose its hardness and fatigue resistance at that temperature. Standard bearing steel, SAE 52100, can operate at higher temperatures, up to 177°C (350°F). However, the bearing life (as well as lubricant life) is lower.

Figure 13-26 shows that the speed limit of standard bearings is quite low. In Sec. 13.19, special steels are discussed that are used for much higher speeds.

13.19 MATERIALS FOR ROLLING BEARINGS

In the United States, the standard steel for ball bearings is SAE 52100 (0.98% C, 1.3% Cr, 0.25% Mn, 0.15% Si). It is widely used for the rings and rolling elements of standard ball bearings as well as certain roller bearings. SAE 52100 is of the through-hardening type of steel. This steel can be hardened thoroughly to Rockwell C 65. In general, steels with carbon content above 0.8%, combined with less than 5% of other alloys, are of the *hypereutectoid* type, where the cross section of the rings can be hardened thoroughly.

However, large bearings with a large cross-sectional area of the rings are made of case-hardening (carburizing) steels. An example of a widely used case-hardening steel is SAE 4118 (0.18% C, 0.4% Cr, 0.4% Mn, 0.15% Si, 0.08% Mo). Case-hardening steels contain less than 0.8% carbon and are of the *hypoeutectoid* type. This means that they must be diffused with additional carbon in order to be hardened by heat treatment. The advantage of a case-hardening steel is that it is less brittle, because only the surface is hardened while the inside cross section remains relatively soft. In comparison, the through-hardening steels have high hardness over the complete cross section.

Rolling bearings made of these two types of steel can be used only at low temperature (below 350°F or 177°C). Above this temperature, these steels lose their hardness. For applications at higher temperatures, high-alloyed steels have been developed that maintain the required hardness at high temperature. Examples of special steels that provide better fatigue resistance at high temperatures appear in Sec. 13.19.2.

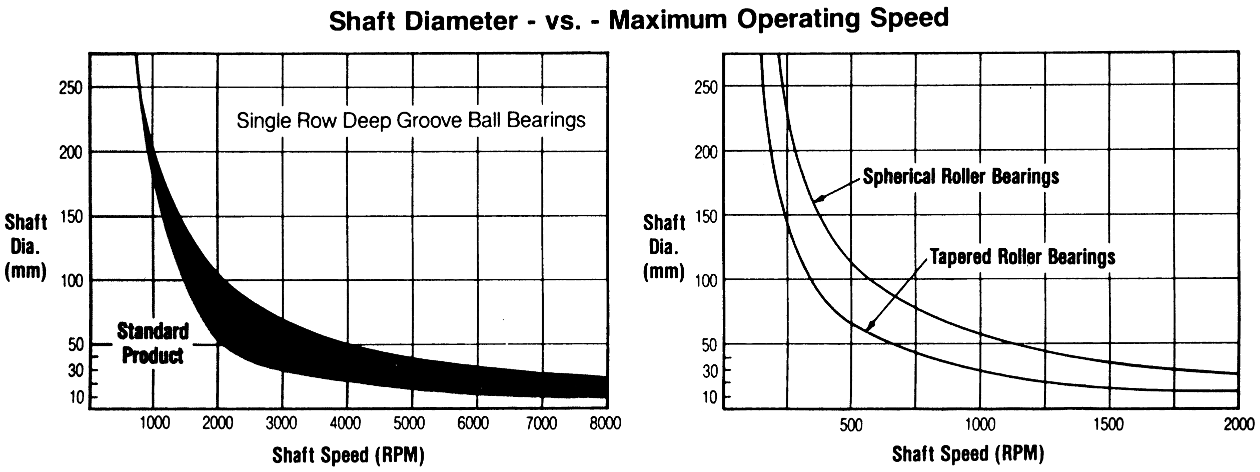


FIG. 13-26 Speed limit of standard bearings (from SKF, 1992, with permission).

13.19.1 Stainless Steel AISI 440C

AISI 440C (1.1% C, 17% Cr, 0.75% Mn, 1% Si, 0.75% Mo) is a high-carbon stainless steel for rolling bearings. AISI 440C does not contain nickel and can be heat-treated and hardened to Rockwell C 60. In the United States, it became a standard stainless steel bearing material that is widely used in corrosive environments, particularly in instruments. A major disadvantage of this steel, in comparison to SAE 52100, is its shorter fatigue life. Therefore, for heavy loads, it is used only where there is no other way to protect the bearing from corrosion. AISI 440C is widely used in instrument ball bearings that must be rust free and where corrosion resistance is much more important than the load capacity. For certain applications, it is possible to combine the characteristics of corrosion resistance and high fatigue resistance by using chrome-coated bearings made of the standard SAE 52100 steel.

13.19.2 Special Steels for Aerospace Applications

For most applications, the preceding two types of steels provide adequate performance. For aerospace applications, however, there is a requirement for fatigue resistance for high-speed bearings operating at elevated temperatures in turbine engines. Special high-alloy-content steels were developed as well as higher purity by using better manufacturing processes such as vacuum induction melting (VIM) and vacuum arc remelting (VAR). The piston engine bearings of early aircraft used tool steels such as M_1 and M_2. During the 1950s, the turbine engine aircraft has been developed, and there was a requirement for better rolling-element bearings that can resist the high speed and high temperature in the aircraft turbine engine. For this purpose, the vacuum melting process was developed and used with high-alloy-content steel AISI M-50 and much later, the recently introduced casehardened steel M50NiL. These bearings are also used for other applications of high speed and elevated temperatures.

