

ANSI/ABMA 9:2015

# AMERICAN NATIONAL STANDARD

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Committee B3*

## Load Ratings And Fatigue Life For Ball Bearings ANSI/ABMA 9:2015 (Revision of ANSI/ABMA 9:1990)

Secretariat

American Bearing  
Manufacturers Association

ANSI/ABMA 9:2015

 **ABMA**  
*American Bearing Manufacturers Association*

ABMA  
2025 M Street, NW  
Suite 800  
Washington, DC 20036  
Ph: 202-367-1155  
Fax: 202-367-2155  
E-mail: [info@americanbearings.org](mailto:info@americanbearings.org)  
[www.americanbearings.org](http://www.americanbearings.org)

## AMERICAN NATIONAL STANDARD

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# Load Ratings and Fatigue Life for Ball Bearings

## 1. INTRODUCTION

### 1.1 Purpose of Standard

Ball bearing performance is a function of many variables. These include the bearing design, the characteristics of the material from which the bearings are made, the way in which they are manufactured, as well as many variables associated with their application. The only sure way to establish the satisfactory operation of a bearing selected for a specific application is by actual performance in the application. As this is often impractical, another basis is required to estimate the suitability of a particular bearing for a given application. This is the purpose of this standard.

This standard specifies the method of calculating the basic dynamic load rating of rolling bearings within the size ranges shown in the relevant ANSI/ABMA standards, manufactured from contemporary, commonly used, good quality hardened bearing steel in accordance with good manufacturing practice and basically of conventional design as regards the shape of rolling contact surfaces.

This standard also specifies the method of calculating the basic rating life, which is the life associated with 90% reliability, with commonly used high quality material, good manufacturing quality and with conventional operating conditions. In addition, it specifies the method of calculating adjusted rating life, in which various reliabilities, special bearing properties and specific operating conditions are taken into account by means of life adjustment factors.

Furthermore, this standard specifies the method of calculating the basic static load rating and the static equivalent load for ball bearings within the size ranges shown in the relevant ANSI/ABMA Standards, manufactured from good quality hardened bearing steel, in accordance with good manufacturing practice and basically of conventional design as regards the shape of rolling contact surfaces.

### 1.2 Life Criterion

Even if ball bearings are properly mounted, adequately lubricated, protected from foreign matter, and are not subjected to extreme operating conditions, they can ultimately fatigue. Under ideal conditions, the repeated stresses developed in the contact areas between the ball and the raceways eventually can result in fatigue of the material which manifests itself as spalling of the load carrying surfaces. In most applications the fatigue life is the maximum useful life of a bearing. This fatigue is the criterion of life used as the basis for the first part of this standard.

Fatigue life calculated in accordance with this standard does not represent the maximum that can be attained by applying all known technology to ball bearing design and application. Neither does it represent the minimum that should be expected of a bearing made by a producer lacking skill and experience in the design and manufacture of ball bearings, even though the bearing meets the geometric parameters given below. The calculated fatigue life represents the performance normally expected from high quality bearings made by reputable manufacturers. Manufacturers can supply longer lived bearings by the application of advanced materials and manufacturing processes. The present standard has evolved as a means for bearing users to specify a reasonable standard of performance for the bearing they wish to purchase.

### 1.3 Static Load Criterion

A static load is a load acting on a non-rotating bearing. Permanent deformations appear in balls and raceways under a static load of moderate magnitude and increase gradually with increasing load.

It is often impractical to establish whether the deformations appearing in a bearing in a specific application are permissible by testing the bearing in that application. Other methods are therefore required to establish the suitability of the bearing selected.



Experience shows that a total permanent deformation of 0.0001 of the rolling element diameter, at the center of the most heavily loaded ball/raceway contact, can be tolerated in most bearing applications without the subsequent bearing operation being impaired. The basic static load rating is, therefore, given a magnitude such that approximately this deformation occurs when the static equivalent load is equal to the load rating.

Tests indicate that a load of the magnitude in question may be considered to correspond to a calculated contact stress of;

4,600 MPa (667,000 psi) for self-aligning ball bearings, and

4,200 MPa (609,000 psi) for all other ball bearings

at the center of the most heavily loaded rolling element/raceway contact. The formulae and factors for the calculation of the basic static load ratings are based on these contact stresses.

The permissible static equivalent load may be smaller than, equal to or greater than the basic static load rating, depending on the requirements for smoothness of operation and friction, as well as on actual contact surface geometry. Bearing users without previous experience of these conditions should consult the bearing manufacturers.

## 2. SYMBOLS

$a_1$	life adjustment factor for reliability
$a_2$	life adjustment factor for special bearing properties
$a_3$	life adjustment factor for operating conditions
$C_a$	basic dynamic axial load rating, in newtons (pounds)
$C_r$	basic dynamic radial load rating, in newtons (pounds)
$C_{0a}$	basic static axial load rating, in newtons (pounds)
$C_{0r}$	basic static radial load rating, in newtons (pounds)
$D_{pw}$	pitch diameter of ball set, in millimeters (inches)
$D_w$	nominal ball diameter, in millimeters (inches)
$e$	limiting value of $F_a/F_r$ for the applicability of different values of factors X and Y
$F_a$	bearing axial load (axial component of the actual bearing load), in newtons (pounds)
$F_r$	bearing radial load (radial component of the actual bearing load), in newtons (pounds)
$f_{cm}$	factor which depends on the geometry of the bearing components, the accuracy to which the various components are made, and the material
$f_0$	factor for calculation of basic static load rating.
$i$	number of rows of rolling elements
$L_{na}$	adjusted rating life, in million revolutions
$L_{10}$	basic rating life, in million revolutions

$n$	speed of rotation, in revolutions per minute
$n$	subscript for probability of failure, in percent
$P_a$	dynamic equivalent axial load, in newtons (pounds)
$P_r$	dynamic equivalent radial load, in newtons (pounds)
$P_{0a}$	static equivalent axial load, in newtons (pounds)
$P_{0r}$	static equivalent radial load, in newtons (pounds)
$S_0$	static safety factor
$X$	dynamic radial load factor
$X_0$	static radial load factor
$Y$	dynamic axial load factor
$Y_0$	static axial load factor
$Z$	number of rolling elements in a single-row bearing; number of rolling elements per row of a multi-row bearing with the same number of rolling elements per row
$\alpha$	nominal contact angle, in degrees
$\kappa$	viscosity ratio, $\nu / \nu_1$
$\Lambda$	film parameter
$\nu$	actual kinematic viscosity at the operating temperature, in square millimeters per second (centiStokes)
$\nu_1$	reference kinematic viscosity, required to obtain adequate lubrication condition, in square millimeters per second (centiStokes)

### 3. DEFINITIONS

For the purposes of this Standard, the definitions given in ANSI/ABMA/ISO Standard 5593 together with the following apply.

#### 3.1 Life

For an individual rolling bearing, the number of revolutions which one of the bearing rings (or washers) makes in relation to the other ring (or washer) before the first evidence of fatigue develops in the material of one of the rings (or washers) or one of the rolling elements.

NOTE: Life may also be expressed in number of hours of operation at a given constant speed of rotation.

#### 3.2 Reliability (in the context of bearing life)

For a group of apparently identical rolling bearings, operating under the same conditions, the percentage of the group that is expected to attain or exceed a specified life.

NOTE: The reliability of an individual rolling bearing is the probability that the bearing will attain or exceed a specified life.



### **3.3 Static Load**

The load acting on a bearing when the speed of rotation of its rings in relation to each other is zero.

### **3.4 Pitch Diameter of a Ball Set, $D_{pw}$**

The diameter of the circle containing the centers of the balls in one row in a bearing.

### **3.5 Rating Life**

The predicted value of life based on a basic dynamic radial load rating or a basic dynamic axial load rating.

### **3.6 Basic Rating Life, $L_{10}$**

The time at which 10% of a bearing population operating under the same conditions will have failed and 90% will have survived. The life is associated with a 10% probability of failure by the time calculated.

### **3.7 Adjusted Rating Life, $L_{na}$**

The rating life obtained by adjustment of the basic rating life for a desired reliability level, and/or special bearing properties, and/or specific operating conditions.

### **3.8 Basic Dynamic Radial Load Rating, $C_r$**

That constant stationary radial load which a rolling bearing can theoretically endure for a basic rating life of one million revolutions.

### **3.9 Basic Static Radial Load Rating, $C_{0r}$**

Static radial load which corresponds to a calculated contact stress at the center of the most heavily loaded rolling element/raceway contact of

4,600 MPa (667,000 psi) for self-aligning ball bearings, and

4,200 MPa (609,000 psi) for all other radial ball bearing types

NOTE 1: In the case of a single-row angular contact bearing, the radial load rating refers to the radial component of that load which causes a purely radial displacement of the bearing rings in relation to each other.

NOTE 2: For these contact stresses, under static load, a total permanent deformation of rolling element and raceway occurs which is approximately 0.0001 of the rolling element diameter.

### **3.10 Basic Dynamic Axial Load Rating, $C_a$**

That constant centric axial load which a rolling bearing can theoretically endure for a basic rating life of one million revolutions.