An interesting survey by Zaretszky (1997b) shows the major breakthroughs, which resulted in bearing fatigue-life improvement of approximately 200 times, between 1948 and 1988. The most important developments are high purity steel processing, composition of special steels, ultrasonic inspection techniques, improvement of bearing design, and better lubrication. After World War II, the requirement for reliable operation of jet engines and helicopter rotors was the major drive for research and development, which resulted in impressive improvements in the performance of rolling-element bearings for aerospace applications.

13.19.2.1 M50 Bearing Material for Aerospace Applications

AISI M-50 (0.8% C, 4% Cr, 0.1%Ni, 0.25% Mn, 0.25% Si, 4.25% Mo) was developed in the 1950, and it is used for rolling bearings in aerospace applications. In addition, it has industrial applications for rolling bearings operating at elevated temperatures up to 315°C (600°F). AISI M-50 is through-hardening steel, because it has relatively high carbon content. This material demonstrated significant improvement in fatigue life, in comparison to the earlier steels. However, the high demand in aircraft engines, with fatigue combined with high temperature and high centrifugal forces, can result in the initiation of cracks and even complete fracture of rings made of through-hardening steels such as M-50. For that reason, the speed of aircraft engines has been limited to 2.4 million DN.

In order to break through this limit, a lot of research has been conducted to improve bearing materials. The recent development (during the 1980s) of high-alloyed casehardened steel M50NiL significantly improved the fatigue resistance of jet engine bearings.

13.19.2.2 M-50NiL Bearing Steel for Aerospace Applications

During the 1980s, M50NiL has been developed and introduced into high-speed aerospace applications. M50NiL is casehardened steel, which has a softer core, and it is less brittle than the through-hardened steel AISI M-50. In turn, M50NiL has improved fracture toughness, better fatigue resistance, better impact resistance in high-speed bearings (and gears), and can operate at high temperatures similar to AISI M-50. Therefore, M50NiL gradually replaces AISI M-50 as the material of choice for jet engine bearings in aircraft.

M50NiL (0.15% C, 4% Cr, 3.5% Ni, 0.15% Mn, 1% V, 4.0% Mo) differs from AISI M-50 by its lower carbon content. M50NiL requires carburizing for getting hard surfaces. The low carbon content makes it casehardened steel with softer and less brittle material inside the cross section. M50NiL has less carbon and more nickel and vanadium in comparison to AISI M-50. These alloys increase hot hardness and form hard carbides that reduce wear. M50NiL has uniformly distributed carbides, which is less likely to initiate fatigue cracks.

The most important advantage of M50NiL is that it is casehardened steel with optimum fatigue properties under rolling contact. In rolling contact fatigue tests, M50NiL demonstrated approximately twice the fatigue life, L_{10}, of standard AISI M-50 (Bamberger, 1983). The two materials were processed by the same VIM-VAR process, and tested under identical conditions of load and speed.

An important characteristic in aircraft engines is that M50NiL allows sufficient time for the detection of spalling damage in the bearing before any catastrophic failure, because the tough core minimizes undesired crack propagation. In addition, M50NiL can operate at higher speeds, is more wear resistant, has higher tensile stress, higher fracture toughness, and lower boundary lubrication friction than AISI M-50.

13.19.2.3 DD400

For instrument ball bearings, corrosion resistance is very important. A stainless steel DD400 has been developed for precision miniature rolling bearings and small instrument rolling bearings. Corrosion resistance, combined with adequate hardness, has been achieved by increasing the quantity of dissolved chromium in the material. However, corrosion-resistant stainless steels have reduced fatigue resistance, and they are applied only for light-duty bearings. The composition of DD400 is 0.7% C, 13% Cr and it is martensitic stainless steel. DD400 replaced AISI 440C (1% C, 17% Cr), which was used for similar applications. DD400 demonstrated better performance in comparison to AISI 440C in small bearings. The most important advantages are: better surface finish of the races and rolling elements, better damping of vibrations, and improved fatigue life. These advantages are explained by the absence of large carbides in the heat-treated material.

13.20 PROCESSES FOR MANUFACTURING HIGH-PURITY STEEL

In addition to the chemical composition, the manufacturing process is very important for improving fatigue resistance, particularly at high temperatures. For critical applications, such as aircraft engines, there is a requirement for fatigue-resistant materials with a high degree of purity. It was realized that there are significant amount of impurities in the bearing rings and rolling elements, in the form of nonmetal particles as well as microscopic bubbles from gas released into the metal during solidification. In fact, these impurities have an adverse effect equivalent to small cracks in the material. These microscopic cracks propagate and cause early fatigue failure. Therefore, a lot of effort has been directed at developing ultrahigh-purity steels for rolling bearings.

An advanced method for high purity steel is the *vacuum induction method* (VIM). The melting furnace is inside a large vacuum chamber. The process uses steel of high purity, and the required alloys are added from hoppers into the vacuum chamber. A second method is the *vacuum arc remelting* (VAR) where a consumable electrode is melted by an electrical arc in a vacuum chamber. The two methods were combined and referred to as VIM-VAR. In the combined method, the steel from the vacuum induction method is melted again by the vacuum arc method. Successive vacuum arc remelting improves the bearing fatigue life.

Zaretzky (1997c) presented a detailed survey of the processing methods and testing of bearing materials for aerospace applications.

13.21 CERAMIC MATERIALS FOR ROLLING BEARINGS

There is an ever-increasing demand for better materials for rolling-element bearings in order to increase the speed and service life of machinery. In addition, machines are often exposed to corrosive environments and high temperatures that cause steel bearings to fail. In a corrosive environment, the life of regular rolling bearings made of steel is short. It would offer a huge economic benefit if an alternate material could be developed that would increase the life of rolling bearings.

For the last several decades, engineers have been searching for alternative materials for the roller bearing. Although there are significant improvements in the manufacturing processes and composition of steel bearings, scientists and engineers have been continually investigating ceramics as the most promising alternative materials.

In aviation, there is an ever-present need for the reduction of weight. It is possible to reduce the size and weight of engines by operating at higher speeds. In addition, weight reduction can be achieved if the engine efficiency can be improved by operating at higher temperatures. Let us recall that according to the basic principles of thermodynamics, the efficiency of the Carnot cycle is proportional to the process temperature. Therefore, there is a need for materials that can operate at high temperatures. It has been recognized that the bearings are one bottleneck that limits the speed and temperature of jet engines. A lot of research has been conducted in developing and testing ceramic materials that can endure higher temperatures in comparison to steel. In addition, ceramics have a low density, which is important in reducing the centrifugal force of the rolling elements, a limiting factor of speed.

Initially, tests were conducted with rolling elements made of aluminum oxide and silicon carbide. However, these tests indicated unacceptable early catastrophic failure, particularly at high speeds and under heavy loads. Better results were obtained later with silicon nitride, Si_3N_4.

The early manufacturing process for silicon nitride involved hot pressing. The parts did not have a uniform structure and had many surface defects. The parts required expensive finishing by diamond-coated tools. Moreover, the finished parts did not have the required characteristics for using them in rolling-element bearings.

Later, the development of a hot isostatically pressed (HIP) manufacturing process significantly improved the structure of silicon nitride. The most important properties of silicon nitride that make it suitable for rolling bearings is fatigue resistance under rolling contact and relatively high fracture toughness. Silicon nitride rolling elements showed a fatigue-failure mode by spalling, similar to steel. In addition, silicon nitride proved to be wear resistant under the high contact pressure of heavily loaded bearings. Most of the applications use silicon nitride ceramic rolling elements in steel rings, referred to as *hybrid ceramic bearings*.

13.21.1 Hot Isostatic Pressing (HIP) Process

The introduction of the HIP process offered many advantages over the previous hot-pressing process. The HIP process is done by applying a high pressure of inert gases—argon, nitrogen, helium—or air at elevated temperatures to all grain surfaces under a uniform temperature. Temperatures up to 2000°C (3630°F) and pressures up to 207 MPa (30,000 psi) are used. The temperature and pressure are accurately controlled. The term *isostatic* means that the static pressure of gas is equal in all directions throughout the part.

This process is already widely used for shaping parts of ceramic powders as well as other mixtures of metals and nonmetal powders. This process minimizes surface defects and internal voids in the parts. The most important feature of this process is that it results in strong bonds between the powder boundaries of similar or dissimilar materials, which improve the characteristics of the parts for many engineering applications.

In addition, the process reduces the cost of manufacturing because it forms net or near-net shapes (close to final shape) from various powders, such as metal, ceramic, and graphite. The cost is reduced because the parts are near the final shape and less expensive machining is needed.

There are also important downsides to ceramics in rolling bearings. The cost of manufacturing of ceramic parts is several times that of similar steel parts. In rolling bearings, a major problem is that the higher elastic modulus and lower Poisson ratios of silicon nitride result in higher contact stresses than for steel bearings (see Hertz equations in Chapter 12). It is obvious that silicon nitride's higher elastic modulus and hardness result in a small contact area between the balls and races. In turn, the maximum compression stress must be higher for ceramic on steel and even more in ceramic on ceramic. The high contact stresses can become critical and can cause failure of the ceramic rolling elements. This is particularly critical in all-ceramic bearings, because ceramic-on-ceramic contact results in higher stresses than ceramic balls on steel races.

13.21.2 Silicon Nitride Bearings

The most widely used type is the *hybrid bearing*. It combines silicon nitride balls with steel races. The second type is the *all-ceramic bearing*, often referred to as a *full-complement ceramic bearing*. The two types benefit from the properties of silicon nitride, which include low density, corrosion resistance, heat resistance, and electrical resistance.

13.21.3 Hybrid Bearings

The surfaces of steel races and ceramic rolling elements are compatible, in the sense that they have relatively low adhesive wear. Ceramics sliding or rolling on metals do not generate high adhesion force or microwelds at the asperity contacts. The ceramic rolling elements have high electrical resistance, which is important in electric motors and generators because they eliminate the problem of arcing in steel bearings. However, the most important advantage of silicon nitride rolling elements is their low density. The specific density of silicon nitride is 3.2, in comparison to 7.8 for steel (about 40% of steel). The centrifugal forces are proportional to the density of the rolling elements, and they become critical at high speeds. Since pressed silicon nitride rolling elements are lighter, the centrifugal forces are reduced.

Many experiments confirmed that hybrid bearings have a longer fatigue life than do M-50 steel rolling elements. At very high speeds, the relative improvement in the fatigue life of silicon nitride hybrid bearings is even higher, due to the lower density, which reduces the centrifugal forces.

The silicon nitride is very hard and has exceptionally high compressive strength, but the tensile strength is low. Low tensile strength is a major problem for mounting the rings on steel shafts; but hybrid bearings have steel rings, so this problem is eliminated.

Although research in hybrid bearings was conducted two decades earlier, it is only since 1990 that they have been in a wide use for precision applications, including machine tools. The high rigidity of silicon nitride balls was recognized for its potential for improvement in precision and reduction of vibrations. This property can be an advantage in high-speed rotors.