### **3.11 Basic Static Axial Load Rating, $C_{0a}$**

Static centric axial load which corresponds to a calculated contact stress at the center of the most heavily loaded rolling element/raceway contact of

4,200 MPa (609,000 psi) for thrust ball bearings.

NOTE: For this contact stress, under static load, a total permanent deformation of rolling element and raceway occurs which is approximately 0.0001 of the rolling element diameter.

### **3.12 Dynamic Equivalent Radial Load, $P_r$**

That constant stationary radial load under the influence of which a rolling bearing should have the same life as it would attain under the actual load conditions.

### **3.13 Static Equivalent Radial Load, $P_{0r}$**

Static radial load which should cause the same contact stress at the center of the most heavily loaded rolling element/raceway contact as that which occurs under the actual load conditions.

### **3.14 Dynamic Equivalent Axial Load, $P_a$**

That constant centric axial load under the influence of which a rolling bearing should have the same life as it would attain under the actual load conditions.

### **3.15 Static Equivalent Axial Load, $P_{0a}$**

Static centric axial load which should cause the same contact stress at the center of the most heavily loaded rolling element/raceway contact as that which occurs under the actual load conditions.

### **3.16 Static Safety Factor, $S_0$**

Ratio between the basic static load rating and the static equivalent load, giving a margin of safety against inadmissible permanent deformation on rolling elements and raceways.

### **3.17 Nominal Contact Angle, $\alpha$**

The angle between a plane perpendicular to a bearing axis (a radial plane) and the nominal line of action of the resultant of the forces transmitted by a bearing ring or washer to a rolling element.

### **3.18 Conventional Operating Conditions**

Conditions which may be assumed to prevail for a bearing which is properly mounted and protected from foreign matter, adequately lubricated, conventionally loaded, not exposed to extreme temperature and not run at exceptionally low or high speed.

### **3.19 Viscosity Ratio, $\kappa$**

Actual kinematic oil viscosity at operating temperature divided by the reference kinematic viscosity for adequate lubrication.

### **3.20 Film Parameter, $\Lambda$**

Ratio of lubricant film thickness to composite r.m.s surface roughness, used to estimate the influence of lubrication on bearing life.

### **3.21 Pressure-viscosity Coefficient**

Parameter characterizing the influence of oil pressure on the oil viscosity in the rolling element contact.

### **3.22 Bearing Arrangements: Paired Mounting**

Arrangement of two rolling bearings mounted side-by-side on the same shaft such that they operate as a unit, mounted back-to-back, face-to-face or tandem.

### **3.23 Bearing Arrangements: Back-to-back**

Arrangement of two rolling bearings mounted side-by-side with the back faces of the outer rings adjacent.

### **3.24 Bearing Arrangements: Face-to-face**

Arrangement of two rolling bearings mounted side-by-side with the front faces of the outer rings adjacent.

### **3.25 Bearing Arrangements: Tandem**

Arrangement of two or more rolling bearings mounted side-by-side with the back face of the outer ring of one bearing adjacent to the front face of the outer ring of the next bearing.

## **4. SCOPE**

### **4.1 Bearing Types**

#### **4.1.1 General**

Ball bearings covered by this standard are presumed to be within the size ranges shown in the relevant ANSI/ABMA dimensional standards, manufactured from contemporary, commonly used, good quality hardened bearing steel in accordance with good manufacturing practice and basically of conventional design as regards the shape of rolling contact surfaces.

#### **4.1.2 Radial, Deep Groove and Angular Contact**

This standard applies to radial, deep groove and angular contact ball bearings with an inner ring cross-sectional raceway groove radius not larger than  $0.52 D_w$  and an outer ring cross-sectional raceway groove radius not larger than  $0.53 D_w$ .

#### **4.1.3 Filling Slot, Deep Groove**

This standard applies to filling slot, deep groove ball bearings with an inner ring cross-sectional raceway groove radius not larger than  $0.52 D_w$  and an outer ring cross-sectional raceway groove radius not larger than  $0.53 D_w$ .

#### **4.1.4 Radial, Self-Aligning**

This standard applies to radial, self-aligning ball bearings with an inner ring cross-sectional raceway groove radius not larger than  $0.53 D_w$ .

#### **4.1.5 Thrust**

This standard applies to thrust ball bearings with cross-sectional raceway groove radii not larger than  $0.54 D_w$ .

#### **4.1.6 Double Row, Radial and Angular Contact**

Double row, radial and angular contact ball bearings and double direction thrust ball bearings, as specified by this standard, are presumed to be symmetrical.

### **4.2 Limitations**

#### **4.2.1 Truncated Contact Area**

This standard may not be safely applied to ball bearings subjected to loading which causes the contact area of the ball with the raceway to be truncated by the raceway shoulder. This limitation depends strongly on details of bearing design which are not standardized.

#### **4.2.2 Materials**

This standard applies to ball bearings made from hardened, good quality bearing steel. While a complete metallurgical description is beyond the scope of this standard, typical cleanliness and material composition specifications for bearing quality steel are given in ASTM A295/A295M and A485 for through hardening steels, and in ASTM A534 for carburizing steels. Typical hardness levels range from HRC 58 to 65 for bearing rings, washers and balls.



### **4.2.3 Bearing Types**

The  $f_{cm}$  factors specified in basic load rating formulae are valid only for those ball bearing configurations specified in section 4.1 above. This standard is not applicable to designs where the rolling elements operate directly on a shaft or housing surface, unless that surface is equivalent in all respects to the bearing ring (or washer) raceway it replaces. This standard is not applicable to instrument ball bearings covered in ANSI/ABMA standards 12.1 and 12.2.

### **4.2.4 Lubrication**

Basic rating life calculated according to this standard is based on the assumption that the bearing is adequately lubricated. Determination of adequate lubrication depends upon the bearing application. An adequate amount of an appropriate type of lubricant is essential to achieving expected performance. The lubricant must be free of excessive contaminants and of a viscosity level that will provide a film thickness somewhat greater than the rolling contact surfaces' composite roughness at the operating temperature ( $\Lambda > 1$ ).

### **4.2.5 Ring Support and Alignment**

Basic rating life calculated according to this standard assumes that the bearing inner and outer rings are rigidly supported, and that the inner and outer ring axes are properly aligned. Bearing rings (or washers) must be mounted so that any deformation of rings as a result of mounting compliance is small compared to contact deformation under the applied load.

### **4.2.6 Internal Clearance**

Radial ball bearing basic rating life calculated according to this standard is based on the assumption that only a nominal internal clearance occurs in the mounted bearing at operating speed, load and temperature. Internal clearance may be needed to account for effects of interference fits and thermal gradients; however, excessive clearance will reduce life. Negative clearance will also decrease life and increase friction. Excessive negative clearance may lead to bearing seizure.

### **4.2.7 High Speed Effects**

Basic rating life calculated according to this standard does not account for high speed effects such as ball centrifugal forces and gyroscopic moments. These effects tend to diminish fatigue life. Analytical evaluation of these effects frequently requires the use of high speed digital computation devices and hence, cannot be included herein.

### **4.2.8 Interference Fits**

Interference fits between the shaft and bearing bore, and centrifugal effects of high-speed operation will introduce a tensile hoop stress in the bearing inner ring. This tensile stress reduces bearing life. In addition, a high tensile stress can lead to catastrophic failure by fracture of the race.

### **4.2.9 Residual Stress**

Compressive residual stress that extends into the zone of maximum shear stress will reduce the effective shearing stress beneath the contacting surfaces of the bearing race. This reduced stress can extend bearing life, especially at light load. Compressive residual stress will counter the negative effect of tensile stress and reduce the risk of race fracture. Compressive residual stress can be introduced by use of case-hardened steel or by mechanically cold working the bearing race.

### **4.2.10 Groove Radii**

The load-carrying ability of a bearing is not necessarily increased by the use of smaller groove radii than those specified in sections 4.1.2 to 4.1.5, but it is reduced by the use of groove radii larger than those specified.

#### **4.2.11 Tolerances**

This standard applies to radial ball bearings made to ABEC 1 level of precision, or better, commensurate with ANSI/ABMA standard 20, and to insert bearings covered by ANSI/ABMA standard 15, and to thrust ball bearings covered by ANSI/ABMA standards 24.1 and 24.2.

#### **4.2.12 Plastic Deformation in the Contact Area**

If  $P_r > C_{0r}$  or  $P_r > 0.5C_r$  or  $P_a > 0.5C_a$  then plastic deformation may occur in the contact area. The user should consult the bearing manufacturer for recommendations and evaluation of equivalent load and life.

#### **4.3 Operating Parameters.**

Calculations according to this standard do not yield satisfactory results for bearings subjected to such application conditions which cause deviations from a normal load distribution in the bearing, for example misalignment, housing or shaft deflection, rolling element centrifugal forces or other high speed effects, and preload or extra large clearance in radial bearings. Where there is reason to assume that such conditions prevail, the user should consult the bearing manufacturer for recommendations and evaluation of equivalent load and life.