13.21.3.1 Fatigue Life of Hybrid Bearings

There is already evidence that hybrid bearings made of silicone nitride balls and steel rings have much longer fatigue life than do steel bearings of similar geometry. Examples of research work are by Hosang (1987) and Chiu (1995). The major disadvantage of hybrid bearings is their high cost.

However, the advantages of the hybrid bearing are expected to outweigh the high cost. Longer life at higher speeds and higher temperatures may end up saving money over the life cycle of the machine by reducing the need for maintenance and replacement parts. In addition, longer bearing life will result in reduced machine downtime, which results in the expensive loss of production. We have to keep in mind that the cost of bearing replacement is often much higher than the cost of the bearing itself.

13.21.3.2 Applications of Hybrid Bearings

Hybrid ceramic bearings have already been applied in high-speed machine tools, instrument bearings, and turbo machinery. Other useful applications of silicon nitride balls include small dental air turbines, food processing, semiconductors, aerospace, electric motors, and robotics.

In hybrid bearings, the ceramic balls prevent galling and adhesive wear even when no liquid lubricant is used. Nonlubricated hybrid bearings wear less than dry all-steel bearings. Operation of steel bearings without lubrication results in the formation of wear debris, which accelerates the wear process. Ceramic balls have a higher modulus of elasticity than steel, which makes the bearing stiffer, useful in reducing vibration and for precision applications.

Hybrid ceramic bearings demonstrated very good results in applications without any conventional grease or oil lubrication, but only a thin solid lubricant layer transferred from the cage material. Example of a successful application is in the propellant turbopump of the Space Shuttle, where grease or oil lubrication must be avoided due to the volatility of the propellants, see Gibson (2001).

For propulsion into orbit, the NASA Space shuttle has three engines. Each engine is fed propellants by four turbopumps, which were equipped with hybrid ceramic bearings with silicon nitride ceramic balls and a self-lubricating cage made of sintered PTFE and bronze powders. The PTFE is transferred as a third body of a thin film solid lubrication on the balls and races. The hybrid ceramic bearing in this severe application did not show any significant wear of the raceways. Tests indicated that various cage material combinations affected the life of the self-lubricated bearing in different ways. The best results were obtained by using silicon nitride ceramic balls and sintered PTFE and bronze cage. This combination was implemented successfully in all NASA Space shuttles.

The hybrid bearing is currently passing extensive tests for ultimate use in jet aircraft engines. However, at this time, it has not reached the stage of being actually used in aircraft engines. For safety reasons, the hybrid bearing must pass many strict tests before it can be approved for use in actual aircraft.

13.21.4 All-Ceramic Bearings

The most important advantage of all-ceramic bearings is that they resist corrosion, even in severe chemical and industrial environments where stainless steel bearings lack sufficient corrosion resistance.

Zaretzky (1989) published a survey of the research and development work in ceramic bearings during the previous three decades. He pointed out that since the elastic modulus of silicon nitride is higher than that of steel, the Hertz stresses are higher than for all-steel bearings. Zaretzky concluded that the dynamic capacity of the all-silicon-nitride bearing is only 5–12% of that of an all-steel bearing of similar geometry. In addition, there are problems mounting the ceramic ring on a steel shaft. The difference in thermal expansion results in high tensile stresses. Silicon nitride has exceptionally high compressive strength, but the tensile strength is low. Therefore, a ceramic ring requires a special design for mounting it on a steel shaft.

The most important advantage of all-silicon-nitride bearings is that they can operate at high temperature above the limits of steel bearings. However, at temperatures above 578 K (300°C), the available liquid lubricants cannot be used. Early tests indicated that all-ceramic bearings can operate with minimal or no lubrication. However, when tests were conducted at higher speeds, similar to those in gas turbine engines, catastrophic bearing failure resulted after a short time. In the future, solid lubricants may be developed to overcome this problem.

Another problem in the way of extending the operating temperature of all-ceramic bearings is that high-temperature cage materials were not available. Tests indicated that the best results could be achieved with graphite cages; see Mutoh et al. (1994).

An important advantage of the all-ceramic bearing remains that it can resist corrosion in very corrosive environments where steel bearings would be damaged. Moreover, regular bearings often fail because an industrial corrosive environment breaks down the lubricant. In such cases, the all-ceramic bearing can be a solution to these problems. It also can operate with minimal or no lubrication. In addition, it has high rigidity, important in precision machines.

The all-ceramic bearings are used in the etching process for silicon wafers, where sulfuric acid and other corrosives are used. Only ceramic bearings can resistant this corrosive environment. Another application is ultraclean vacuum systems. Liquid lubricants evaporate in a vacuum, and ceramic bearings are an alternative for this purpose. All-ceramic bearings can also be used in applications where nonmagnetic bearings are required. Hybrid bearings with stainless steel rings are also used for this purpose.

Sealed pumps driven by magnetic induction are widely used for pumping various corrosive chemicals. Most sealed pumps operate with hydrodynamic journal bearing with silicon carbide sleeve. The ceramic sleeves are used because of their corrosive resistance and for their nonmagnetic properties.

However, the viscosity of the process fluids is usually low, and the hydrodynamic fluid film is generated only at high speeds. For pumps that operate with frequent start-ups, there is high wear and the bearings do not last for a long time. All-ceramic rolling bearings made of silicon nitride proved to be a better selection for sealed pumps. The silicone nitride rolling bearings are not sensitive to frequent start-ups and have good chemical corrosion resistance as well as the desired nonmagnetic properties for this application.

13.21.5 Cage Materials for Hybrid Bearings

Different cage materials have been tested in ceramic hybrid bearings. Appropriate cage material is a critical problem in applications where solid lubrication or operation without lubricant is required. In such cases, the cage material provides the solid lubricant.

A graphite cage offered a low wear rate in high-temperature applications. Self-lubricating (soft) cage materials resulted in a longer bearing life with lower wear rate and lower friction in comparison to hard cage material. However, at high temperatures, self-lubricating cage materials resulted in excessive degradation of the cage material by high-temperature oxidation.

13.22 ROLLING BEARING CAGES

The rolling bearing cage, often referred to as a *separator* or a *retainer*, is mounted in the bearing in order to equally space the rolling elements (balls or rollers) and prevent contact friction between them. The cage rotates with the rolling elements, which are freely rotating in the confinement of the cage. In addition, the cage retains the grease to provide for effective lubrication. Cages made of porous materials, such as phenolic, absorb liquid lubricants and assist in providing a very thin layer of oil for a long time. Examples of rolling bearing cage designs are shown in Fig. 13-27 (FAG, 1998).

Cages are made of the following materials.

Cages made of brass are commonly used in medium and large roller bearings.
Cages made of nylon strengthened by two round strips of steel are commonly used in small ball bearings.
Cages made of steel are used in miniature ball bearings.
Cages made of phenolic are used in ultrahigh-precision bearings.

13.23 BEARING SEALS

Seals act as a barrier that prevents the loss of the lubricant from the bearing housing. In addition, seals restrict the entry of any foreign particles or undesired process liquids into the bearings. Reliable operation of the seals is very important. In the case of lubricant loss, it can result in bearing failure. Any penetration of foreign particles into the bearing will result in reduction of its service life. Thus seals are essential for the proper functioning of the bearing. Seals are generally classified into two types, contact seals and noncontact seals.

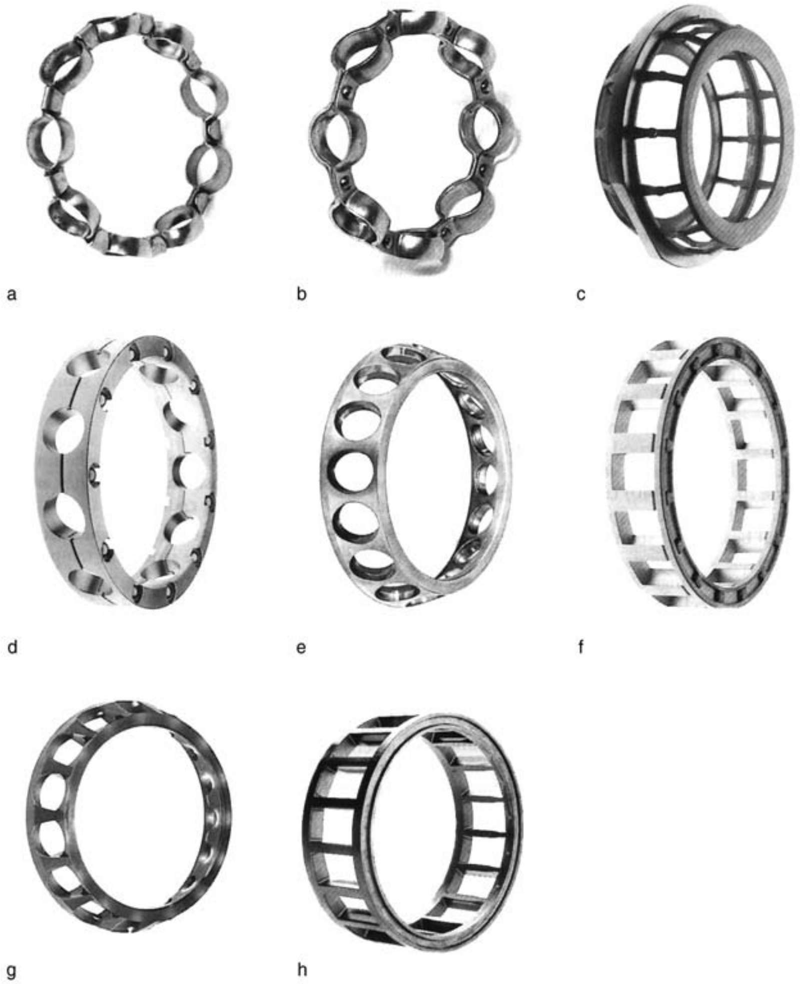


FIG. 13-27 Examples of rolling bearing cages. *Pressed cages of steel*: Lug cage (a) and rivet cage (b) for deep-groove ball bearings, window-type cage (c) for spherical roller bearings. *Machined brass cages*: Riveted machined cage (d) for deep-groove ball bearings, brass window-type cage (e) for angular contact ball bearings and machined brass cage with integral crosspiece rivets (f) for cylindrical roller bearings. *Molded cages* made of glass-fibre reinforced polyamide: window-type cage (g) for single-row angular contact ball bearings and window-type cage (h) for cylindrical roller bearings. (From FAG, 1998, with permission.)

13.23.1 Contact Seals

These seals remain in contact with the sliding surface, and thus they wear after a certain period of operation and need replacement. They are also referred to as rubbing seals. In order to make these seals effective; a certain amount of contact pressure should always be present between the seal and shaft. The wear of contact seals increases by the following factors:

Friction coefficient
Bearing temperature
Sliding velocity
Surface roughness
Contact pressure

Under favorable conditions, there is a very thin layer of lubricant that separates the seal and the shaft surfaces (similar to fluid film but much thinner). The film thickness can reach the magnitude of 500 nm, at shaft surface speed of 0.4 m/s (Lou Liming, 2001). A few examples of widely used contact seals are presented in Figs. 13-28a–f.