## 5. RADIAL AND ANGULAR CONTACT BALL BEARINGS

### 5.1 Basic Dynamic Radial Load Rating

#### 5.1.1 Basic Dynamic Radial Load Rating for Single Bearings

The basic dynamic radial load rating for radial and angular contact ball bearings is given by the following;

for  $D_w \leq 25.4\text{mm}$  (1 inch)

$$C_r = f_{cm} (i \cos \alpha)^{0.7} Z^{2/3} D_w^{1.8}$$

for  $D_w > 25.4\text{mm}$  (1 inch)

$$C_r = 3.647 f_{cm} (i \cos \alpha)^{0.7} Z^{2/3} D_w^{1.4} \quad (\text{metric})$$

$$C_r = f_{cm} (i \cos \alpha)^{0.7} Z^{2/3} D_w^{1.4} \quad (\text{inch})$$

Values of  $f_{cm}$  are obtained from the appropriate column of Table 1. They apply to bearings with a cross-sectional raceway groove radius not larger than  $0.52 D_w$  in radial and angular contact ball bearing inner rings and not larger than  $0.53 D_w$  in radial and angular contact ball bearing outer rings and self-aligning ball bearing inner rings.

NOTE: The load-carrying ability of a bearing is not necessarily increased by the use of smaller groove radii, but it is reduced by the use of groove radii larger than those specified above.

#### 5.1.2 Basic Dynamic Radial Load Rating for Bearing Combinations

##### 5.1.2.1 Two Single-Row Radial Contact Ball Bearings Operating as a Unit

When calculating the basic dynamic radial load rating for two similar single-row radial contact ball bearings mounted side by side on the same shaft, such that they operate as a unit (paired mounting), the pair is considered as one double-row radial contact ball bearing.

##### 5.1.2.2 Back-to-back and Face-to-face Arrangements of Single-Row Angular Contact Ball Bearings

When calculating the basic dynamic radial load rating for two similar single-row angular contact ball bearings mounted side by side on the same shaft, such that they operate as a unit (paired mounting), in a back-to-back or face-to-face arrangement, the pair is considered as one double-row angular contact ball bearing.

##### 5.1.2.3 Tandem Arrangement

The basic dynamic radial load rating for two or more similar single-row radial contact ball bearings, or two or more similar angular contact ball bearings, mounted side by side on the same shaft, such that they operate as a unit (paired or stack mounting) in a tandem arrangement, is the number of bearings to the power of 0.7 times the rating of one single-row bearing. The bearings need to be properly manufactured and mounted for equal load distribution between them.

##### 5.1.2.4 Independently Replaceable Bearings in Tandem Arrangement

If, for some technical reason, the bearing arrangement is regarded as a number of single row bearings which are replaceable independently of each other, then 5.1.2.3 does not apply.



**TABLE 1. Part 1 – Metric Values for  $f_{cm}$  for Radial and Angular Contact Ball Bearings**

$\frac{D_w \cos \alpha^a)}{D_{pw}}$	$f_{cm}$ Use to obtain $C_r$ in newtons when $D_w$ and $D_{pw}$ are given in millimeters				
	Single Row Radial Contact Ball Bearings and Single Row and Double Row Angular Contact Ball Bearings and Insert Bearings <sup>b)</sup>	Filling Slot Ball Bearings	Double Row Radial Contact Ball Bearings	Single Row and Double Row Self-Aligning Ball Bearings	Single Row Radial Contact Separable Ball Bearings (Magneto Bearings)
0.01	37.83	32.01	35.75	12.87	12.22
0.02	46.54	39.38	44.07	16.12	15.21
0.03	52.39	44.33	49.66	18.59	17.42
0.04	56.94	48.18	53.95	20.67	19.37
0.05	60.71	51.37	57.46	22.49	21.06
0.06	63.83	54.01	60.45	24.18	22.62
0.07	66.43	56.21	62.92	25.87	24.05
0.08	68.64	58.08	65.00	27.43	25.35
0.09	70.59	59.73	66.82	28.99	26.78
0.10	72.15	61.05	68.38	30.42	27.95
0.11	73.58	62.26	69.68	31.85	29.25
0.12	74.75	63.25	70.85	33.28	30.42
0.13	75.66	64.02	71.76	34.58	31.72
0.14	76.44	64.68	72.41	36.01	32.89
0.15	77.09	65.23	72.93	37.31	34.06
0.16	77.48	65.56	73.45	38.61	35.23
0.17	77.74	65.78	73.71	39.91	36.27
0.18	77.87	65.89	73.84	41.21	37.44
0.19	78.00	66.00	73.84	42.38	38.61
0.20	77.87	65.89	73.84	43.55	39.65
0.21	77.74	65.78	73.58	44.72	40.69
0.22	77.48	65.56	73.45	45.76	41.73
0.23	77.09	65.23	73.06	46.93	42.77
0.24	76.70	64.90	72.67	47.84	43.81
0.25	76.18	64.46	72.15	48.75	44.85
0.26	75.66	64.02	71.63	49.66	45.76
0.27	75.01	63.47	70.98	50.44	46.67
0.28	74.23	62.81	70.33	51.22	47.58
0.29	73.58	62.26	69.68	51.87	48.36
0.30	72.80	61.60	68.90	52.39	49.14
0.31	71.89	60.83	68.12	52.78	49.92
0.32	70.98	60.06	67.34	53.17	50.57
0.33	70.07	59.29	66.43	53.43	51.22
0.34	69.16	58.52	65.52	53.56	51.74
0.35	68.12	57.64	64.61	53.69	52.13
0.36	67.21	56.87	63.57	53.69	52.52
0.37	66.17	55.99	62.66	53.56	52.91
0.38	65.00	55.00	61.62	53.30	53.04
0.39	63.96	54.12	60.58	52.91	53.17
0.40	62.92	53.24	59.54	52.52	53.17

<sup>a</sup> Values of  $f_{cm}$  for intermediate values of  $\frac{D_w \cos \alpha}{D_{pw}}$  are obtained by linear interpolation.

<sup>b</sup> Insert bearings are not in accordance with ISO 281.

**TABLE 1. Part 2 – Inch Values for  $f_{cm}$  for Radial and Angular Contact Ball Bearings**

$f_{cm}$ Use to obtain $C_r$ in pounds when $D_w$ and $D_{pw}$ are given in inches					
$\frac{D_w \cos \alpha^a)}{D_{pw}}$	Single Row Radial Contact Ball Bearings and Single Row and Double Row Angular Contact Ball Bearings and Insert Bearings <sup>b)</sup>	Filling Slot Ball Bearings	Double Row Radial Contact Ball Bearings	Single Row and Double Row Self-Aligning Ball Bearings	Single Row Radial Contact Separable Ball Bearings (Magneto Bearings)
0.01	2875	2433	2717	978	929
0.02	3537	2993	3349	1225	1156
0.03	3982	3369	3774	1413	1324
0.04	4327	3662	4100	1571	1472
0.05	4614	3904	4367	1709	1601
0.06	4851	4105	4594	1838	1719
0.07	5049	4272	4782	1966	1828
0.08	5217	4414	4940	2085	1927
0.09	5365	4539	5078	2203	2035
0.10	5483	4640	5197	2312	2124
0.11	5592	4732	5296	2421	2223
0.12	5681	4807	5385	2529	2312
0.13	5750	4866	5454	2628	2411
0.14	5809	4916	5503	2737	2500
0.15	5859	4957	5543	2836	2589
0.16	5888	4983	5582	2934	2677
0.17	5908	4999	5602	3033	2757
0.18	5918	5008	5612	3132	2845
0.19	5928	5016	5612	3221	2934
0.20	5918	5008	5612	3310	3013
0.21	5908	4999	5592	3399	3092
0.22	5888	4983	5582	3478	3171
0.23	5859	4957	5553	3567	3251
0.24	5829	4932	5523	3636	3330
0.25	5790	4899	5483	3705	3409
0.26	5750	4866	5444	3774	3478
0.27	5701	4824	5394	3833	3547
0.28	5641	4774	5345	3893	3616
0.29	5592	4732	5296	3942	3675
0.30	5533	4682	5236	3982	3735
0.31	5464	4623	5177	4011	3794
0.32	5394	4565	5118	4041	3843
0.33	5325	4506	5049	4061	3893
0.34	5256	4448	4980	4071	3932
0.35	5177	4381	4910	4080	3962
0.36	5108	4322	4831	4080	3992
0.37	5029	4255	4762	4071	4021
0.38	4940	4180	4683	4051	4031
0.39	4861	4113	4604	4021	4041
0.40	4782	4046	4525	3992	4041

<sup>a</sup> Values of  $f_{cm}$  for intermediate values of  $\frac{D_w \cos \alpha}{D_{pw}}$  are obtained by linear interpolation.

<sup>b</sup> Insert bearings are not in accordance with ISO 281.

## **5.2 Dynamic Equivalent Radial Load**

### **5.2.1 Dynamic Equivalent Radial Load for Single Bearings**

The dynamic equivalent radial load for radial and angular contact ball bearings, under constant radial and axial loads, is given by

$$P_r = X F_r + Y F_a$$

Values of X and Y are given in Table 2.