13.23.1.1 Felt Ring Seals

These seals (Fig. 13.28a) are widely used for grease lubrication. Felt ring seals are soaked in a bath of oil before installation, for reduction of friction. Felt seals provide excellent sealing without much contact pressure and are effective against penetration of dust. Therefore, they do not cause much friction power loss. The number of felt rings depends on the environment of the machine. The dimensions of felt seals are standardized.

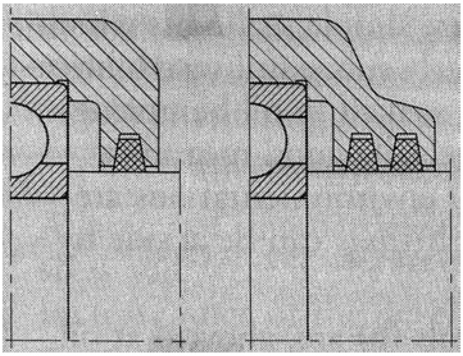


FIG. 13-28a Felt ring seal (from FAG, 1998, with permission).

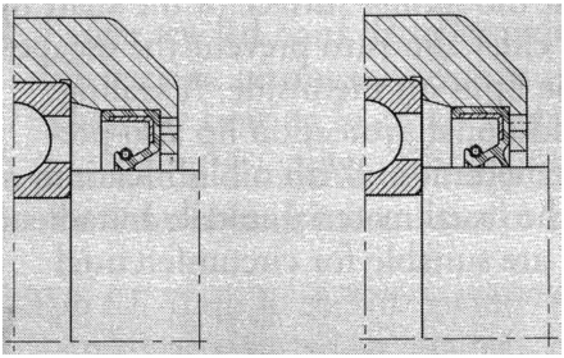


FIG. 13-28b Radial shaft seals (from FAG, 1998, with permission).

13.23.1.2 Radial Shaft Sealing Rings

These are the most widely used contact lip seals for liquid lubricant (Fig. 13-28b). The basic construction incorporates the lip of the seal pressed on the sliding surface of a shaft with the help of a spring.

13.23.1.3 Double-Lip Radial Seals

These seals (Fig. 13-28c) consist of two lips. The outer lip restricts any entry of foreign particle, and the inner lip retains the lubricant inside the bearing housing. When grease is applied between the two lips, the bearing life increases.

13.23.1.4 Axially Acting Lip Seals

The major advantage of this seal (Fig. 13-28d) is that it is not sensitive to radial misalignment. The seal is installed by pushing it on the surface of the shaft until its lip comes in contact with the housing wall. These seals are often used as extra

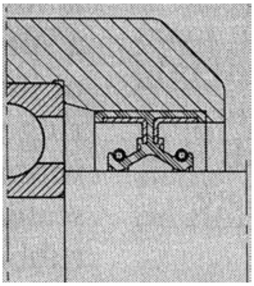


FIG. 13-28c Double-lip radial seal (from FAG, 1998, with permission).

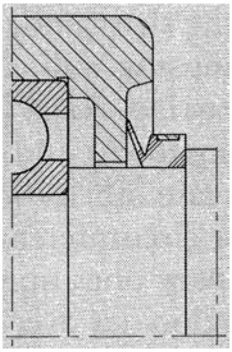


FIG. 13-28d Axially acting lip seal (from FAG, 1998, with permission).

seals in a contaminated environment. At very high speeds, these seals are not effective due to the centrifugal forces.

13.23.1.5 Spring Seals

These seals (Fig. 13.28e) are effective only for grease lubrication. A thin round sheet metal is clamped to the inner or outer ring, and provides a light contact pressure with the second ring.

13.23.1.6 Sealed Bearing

This seal (Fig. 13.28f) is manufactured with the bearing, and widely used for sealed for life bearings. The seal is made of oil resistant rubber, which is connected to the outer ring, and lightly pressed on the inner ring.

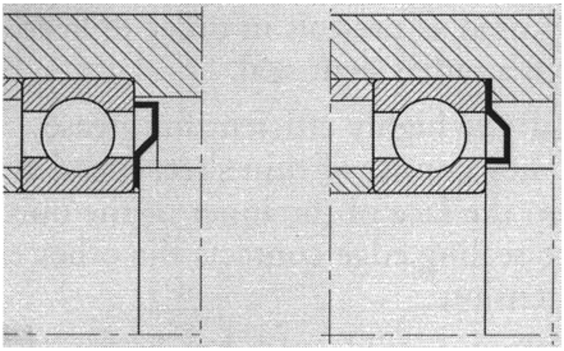


FIG. 13-28e Spring seals (from FAG, 1998, with permission).

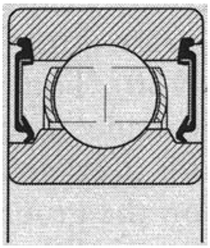


FIG. 13-28f Sealed bearing (from FAG, 1998, with permission).

13.23.2 Noncontact Seals

Noncontact seals are also known as nonrubbing seals. These seals are widely used for grease lubrication. In these seals there is only viscous friction, and thus they perform well for a longer time. In noncontact seals there is a small radial clearance between the housing and the shaft (0.1–0.3 mm). These seals are not so sensitive to radial misalignment of the shaft. Most important, since there is no contact, not much heat is generated by friction, which make it ideal for high-speed applications.

A number of grooves are designed into the housing, which contain grease. The grease filled grooves form effective sealing. If the environment is contaminated, the grease should be replaced frequently. If oil is used for lubrication, the grooves are bored spirally in the direction opposite to that of the rotation of the shaft. Such seals are also known as shaft-threaded seals.