### **5.2.2 Dynamic Equivalent Radial Load for Bearing Combinations**

#### **5.2.2.1 Back-to-back and Face-to-face Arrangements of Single-row Angular Contact Ball Bearings**

When calculating the equivalent radial load for two similar single-row angular contact ball bearings mounted side by side on the same shaft, such that they operate as a unit (paired mounting) in a back-to-back or a face-to-face arrangement, the pair is considered as one double-row angular contact ball bearing.

NOTE: If two similar single-row radial contact ball bearings are operating in back-to-back or face-to-face arrangement, the user should consult the bearing manufacturer about calculation of equivalent radial load.

#### **5.2.2.2 Tandem Arrangement**

When calculating the equivalent radial load for two or more similar single-row radial contact ball bearings or two or more similar single-row angular contact ball bearings mounted side by side on the same shaft, such that they operate as a unit (paired or stack mounting) in a tandem arrangement, the values of X and Y for a single-row bearing shall be used.

The “relative axial load” (see Table 2) is established by using  $i = 1$  and  $F_a$  and  $C_{0r}$  values which both refer to one of the bearings only (even though the  $F_r$  and  $F_a$  values referring to the total loads are used for the calculation of the equivalent load for the complete arrangement).

## **5.3 Basic Rating Life**

### **5.3.1 Life Equation**

The basic rating life for a radial ball bearing is given by the life equation:

$$L_{10} = \left( \frac{C_r}{P_r} \right)^3$$

The values of  $C_r$  and  $P_r$  are calculated in accordance with 5.1 and 5.2.

This life equation is also used for the evaluation of the life of two or more single-row bearings operating as a unit, as referred to in 5.1.2. In this case, the load rating  $C_r$  is calculated for the complete bearing arrangement and the equivalent load  $P_r$  is calculated for the total loads acting on the arrangement, using the values of X and Y indicated in 5.2.2.



**TABLE 2. Part 1 – Metric Values for X and Y for Radial and Angular Contact Ball Bearings**

Use to obtain $P_r$ in newtons when $D_w$ is given in millimeters												
Bearing type	“Relative axial load” <sup>a) b)</sup>		Single row bearings				Double row bearings				e	
			$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$		$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$			
			X	Y	X	Y	X	Y	X	Y		
Radial contact ball bearings	$\frac{f_0 F_a^{c)}}{C_{0r}}$	$\frac{F_a}{i Z D_w^2}$										
	0.172	0.172				2.3				2.3	0.19	
	0.345	0.345				1.99				1.99	0.22	
	0.689	0.689				1.71				1.71	0.26	
	1.03	1.03				1.55				1.55	0.28	
	1.38	1.38	1	0	0.56	1.45	1	0	0.56	1.45	0.3	
	2.07	2.07				1.31				1.31	0.34	
	3.45	3.45				1.15				1.15	0.38	
	5.17	5.17				1.04				1.04	0.42	
	6.89	6.89				1				1	0.44	
Angular contact ball bearings	$\frac{f_0 i F_a^{c)}}{C_{0r}}$	$\frac{F_a}{Z D_w^2}$										
	$\alpha = 5^\circ$	0.173	0.172			For this type, use the X, Y and e values applicable to single-row radial contact ball bearings.			2.78		3.74	0.23
		0.346	0.345						2.4		3.23	0.26
		0.692	0.689						2.07		2.78	0.3
		1.04	1.03						1.87		2.52	0.34
		1.38	1.38	1	0				1.75	0.78	2.36	0.36
		2.08	2.07						1.58		2.13	0.4
		3.46	3.45						1.39		1.87	0.45
		5.19	5.17						1.26		1.69	0.5
		6.92	6.89						1.21		1.63	0.52
		$\alpha = 10^\circ$	0.175	0.172						1.88		2.18
	0.35		0.345				1.71		1.98		2.78	0.32
	0.7		0.689				1.52		1.76		2.47	0.36
	1.05		1.03				1.41		1.63		2.29	0.38
	1.4		1.38	1	0	0.46	1.34	1	1.55	0.75	2.18	0.4
	2.1		2.07				1.23		1.42		2	0.44
	3.5		3.45				1.1		1.27		1.79	0.49
	5.25		5.17				1.01		1.17		1.64	0.54
	7		6.89				1		1.16		1.63	0.54
	$\alpha = 15^\circ$		0.178	0.172				1.47		1.65		2.39
		0.357	0.345				1.4		1.57		2.28	0.4
		0.714	0.689				1.3		1.46		2.11	0.43
		1.07	1.03				1.23		1.38		2	0.46
		1.43	1.38	1	0	0.44	1.19	1	1.34	0.72	1.93	0.47
		2.14	2.07				1.12		1.26		1.82	0.5
		3.57	3.45				1.02		1.14		1.66	0.55
		5.35	5.17				1		1.12		1.63	0.56
		7.14	6.89				1		1.12		1.63	0.56
		$\alpha = 20^\circ$ $\alpha = 25^\circ$ $\alpha = 30^\circ$ $\alpha = 35^\circ$ $\alpha = 40^\circ$ $\alpha = 45^\circ$	---	---			0.43	1		1.09	0.7	1.63
	---		---			0.41	0.87		0.92	0.67	1.41	0.68
	---		---	1	0	0.39	0.76	1	0.78	0.63	1.24	0.8
	---		---			0.37	0.66		0.66	0.6	1.07	0.95
	---		---			0.35	0.57		0.55	0.57	0.93	1.14
	---		---			0.33	0.5		0.47	0.54	0.81	1.34
	Self-aligning ball bearings			1	0	0.4	$0.4 \cot \alpha$	1	$0.42 \cot \alpha$	0.65	$0.65 \cot \alpha$	$1.5 \tan \alpha$
	Single-row radial contact separable ball bearings (magneto bearings)			1	0	0.5	2.5	---	---	---	---	0.2

<sup>a</sup> Permissible maximum value depends on the bearing design (internal clearance and raceway groove depth). Use the first or second column depending on available information.

<sup>b</sup> Values of X, Y and e for intermediate “relative axial loads” and/or contact angles are obtained by linear interpolation.

<sup>c</sup> For values of  $f_0$  see Table 3.

**TABLE 2. Part 2 – Inch Values for X and Y for Radial and Angular Contact Ball Bearings**

Use to obtain $P_r$ in newtons when $D_w$ is given in millimeters												
Bearing type	“Relative axial load” <sup>a) b)</sup>		Single row bearings				Double row bearings				e	
			$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$		$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$			
			X	Y	X	Y	X	Y	X	Y		
Radial contact ball bearings	$\frac{f_0 F_a^{c)}}{C_{0r}}$	$\frac{F_a}{i Z D_w^2}$										
	24.94	24.94				2.3				2.3	0.19	
	50.03	50.03				1.99				1.99	0.22	
	99.91	99.91				1.71				1.71	0.26	
	149.35	149.35				1.55				1.55	0.28	
	200.1	200.1	1	0	0.56	1.45	1	0	0.56	1.45	0.3	
	300.15	300.15				1.31				1.31	0.34	
	500.25	500.25				1.15				1.15	0.38	
	749.65	749.65				1.04				1.04	0.42	
	999.05	999.05				1				1	0.44	
Angular contact ball bearings	$\frac{f_0 i F_a^{c)}}{C_{0r}}$	$\frac{F_a}{Z D_w^2}$										
	$\alpha = 5^\circ$	25.09	24.94			For this type, use the X, Y and e values applicable to single-row radial contact ball bearings.			2.78		3.74	0.23
		50.17	50.03					2.4		3.23	0.26	
		100.34	99.91					2.07		2.78	0.3	
		150.8	149.35					1.87		2.52	0.34	
		200.1	200.1	1	0			1	1.75	0.78	2.36	0.36
		301.6	300.15					1.58		2.13	0.4	
		501.7	500.25					1.39		1.87	0.45	
		752.55	749.65					1.26		1.69	0.5	
		1003.4	999.05					1.21		1.63	0.52	
		$\alpha = 10^\circ$	25.35	24.94						1.88		2.18
	50.75		50.03				1.71		1.98		2.78	0.32
	101.5		99.91				1.52		1.76		2.47	0.36
	152.25		149.35				1.41		1.63		2.29	0.38
	203		200.1	1	0	0.46	1.34	1	1.55	0.75	2.18	0.4
	304.5		300.15				1.23		1.42		2	0.44
	507.5		500.25				1.1		1.27		1.79	0.49
	761.25		749.65				1.01		1.17		1.64	0.54
	1015		999.05				1		1.16		1.63	0.54
	$\alpha = 15^\circ$		25.81	24.94				1.47		1.65		2.39
		51.77	50.03				1.4		1.57		2.28	0.4
		103.53	99.91				1.3		1.46		2.11	0.43
		155.15	149.35				1.23		1.38		2	0.46
		207.35	200.1	1	0	0.44	1.19	1	1.34	0.72	1.93	0.47
		310.3	300.15				1.12		1.26		1.82	0.5
		517.65	500.25				1.02		1.14		1.66	0.55
		775.75	749.65				1		1.12		1.63	0.56
		1035.3	999.05				1		1.12		1.63	0.56
		$\alpha = 20^\circ$ $\alpha = 25^\circ$ $\alpha = 30^\circ$ $\alpha = 35^\circ$ $\alpha = 40^\circ$ $\alpha = 45^\circ$	---	---			0.43	1		1.09	0.7	1.63
	---		---			0.41	0.87		0.92	0.67	1.41	0.68
	---		---	1	0	0.39	0.76	1	0.78	0.63	1.24	0.8
	---		---			0.37	0.66		0.66	0.6	1.07	0.95
	---		---			0.35	0.57		0.55	0.57	0.93	1.14
	---		---			0.33	0.5		0.47	0.54	0.81	1.34
	Self-aligning ball bearings			1	0	0.4	0.4 cot $\alpha$	1	0.42 cot $\alpha$	0.65	0.65 cot $\alpha$	1.5 tan $\alpha$
	Single-row radial contact separable ball bearings (magneto bearings)			1	0	0.5	2.5	---	---	---	---	0.2

<sup>a</sup> Permissible maximum value depends on the bearing design (internal clearance and raceway groove depth). Use the first or second column depending on available information.

<sup>b</sup> Values of X, Y and e for intermediate “relative axial loads” and/or contact angles are obtained by linear interpolation.

<sup>c</sup> For values of  $f_0$  see Table 3.

### 5.3.2 Loading Restriction on the Life Equation

The life equation gives satisfactory results for a broad range of bearing loads. However, extra-heavy loads may cause detrimental plastic deformations at the rolling element/raceway contacts. The user should therefore consult the bearing manufacturer to establish the applicability of the life equation in cases where  $P_r$  exceeds  $C_{0r}$  or  $0.5 C_r$ , whichever is the smaller.

Very light loads may cause different failure modes to occur. These failure modes are not covered by this standard.

## 5.4 Basic Static Radial Load Rating

### 5.4.1 Basic Static Radial Load Rating for Single Bearings

The basic static radial load rating for radial ball bearings is given by the equation:

$$C_{0r} = f_0 i Z D_w^2 \cos \alpha$$

where the values of  $f_0$  are given in Table 3.

The equation applies to bearings with a cross-sectional raceway groove radius not larger than  $0.52D_w$  in radial and angular contact ball bearing inner rings, and  $0.53D_w$  in radial and angular contact ball bearing outer rings and self-aligning ball bearing inner rings.

The load-carrying ability of a bearing is not necessarily increased by the use of a smaller groove radius, but is reduced by the use of a groove radius larger than those indicated in the previous paragraph. In the latter case, a correspondingly reduced value of  $f_0$  shall be used.

### 5.4.2 Basic Static Radial Load Rating for Bearing Combinations

#### 5.4.2.1 Two Single-row Radial Contact Ball Bearings Operating as a Unit

The basic static radial load rating for two similar single-row radial contact ball bearings mounted side by side on the same shaft, such that they operate as a unit (paired mounting), is twice the basic static radial load rating of one single-row bearing.

#### 5.4.2.2 Back-to-back and Face-to-face Arrangements of Single-row Angular Contact Ball Bearings

The basic static radial load rating for two similar single-row angular contact ball bearings mounted side by side on the same shaft, such that they operate as a unit (paired mounting), in a back-to-back or face-to-face arrangement, is twice the basic static radial load rating of one single-row bearing.

#### 5.4.2.3 Tandem Arrangement

The basic static radial load rating for two or more similar single-row radial contact ball bearings or two or more similar single-row angular contact ball bearings mounted side by side on the same shaft, such that they operate as a unit (paired or stack mounting) in a tandem arrangement, properly manufactured and mounted for equal load distribution, is the number of bearings multiplied by the basic static radial load rating of one single-row bearing.



**TABLE 3. Values for  $f_0$  for Ball Bearings <sup>a)</sup>**

$\frac{D_w \cos \alpha^b}{D_{pw}}$	$f_0$ Use to obtain $C_{0r}$ in newtons when $D_w$ and $D_{pw}$ are given in millimeters			$f_0$ Use to obtain $C_{0r}$ in pounds when $D_w$ and $D_{pw}$ are given in inches		
	Radial Ball Bearings		Thrust Ball Bearings	Radial Ball Bearings		Thrust Ball Bearings
	Radial and Angular Contact	Self-aligning		Radial and Angular Contact	Self-aligning	
0	14.7	1.9	61.6	2131.5	275.5	8932.0
0.01	14.9	2.0	60.8	2160.5	290.0	8816.0
0.02	15.1	2.0	59.9	2189.5	290.0	8685.5
0.03	15.3	2.1	59.1	2218.5	304.5	8569.5
0.04	15.5	2.1	58.3	2247.5	304.5	8453.5
0.05	15.7	2.1	57.5	2276.5	304.5	8337.5
0.06	15.9	2.2	56.7	2305.5	319.0	8221.5
0.07	16.1	2.2	55.9	2334.5	319.0	8105.5
0.08	16.3	2.3	55.1	2363.5	333.5	7989.5
0.09	16.5	2.3	54.3	2392.5	333.5	7873.5
0.10	16.4	2.4	53.5	2378.0	348.0	7757.5
0.11	16.1	2.4	52.7	2334.5	348.0	7641.5
0.12	15.9	2.4	51.9	2305.5	348.0	7525.5
0.13	15.6	2.5	51.2	2262.0	362.5	7424.0
0.14	15.4	2.5	50.4	2233.0	362.5	7308.0
0.15	15.2	2.6	49.6	2204.0	377.0	7192.0
0.16	14.9	2.6	48.8	2160.5	377.0	7076.0
0.17	14.7	2.7	48.0	2131.5	391.5	6960.0
0.18	14.4	2.7	47.3	2088.0	391.5	6858.5
0.19	14.2	2.8	46.5	2059.0	406.0	6742.5
0.20	14.0	2.8	45.7	2030.0	406.0	6626.5
0.21	13.7	2.8	45.0	1986.5	406.0	6525.0
0.22	13.5	2.9	44.2	1957.5	420.5	6409.0
0.23	13.2	2.9	43.5	1914.0	420.5	6307.5
0.24	13.0	3.0	42.7	1885.0	435.0	6191.5
0.25	12.8	3.0	41.9	1856.0	435.0	6075.5
0.26	12.5	3.1	41.2	1812.5	449.5	5974.0
0.27	12.3	3.1	40.5	1783.5	449.5	5872.5
0.28	12.1	3.2	39.7	1754.5	464.0	5756.5
0.29	11.8	3.2	39.0	1711.0	464.0	5655.0
0.30	11.6	3.3	38.2	1682.0	478.5	5539.0
0.31	11.4	3.3	37.5	1653.0	478.5	5437.5
0.32	11.2	3.4	36.8	1624.0	493.0	5336.0
0.33	10.9	3.4	36.0	1580.5	493.0	5220.0
0.34	10.7	3.5	35.3	1551.5	507.5	5118.5
0.35	10.5	3.5	34.6	1522.5	507.5	5017.0
0.36	10.3	3.6	--	1493.5	522.0	--
0.37	10.0	3.6	--	1450.0	522.0	--
0.38	9.8	3.7	--	1421.0	536.5	--
0.39	9.6	3.8	--	1392.0	551.0	--
0.40	9.4	3.8	--	1363.0	551.0	--

<sup>a</sup> This table is based on the Hertzian point contact equation with a modulus of elasticity of  $2.07 \times 10^5$  MPa ( $30 \times 10^6$  PSI) and a Poisson's ratio of 0.3. It is assumed that the load distribution results in a maximum ball load of  $\frac{5F_r}{Z \cos \alpha}$  for radial ball bearings and a

maximum ball load of  $\frac{F_a}{Z \sin \alpha}$  for thrust ball bearings.

<sup>b</sup> Values of  $f_0$  for intermediate values of  $\frac{D_w \cos \alpha}{D_{pw}}$  are obtained by linear interpolation.

## 5.5 Static Equivalent Radial Load

### 5.5.1 Static Equivalent Radial Load for Single Bearings

The static equivalent radial load for radial ball bearings is the greater of the two values given by the equations:

$$P_{0r} = X_0 F_r + Y_0 F_a$$

$$P_{0r} = F_r$$

where the values of factors  $X_0$  and  $Y_0$  are given in Table 4.

**TABLE 4. Values for  $X_0$  and  $Y_0$  for Radial Ball Bearings**

Bearing Type	Single Row <sup>b)</sup> Bearings		Double Row Bearings		
	X <sub>0</sub>	Y <sub>0</sub> <sup>c)</sup>	X <sub>0</sub>	Y <sub>0</sub> <sup>c)</sup>	
Radial Contact Ball Bearings <sup>a) b)</sup>	0.6	0.5	0.6	0.5	
Angular Contact Ball Bearings	α = 5°	0.5	0.52	1	1.04
	α = 10°	0.5	0.50	1	1.00
	α = 15°	0.5	0.46	1	0.92
	α = 20°	0.5	0.42	1	0.84
	α = 25°	0.5	0.38	1	0.76
	α = 30°	0.5	0.33	1	0.66
	α = 35°	0.5	0.29	1	0.58
	α = 40°	0.5	0.26	1	0.52
	α = 45°	0.5	0.22	1	0.44
Self-aligning Ball Bearings, α ≠ 0°	0.5	0.22 cota	1	0.44 cota	

<sup>a)</sup> The permissible maximum value of  $F_a/C_{0r}$  depends on bearing design (internal clearance and raceway groove depth).