Some examples of noncontact seals are shown in Fig. 13-29.

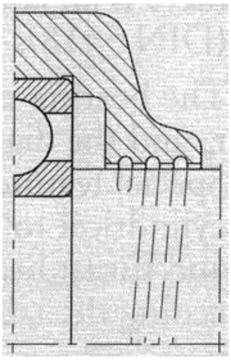


FIG. 13-29a Grooved labyrinth seal (from FAG, 1998, with permission).

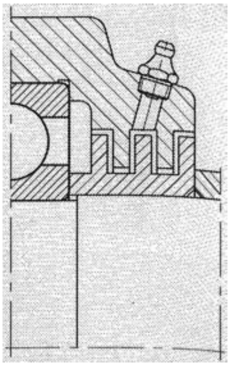


FIG. 13-29b Axial webbed noncontact seal (from FAG, 1998, with permission).

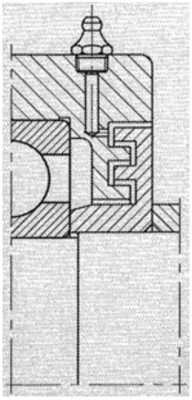


FIG. 13-29c Radial webbed noncontact seal (from FAG, 1998, with permission).

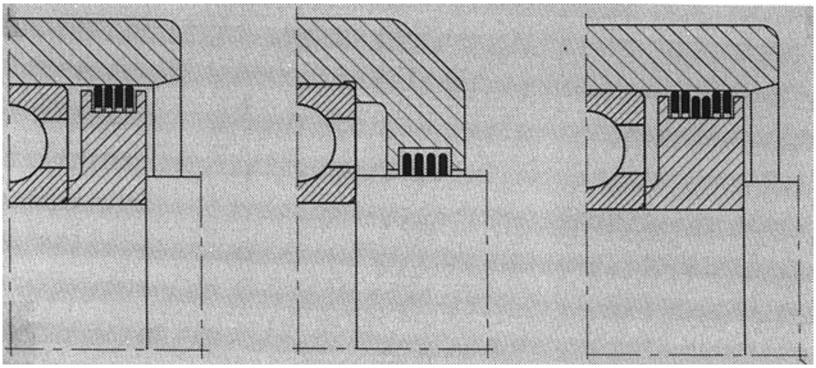


FIG. 13-29d Noncontact seal with lamellar rings (from FAG, 1998, with permission).

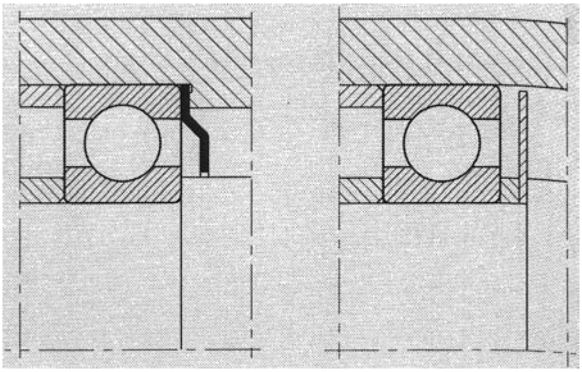


FIG. 13-29e Baffle plates seal (from FAG, 1998, with permission).

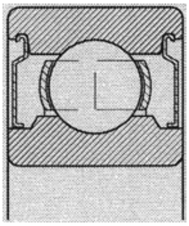


FIG. 13-29f Bearing with shields (from FAG, 1998, with permission).

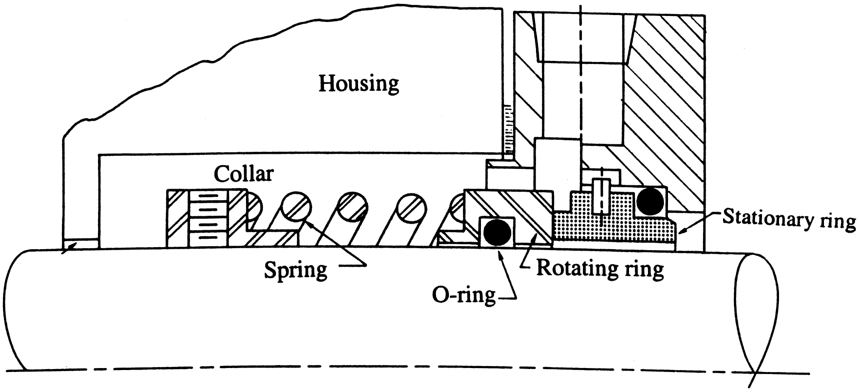


FIG. 13-30 Mechanical seal.

13.24 MECHANICAL SEALS

This seal is widely used in pumps. The sealing surfaces are normal to the shaft, as shown in Fig. 13-30. The concept is that of two rubbing surfaces, one stationary and one rotating with the shaft. The surfaces are lubricated and cooled by the process fluid. The normal force between the rubbing surfaces is from the spring force and the fluid pressure. The materials of the rubbing rings are a combination of very hard and very soft materials, such as silicon carbide and graphite. The lubrication film is very thin, and the leak is negligible.

Problems

13-1 A single-row, standard deep-groove ball bearing operates in a machine tool. It is supporting a shaft of 30-mm diameter. The bearing is designed for 90% reliability. The radial load on the bearing is 3000 N (no axial load). The shaft speed is 7200 RPM. The lubricant is SAE 20 oil, and the maximum expected surrounding (ambient) temperature is 30°C. Assume the oil operating temperature is 10°C above ambient temperature.

a. Find the life adjustment factor a_3.
b. Find the adjusted fatigue life L_{10} of a deep-groove ball bearing.
c. Find the maximum static radial equivalent load.