<sup>b)</sup>  $P_{0r}$  is always  $\geq F_r$

<sup>c)</sup> Values of  $Y_0$  for intermediate contact angles are obtained by linear interpolation.

### 5.5.2 Static Equivalent Radial Load for Bearing Combinations

#### 5.5.2.1 Back-to-back and Face-to-face Arrangements of Single-row Angular Contact Ball Bearings

When calculating the static equivalent radial load for two similar angular contact ball bearings mounted side by side on the same shaft, such that they operate as a unit (paired mounting) in a back-to-back or a face-to-face arrangement, the  $X_0$  and  $Y_0$  values for a double-row bearing and the  $F_r$  and  $F_a$  values for the total loads on the arrangement shall be used.

#### 5.5.2.2 Tandem Arrangement

When calculating the static equivalent radial load for two or more similar single-row radial contact ball bearings or two or more similar single-row angular contact ball bearings mounted side by side on the same shaft, such that they operate as a unit (paired or stack mounting) in a tandem arrangement, the  $X_0$  and  $Y_0$  values for a single-row bearing and the  $F_r$  and  $F_a$  values for the total loads on the arrangement shall be used.

## 6. THRUST BALL BEARINGS

## 6.1 Basic Dynamic Axial Load Rating

### 6.1.1 Basic Dynamic Axial Load Rating for Single-row Bearings

The basic dynamic axial load rating for single-row, single-direction or double-direction thrust ball bearings is given by;

for  $D_w \leq 25.4\text{mm}$  (1 inch)

$$\text{for } \alpha=90^\circ: C_a = f_{cm} Z^{2/3} D_w^{1.8}$$

$$\text{for } \alpha \neq 90^\circ: C_a = f_{cm} (\cos \alpha)^{0.7} Z^{2/3} D_w^{1.8} \tan \alpha$$

for  $D_w > 25.4\text{mm}$  (1 inch)

$$\text{for } \alpha=90^\circ: C_a = 3.647 f_{cm} Z^{2/3} D_w^{1.4} \quad (\text{metric})$$

$$C_a = f_{cm} Z^{2/3} D_w^{1.4} \quad (\text{inch})$$

$$\text{for } \alpha \neq 90^\circ: C_a = 3.647 f_{cm} (\cos \alpha)^{0.7} Z^{2/3} D_w^{1.4} \tan \alpha \quad (\text{metric})$$

$$C_a = f_{cm} (\cos \alpha)^{0.7} Z^{2/3} D_w^{1.4} \tan \alpha \quad (\text{inch})$$

where  $Z$  is the number of balls carrying load in one direction.

Values of  $f_{cm}$  are given in Table 5 and apply to bearings with cross-sectional raceway groove radii not larger than  $0.54 D_w$ .

The load-carrying ability of a bearing is not necessarily increased by the use of a smaller groove radius, but is reduced by the use of a larger groove radius than that indicated above. In the latter case, a correspondingly reduced value of  $f_{cm}$  shall be used.

### 6.1.2 Basic Dynamic Axial Load Rating for Bearings with Two or More Rows of Balls.

The basic dynamic axial load rating for thrust ball bearings with two or more rows of similar balls carrying load in the same direction is given by

$$C_a = (Z_1 + Z_2 + \dots + Z_n) \times \left[ \left( \frac{Z_1}{C_{a1}} \right)^{10/3} + \left( \frac{Z_2}{C_{a2}} \right)^{10/3} + \dots + \left( \frac{Z_n}{C_{an}} \right)^{10/3} \right]^{-3/10}$$

The load ratings  $C_{a1}, C_{a2}, \dots, C_{an}$  for the rows with  $Z_1, Z_2, \dots, Z_n$  balls are calculated from the appropriate single-row bearing equation given in 6.1.1.



**TABLE 5. Values for  $f_{cm}$  for Thrust Ball Bearings**

Use to obtain $C_a$ in newtons when $D_w$ and $D_{pw}$ are given in millimeters						Use to obtain $C_a$ in pounds when $D_w$ and $D_{pw}$ are given in inches					
$\frac{D_w^{a)}}{D_{pw}}$	$f_{cm}$	$\frac{D_w \cos \alpha^{a)}}{D_{pw}}$	$f_{cm}$			$\frac{D_w^{a)}}{D_{pw}}$	$f_{cm}$	$\frac{D_w \cos \alpha^{a)}}{D_{pw}}$	$f_{cm}$		
	$\alpha = 90^\circ$		$\alpha = 45^\circ{}^b)$	$\alpha = 60^\circ$	$\alpha = 75^\circ$		$\alpha = 90^\circ$		$\alpha = 45^\circ{}^b)$	$\alpha = 60^\circ$	$\alpha = 75^\circ$
0.01	47.71	0.01	54.73	50.96	48.49	0.01	3626	0.01	4159	3873	3685
0.02	58.76	0.02	67.21	62.53	59.67	0.02	4466	0.02	5108	4752	4535
0.03	66.43	0.03	75.66	70.46	67.21	0.03	5049	0.03	5750	5355	5108
0.04	72.41	0.04	82.29	76.57	72.93	0.04	5503	0.04	6254	5819	5543
0.05	77.35	0.05	87.49	81.38	77.61	0.05	5879	0.05	6649	6185	5898
0.06	81.77	0.06	91.91	85.54	81.51	0.06	6215	0.06	6985	6501	6195
0.07	85.54	0.07	95.55	88.92	84.76	0.07	6501	0.07	7262	6758	6442
0.08	89.05	0.08	98.67	91.91	87.49	0.08	6768	0.08	7499	6985	6649
0.09	92.30	0.09	101.40	94.38	89.96	0.09	7015	0.09	7706	7173	6837
0.10	95.29	0.10	103.61	96.46	91.91	0.10	7242	0.10	7874	7331	6985
0.11	98.02	0.11	105.43	98.15	--	0.11	7450	0.11	8013	7459	--
0.12	100.62	0.12	106.99	99.58	--	0.12	7647	0.12	8131	7568	--
0.13	103.09	0.13	108.29	100.75	--	0.13	7835	0.13	8230	7657	--
0.14	105.43	0.14	109.33	101.79	--	0.14	8013	0.14	8309	7736	--
0.15	107.51	0.15	110.11	102.44	--	0.15	8171	0.15	8368	7785	--
0.16	109.72	0.16	110.63	102.96	--	0.16	8339	0.16	8408	7825	--
0.17	111.67	0.17	111.02	103.35	--	0.17	8487	0.17	8438	7855	--
0.18	113.62	0.18	111.15	103.48	--	0.18	8635	0.18	8447	7864	--
0.19	115.44	0.19	111.15	103.48	--	0.19	8773	0.19	8447	7864	--
0.20	117.26	0.20	111.02	103.35	--	0.20	8912	0.20	8438	7855	--
0.21	118.95	0.21	110.76	--	--	0.21	9040	0.21	8418	--	--
0.22	120.64	0.22	110.37	--	--	0.22	9169	0.22	8388	--	--
0.23	122.33	0.23	109.85	--	--	0.23	9297	0.23	8349	--	--
0.24	123.89	0.24	109.20	--	--	0.24	9416	0.24	8299	--	--
0.25	125.32	0.25	108.42	--	--	0.25	9524	0.25	8240	--	--
0.26	126.88	0.26	107.64	--	--	0.26	9643	0.26	8181	--	--
0.27	128.31	0.27	106.60	--	--	0.27	9752	0.27	8102	--	--
0.28	129.74	0.28	105.69	--	--	0.28	9860	0.28	8032	--	--
0.29	131.04	0.29	104.52	--	--	0.29	9959	0.29	7944	--	--
0.30	132.47	0.30	103.48	--	--	0.30	10068	0.30	7864	--	--
0.31	133.77	--	--	--	--	0.31	10167	--	--	--	--
0.32	135.07	--	--	--	--	0.32	10265	--	--	--	--
0.33	136.24	--	--	--	--	0.33	10354	--	--	--	--
0.34	137.54	--	--	--	--	0.34	10453	--	--	--	--
0.35	138.71	--	--	--	--	0.35	10542	--	--	--	--

<sup>a</sup> Values of  $f_{cm}$  for  $\frac{D_w}{D_{pw}}$  or  $\frac{D_w \cos \alpha}{D_{pw}}$  and/or contact angles other than those shown in the table are obtained by linear interpolation.

<sup>b</sup> For thrust bearings  $\alpha > 45^\circ$ . Values for  $\alpha = 45^\circ$  permit interpolation of values for  $\alpha$  between  $45^\circ$  and  $60^\circ$ .