The deep groove bearing data, as specified in a bearing catalogue, is as follows:

Designation bearing: No. 6006
Bore diameter: $d = 30$ mm
Outside diameter: $D = 55$ mm
Dynamic load rating: $C = 2200$ lb
Static load rating: $C_0 = 1460$ lb

13-2 In a gearbox, two identical standard deep-groove ball bearings support a shaft of 35-mm diameter. There is locating/floating arrangement where the floating bearing supports a radial load of 10,000N and the locating bearing supports a radial load of 4000N and a thrust load of 5000N. The shaft speed is 3600 RPM. The lubricant is SAE 30 oil, and the maximum expected surrounding (ambient) temperature is 30°C. Assume that the oil operating temperature is 5°C above ambient temperature. The two deep-groove bearings are identical. The data, as specified in a bearing catalogue, is as follows:

Designation bearing: No. 6207
Bore diameter: $d = 35$ mm
Outside diameter: $D = 72$ mm
Dynamic load rating: $C = 4400$ lb
Static load rating: $C_0 = 3100$ lb

a. Find the life adjustment factor a_3 for the locating and floating bearings.
b. Find the adjusted fatigue life L_{10} of a deep-groove ball bearing for the locating and floating bearings.
c. Find the static radial equivalent load.
d. Find the radial static equivalent load for the locating and floating bearing.

13-3 Find the operating clearance (or interference) for a standard deep-groove ball bearing No. 6312 that is fitted on a shaft and inside housing as shown in Fig. 13-6. During operation, inner ring as well as shaft temperature is 8°C higher than the temperature of outer ring and housing. The bearing is of C3 class of radial clearance (radial clearance of 23–43 μm from Table 13-2).

The dimensions and tolerances of inner ring and shaft are
Bore diameter: $d = 60$ mm $(-15/+0)$ μm
Shaft diameter: $d_s = 60$ mm $(+21/+2)$ μm k6
OD of inner ring: $d_1 = 81.3$ mm

The dimensions and tolerances of outer ring and housing seat are

OD of outer ring: $D = 130$ mm $(+0/-18)$ μm
ID of outer ring: $D_1 = 108.4$ mm
ID of housing seat: $D_H = 130$ mm $(-21/+4)$ μm K6

Neglect the surface smoothing effect, and assume that the housing and shaft seats were measured, and the actual dimension is at 1/3 of the tolerance zone, measured from the tolerance boundary close to the surface where the machining started, e.g., the shaft diameter is $60\text{ mm} + [21-(21-3)/3]\,\mu\text{m} = 60.015$ mm.

Consider elastic deformation and thermal expansion for the calculation of the two boundaries of the operating radial clearance tolerance zone.

Coefficient of thermal expansion of steel is $\alpha = 0.000011$ [1/C]

13-4 A standard deep-groove ball bearing No. 6312 that is mounted on a shaft and into a housing as shown in Fig. 13-6. The bearing width is $B = 31$ mm. The shaft and ring are made of steel $E = 2 \times 10^{11}$ Pa.

The dimensions and tolerances of inner ring and shaft are

Bore diameter: $d = 60\,\text{mm}\ (-15/+0)\ \mu\text{m}$
Shaft diameter: $d_s = 60\,\text{mm}\ (+24/+11)\ \mu\text{m}$, m5
OD of inner ring: $d_1 = 81.3\,\text{mm}$

The dimensions and tolerances of outer ring and housing seat are

OD of outer ring: $D = 130\,\text{mm}\ (+0/-18)\ \mu\text{m}$
ID of outer ring: $D_1 = 108.4\,\text{mm}$
ID of housing seat: $D_H = 130\,\text{mm}\ (-21/+4)\ \mu\text{m}$, K6

Neglect the surface smoothing effect, and assume a rectangular cross section of the bearing rings for all calculations.

1. Find the maximum and minimum pressure between the shaft and bore surfaces.
2. Find the minimum and maximum tensile stress in the inner ring after it is tightly fitted on the shaft.
3. If the friction coefficient is $f = 0.5$, find the maximum axial force (for the tightest tolerance), which is needed for sliding the inner ring on the shaft.
4. Find the minimum inertial torque (N–m), which can result in undesired rotation sliding of the shaft inside the inner ring during the start-up ($f = 0.5$).
5. The bearing is heated for mounting it on the shaft without any axial force. Find the temperature rise of the bearing (relative to the shaft), for all bearings and shaft within the specified tolerances. Coefficient of thermal expansion of steel is $\alpha = 0.000011$ [1/C].

13-5 Modify the design of the bearing arrangement of the NC–lathe main spindle in Fig. 13-10b. The modified design will be used for rougher machining at lower speeds. Adjustable bearing arrangement with two tapered roller bearings will replace the current bearing arrangement. For a rigid support, an adjustable bearing arrangement was selected with the apex points between the two bearings.

a. Design and sketch the cross-section view of the modified lathe main spindle.
b. Show the centerlines of the tapered rolling elements and the apex points, if the bearings preload must not be affected by temperature rise during operation.
c. Specify the tolerances for the seats of the two bearings.

13-6 Modify the design of the bearing arrangement of the NC–lathe main spindle in Fig. 13-10b to a locating/floating bearing arrangement. On the right hand (the locating side), the modified design entails three adjacent angular ball bearings, two in an adjustable arrangement, and the third in tandem arrangement to machining thrust force. On the left hand, two adjacent cylindrical roller bearings are the floating bearings that support only radial force.

a. Design and sketch the cross-section view of the modified design.
b. Specify tolerances for all the bearing seats.