## 6.2 Dynamic Equivalent Axial Load

The dynamic equivalent axial load for thrust ball bearings with  $\alpha \neq 90^\circ$ , under constant radial and axial loads, is given by

$$P_a = X F_r + Y F_a$$

where values of X and Y are given in Table 6.

Thrust ball bearings with  $\alpha = 90^\circ$  can support axial loads only. The dynamic equivalent axial load for this type of bearing is given by

$$P_a = F_a$$

**TABLE 6. Values for X and Y for Thrust Ball Bearings**

$\alpha$ <sup>a)</sup>	Single-direction bearings <sup>b)</sup>		Double-direction bearings				e
	$\frac{F_a}{F_r} > e$		$\frac{F_a}{F_r} \leq e$		$\frac{F_a}{F_r} > e$		
	X	Y	X	Y	X	Y	
45° <sup>c)</sup>	0.66	1	1.18	0.59	0.66	1	1.25
50°	0.73		1.37	0.57	0.73		1.49
55°	0.81		1.60	0.56	0.81		1.79
60°	0.92		1.90	0.55	0.92		2.17
65°	1.06		2.30	0.54	1.06		2.68
70°	1.28		2.90	0.53	1.28		3.43
75°	1.66		3.89	0.52	1.66		4.67
80°	2.43		5.86	0.52	2.43		7.09
85°	4.80		11.75	0.51	4.80		14.29
$\alpha \neq 90^\circ$	$1.25 \tan \alpha \left( 1 - \frac{2}{3} \sin \alpha \right)$	1	$\frac{20}{13} \tan \alpha \left( 1 - \frac{1}{3} \sin \alpha \right)$	$\frac{10}{13} \left( 1 - \frac{1}{3} \sin \alpha \right)$	$1.25 \tan \alpha \left( 1 - \frac{2}{3} \sin \alpha \right)$	1	$1.25 \tan \alpha$

<sup>a</sup> Values of X, Y and e for intermediate values of  $\alpha$  are obtained by linear interpolation.

<sup>b</sup>  $\frac{F_a}{F_r} \leq e$  is unsuitable for single-direction bearings.

<sup>c</sup> For thrust bearings,  $\alpha > 45^\circ$ . Values for  $\alpha = 45^\circ$  are given to permit interpolation of values for  $\alpha$  between  $45^\circ$  and  $50^\circ$ .

## 6.3 Basic Rating Life

### 6.3.1 Life Equation

The basic rating life for a thrust ball bearing is given by the life equation:

$$L_{10} = \left( \frac{C_a}{P_a} \right)^3$$

The values of  $C_a$  and  $P_a$  are calculated in accordance with 6.1 and 6.2.

### 6.3.2 Loading Restriction on the Life Equation

The life equation gives satisfactory results for a broad range of bearing loads. However, extra-heavy loads may cause detrimental plastic deformations at the ball/raceway contacts. The user should therefore consult the bearing manufacturer to establish the applicability of the life equation in cases where  $P_a$  exceeds  $0.5 C_a$ .

Very light loads may cause different failure modes to occur. These failure modes are not covered by this standard.

## 6.4 Basic Static Axial Load Rating

The basic static axial load rating for single-direction and double-direction thrust ball bearings is given by the equation:

$$C_{0a} = f_0 Z D_w^2 \sin \alpha$$

Where the values of  $f_0$  are given in Table 3 and  $Z$  is the number of balls carrying load in one direction.

The equation applies to bearings with cross-sectional raceway groove radii not larger than  $0.54 D_w$ .

The load-carrying ability of a bearing is not necessarily increased by the use of a smaller groove radius, but is reduced by the use of a larger groove radius. In the latter case, a correspondingly reduced value of  $f_0$  shall be used.

## 6.5 Static Equivalent Axial Load

The static equivalent axial load for thrust ball bearings with  $\alpha \neq 90^\circ$  is given by the equation:

$$P_{0a} = 2.3 F_r \tan \alpha + F_a$$

This equation is valid for all ratios of radial load to axial load in the case of double-direction bearings. For single-direction bearings, it is valid where  $F_r/F_a \leq 0.44 \cot \alpha$  and gives satisfactory but less conservative values of  $P_{0a}$  for  $F_r/F_a$  up to  $0.67 \cot \alpha$ .

Thrust ball bearings with  $\alpha = 90^\circ$  can support axial loads only. The static equivalent axial load for this type of bearing is given by the equation:

$$P_{0a} = F_a$$



## 7. STATIC SAFETY FACTOR

### 7.1 General

The suitability of a bearing selected for heavily loaded applications should be checked to ensure that its basic static load rating is adequate. This can be determined with the aid of the static safety factor  $S_0$ , which is given by the following equations:

$$\text{For radial ball bearings: } S_0 = \frac{C_{0r}}{P_{0r}}$$

$$\text{For thrust ball bearings: } S_0 = \frac{C_{0a}}{P_{0a}}$$

Where the bearing is dynamically loaded and the selection has been made on the basis of life, it is also advisable to check that the basic static load rating is adequate for attaining the performance requirements of the application.

The guideline values of  $S_0$  given in Table 7 for various types of operation and application requirements regarding smooth and vibration-free running are applicable to rotating bearings and are based on experience.

For other specific operating conditions, the bearing manufacturer should be consulted for guidance on the applicable  $S_0$  values.

**TABLE 7. Guideline Values for the Static Safety Factor  $S_0$  for Ball Bearings**

Type of Operation	$S_0$ min.
Quiet-running applications: smooth-running, vibration-free, high rotational accuracy	2
Normal-running applications: smooth-running, vibration-free, normal rotational accuracy	1
Applications subjected to shock loads: pronounced shock loads <sup>a)</sup>	1.5

<sup>a</sup> Where the magnitude of the load is not known, values of  $S_0$  which are at least 1.5 should be used. If the magnitude of the shock loads is known exactly, smaller values of  $S_0$  can be applied.

## 8. ADJUSTED RATING LIFE

### 8.1 General

It is often satisfactory to use the basic rating life  $L_{10}$  as a criterion of bearing performance. This life is associated with 90% reliability, with commonly used high quality material, good manufacturing quality, and with conventional operating conditions.

However, for many applications it may be desirable to calculate the life for a different level of reliability and/or for special bearing properties and operating conditions which deviate from the conventional in such a way that it is justified to take their influence into special consideration.

The adjusted rating life,  $L_{na}$ , i.e. the basic rating life adjusted for a reliability of  $(100-n)\%$ , for special bearing properties and for specific operating conditions, is given by

$$L_{na} = a_1 a_2 a_3 L_{10}$$

Values of  $a_1$  are given in Table 8. Values of  $a_2$  and  $a_3$  are discussed in 8.4 and 8.5. The value of  $L_{10}$  is calculated in accordance with 5.3 and 6.3.

## 8.2 Limitations

In addition to the required fatigue life, other factors, such as maximum permissible bearing deflection and minimum shaft and housing strength, should be given due consideration when selecting the size of bearings for a given application. Particular discretion shall be exercised when using adjusted rating life values which are based on values of  $a_2$  and  $a_3$  greater than 1.

## 8.3 Life Adjustment Factor for Reliability, $a_1$

Reliability is defined in 3.2. The adjusted rating life is calculated in accordance with 8.1. Values of the life adjustment factor  $a_1$  are given in Table 8.

**TABLE 8. Life Adjustment Factor for Reliability,  $a_1$**

Reliability %	$L_{na}$	$a_1$
90	$L_{10a}$	1
95	$L_{5a}$	0.64
96	$L_{4a}$	0.55
97	$L_{3a}$	0.47
98	$L_{2a}$	0.37
99	$L_{1a}$	0.25
99.2	$L_{0.8a}$	0.22
99.4	$L_{0.6a}$	0.19
99.6	$L_{0.4a}$	0.16
99.8	$L_{0.2a}$	0.12
99.9	$L_{0.1a}$	0.093
99.92	$L_{0.08a}$	0.087
99.94	$L_{0.06a}$	0.080
99.95	$L_{0.05a}$	0.077

## 8.4 Life Adjustment Factor for Special Bearing Properties, $a_2$

A bearing may acquire special properties, with regard to life, by the use of a special type and quality of material and/or special manufacturing processes and/or special design. Such special life properties are taken into account by the application of the life adjustment factor  $a_2$ .

The present state of knowledge does not make it possible to define relationships between the values of  $a_2$  and quantifiable characteristics of the material or bearing raceway geometry, for example. The values of  $a_2$  have therefore to be based on experience, and may usually be obtained from the manufacturer of the bearing.

The use of a certain steel analysis and/or process as such is not sufficient justification for the use of an  $a_2$  value other than 1. Values of  $a_2$  greater than 1, may however be applicable to bearings made of steel of particularly low impurity content or of special analysis. However, if a reduced life is expected because of a hardness reduction caused by special heat treatment, this should be considered by the selection of a correspondingly reduced  $a_2$  value.

A special design involving an increased or reduced uniformity of the stress in the contacts between rolling elements and raceways should also be considered in the selection of the value of  $a_2$ .

It may not be assumed that the use of a special material, process or design will overcome a deficiency in lubrication. Values of  $a_2$  greater than 1, should therefore normally not be applied if  $a_3$  is less than 1, because of such deficiency.

## 8.5 Life Adjustment Factor for Operating Conditions, $a_3$

### 8.5.1 General

Of the operating conditions directly influencing bearing life, the direction and magnitude of the load are considered in the calculation of the equivalent load, (5.2, 5.5, 6.2 and 6.5), and deviations from normal load distribution are discussed in 4.3

Operating conditions which remain to be taken into account here include the adequacy of the lubrication (at the operating speed and temperature), presence of foreign matter, conditions causing changes in material properties (for example high temperature causing reduced hardness) and mounting conditions. The influence on bearing life of such conditions may be taken into account by the introduction of a life adjustment factor  $a_3$ .

The calculation of basic rating life in this standard assumes that the lubrication is normal, i.e. that the lubricant film in the rolling element/raceway contacts has a thickness which is equal to or slightly greater than the composite roughness of the contact surfaces ( $\Lambda \geq 1$ ). Where this requirement is fulfilled,  $a_3$  is equal to 1, provided a lower value does not apply, for example because of a change in material properties caused by the operating conditions.

Values of  $a_3$  greater than 1 may be considered only where the lubrication conditions are so favorable that the probability of failure caused by surface distress is greatly reduced. Bearing manufacturers should be consulted for recommendations regarding appropriate values of  $a_3$  to be used in the calculation of adjusted rating life in accordance with 8.1.

Insert bearings are normally mounted to the shaft with a loose fit. Due to the possible variation which may occur in the shaft fit, the  $a_3$  factor should include a mounting factor multiplier of 0.456 unless otherwise recommended by the bearing manufacturer.

In addition to the guidelines listed in this section, additional lubricant parameters commonly used for calculation of the  $a_3$  factor are described in 8.5.2.

### 8.5.2 Viscosity Ratio

#### 8.5.2.1 Calculation of Viscosity Ratio

The effectiveness of the lubricant is primarily determined by the degree of surface separation between the rolling contact surfaces. If an adequate lubricant separation film is to be formed, the lubricant must have a given minimum viscosity when the application has reached its operating temperature. The condition of the lubricant separation is described by the viscosity ratio,  $\kappa$ , as the ratio of the actual kinematic viscosity,  $v$ , to the reference kinematic viscosity,  $v_1$ . The actual kinematic viscosity,  $v$  is considered when the lubricant is at operating temperature.

$$\kappa = \frac{v}{v_1}$$

In order to form an adequate lubricant film between the rolling contact surfaces, the lubricant must retain a certain minimum viscosity when the lubricant is at operating temperature. The bearing life may be extended by increasing the actual kinematic viscosity  $v$ .

The reference kinematic viscosity,  $v_1$ , can be estimated by means of the diagram in Figure 1, depending on the bearing speed and pitch diameter  $D_{pw}$ , [the mean bearing diameter (average of bearing bore and OD) can also be used] or be calculated with the following equations;

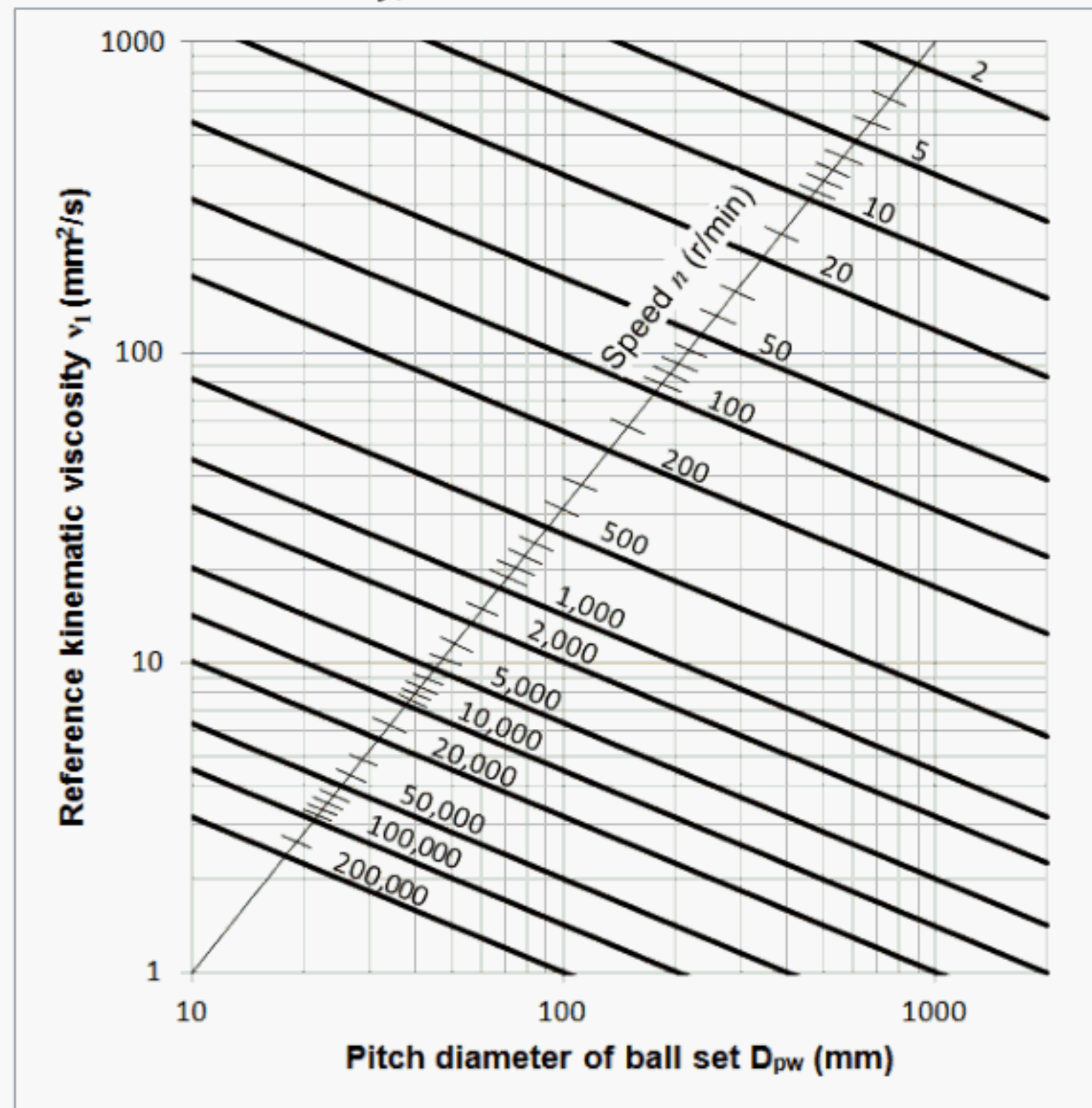
$$v_1 = 45000 \, n^{-0.83} D_{pw}^{-0.5} \quad \text{for } n < 1000 \text{ r/min}$$

$$v_1 = 4500 \, n^{-0.5} D_{pw}^{-0.5} \quad \text{for } n \geq 1000 \text{ r/min}$$

Use to calculate  $v_1$  in  $\text{mm}^2/\text{s}$  for values of  $D_{pw}$  given in mm.



**FIGURE 1. Reference Kinematic Viscosity,  $\nu_1$**



#### 8.5.2.2 Restriction of the Calculation of the Viscosity Ratio

The calculation of  $\kappa$  is based on mineral oils and on bearing raceway surfaces machined with good manufacturing quality.

The diagram in Figure 1 can also be used approximately for synthetic oils by using a lower pressure-viscosity coefficient than that used for mineral oils so the same oil film is built up at a different temperature, provided both oil types have the same reference viscosity at 40 °C.

If, however, there is need for a more detailed calculation of the  $\kappa$  value for cases of special raceway finishing, specific pressure-viscosity coefficients, or specific oil densities, the film parameter,  $\Lambda$ , can be applied. The parameter,  $\Lambda$ , is usually defined as the ratio of film thickness divided by composite surface roughness of the raceway and rolling element finishes. An approximate relationship between  $\kappa$  and  $\Lambda$  is given by the following equation:

$$\kappa \approx \Lambda^{1.3}$$

#### 8.5.2.3 Grease Lubrication

The diagram in Figure 1 applies equally to the base oil viscosity of greases. With grease lubrication, the contacts may be operating in a severely starved condition because of the poor bleeding capability of the grease leading to poor lubrication and possible reduction of life.

In these cases, a  $\Lambda$  parameter calculated with a reduced film thickness may need to be employed to approximate a  $\kappa$  value to be used in Figure 1.

#### 8.5.3 Calculation of Life under Low Load Conditions

Typically, fatigue lives in field applications which operate under light loads (contact stresses less than 1500 Mpa or 218 ksi), are much higher than lives predicted using load ratings. In such cases, a "low load" factor can be employed as part of the  $a_3$  life adjustment factor to calculate extended life. Such a low load factor, however, should not be employed if inadequate manufacturing quality, bearing material quality and/or lubrication exists. The bearing manufacturer should be consulted for recommendations regarding appropriate values of  $a_3$ .