TECHNICAL REPORT

ISO/TR 1281-1

Second edition
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Rolling bearings — Explanatory notes on ISO 281 —

Part 1:
Basic dynamic load rating and basic rating life

Roulements — Notes explicatives sur l'ISO 281 —

Partie 1: Charges dynamiques de base et durée nominale de base

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular, the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation of the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT), see www.iso.org/iso/foreword.html.

This document was prepared by Technical Committee ISO/TC 4, *Rolling bearings*, Subcommittee SC 8, *Load ratings and life*.

This second edition cancels and replaces the Technical Corrigendum 1 (ISO/TR 1281-1:2008/Cor 1:2009) and the first edition (ISO/TR 1281-1:2008), which has been technically revised.

The main changes compared to the previous edition are as follows:

— The old Clause 7 "Life adjustment factor for reliability" of ISO/TR 1281-1:2008 has been deleted, this subject is covered in ISO/TR 1281-2 (see ISO/TR 1281-1:2008/Cor 1:2009).

— The derivation of the old Formulae (29) and (46) [Formulae (28) and (45) in this edition] has been corrected.

— Typing errors have been corrected in Formulae (30) and (31) and in the derivation of the factor Y_3.

A list of all parts in the ISO/TR 1281 series can be found on the ISO website.

Any feedback or questions on this document should be directed to the user's national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

ISO/R281:1962

A first discussion on an international level of the question of standardizing calculation methods for load ratings of rolling bearings took place at the 1934 conference of the International Federation of the National Standardizing Associations (ISA). When ISA held its last conference in 1939, no progress had been made. However, in its 1945 report on the state of rolling bearing standardization, the ISA 4 Secretariat included proposals for definition of concepts fundamental to load rating and life calculation standards. The definitions it contained are in essence those given in ISO 281:2007 for the concepts "life" and "basic dynamic load rating" (now divided into "basic dynamic radial load rating" and "basic dynamic axial load rating").

In 1946, on the initiative of the Anti-Friction Bearing Manufacturers Association (AFBMA), New York, discussions of load rating and life calculation standards started between industries in the USA and Sweden. Chiefly on the basis of the results appearing in Reference [5], an AFBMA standard, *Method of evaluating load ratings of annular ball bearings*[3], was worked out and published in 1949. On the same basis, the member body for Sweden presented, in February 1950, a first proposal to ISO, "Load rating of ball bearings".

In view of the results of both further research and a modification to the AFBMA standard in 1950, as well as interest in roller bearing rating standards, in 1951, the member body for Sweden submitted a modified proposal for rating of ball bearings as well as a proposal for rating of roller bearings.

Load rating and life calculation methods were then studied. Reference [6] was then of considerable use, serving as a major basis for the sections regarding roller bearing rating.

ISO 281-1:1977

In 1964, in view of the development of improved bearing steels, the time had come to review ISO/R281 and submitted a proposal

In 1969, on the other hand, TC 4 followed a suggestion by the member body for Japan and reconstituted its WG 3, giving it the task of revising ISO/R281. The AFBMA load rating working group had at this time started revision work.

The major part of ISO 281-1:1977 constituted a re-publication of ISO/R281, the substance of which had been only very slightly modified. However, based mainly on American investigations during the 1960s, a new clause was added, dealing with adjustment of rating life for reliability other than 90 % and for material and operating conditions.

Furthermore, supplementary background information regarding the derivation of mathematical expressions and factors given in ISO 281-1:1977 was published as ISO/TR 8646:1985.

ISO 281:1990

ISO 281:1990 was published as "First edition" and entitled "Dynamic load ratings and rating life". It is referred to as the "technical revision" of ISO 281-1:1977. The new rating factor b_m for "contemporary, normally used material and manufacturing quality, the value of which varies with bearing type and design" was the introduction as a co-value to the basic dynamic load ratings.

ISO 281:2007 (second edition)

Since the publication of ISO 281:1990 additional knowledge regarding the influence on bearing life of contamination, lubrication, internal stresses from mounting, stresses from hardening, fatigue load limit of the material, has been gained. In ISO 281:1990/Amd.2:2000, a general method was presented to consider such influences in the calculation of a modified rating life of a bearing. The said Amendment was incorporated into the second edition, which also provides a practical method to consider the influence on bearing life of lubrication conditions, contaminated lubricant and fatigue load of bearing

material. The life modification factors for reliability, a_1, have been slightly adjusted and extended to 99,95 % reliability.

Rolling bearings — Explanatory notes on ISO 281 —

Part 1:
Basic dynamic load rating and basic rating life

1   Scope

This document specifies supplementary background information regarding the derivation of mathematical expressions and factors given in ISO 281:2007.

2   Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 281:2007, *Rolling bearings — Dynamic load ratings and rating life*

3   Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 281:2007 apply.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at http://www.electropedia.org/

4   Symbols

A constant of proportionality

A_1 constant of proportionality determined experimentally

B_1 constant of proportionality determined experimentally

C_1 basic dynamic radial load rating of a rotating ring

C_2 basic dynamic radial load rating of a stationary ring

C_a basic dynamic axial load rating for thrust ball or roller bearing

C_{a1} basic dynamic axial load rating of the rotating ring of an entire thrust ball or roller bearing

C_{a2} basic dynamic axial load rating of the stationary ring of an entire thrust ball or roller bearing

C_{ak} basic dynamic axial load rating as a row k of an entire thrust ball or roller bearing

C_{a1k} basic dynamic axial load rating as a row k of the rotating ring of thrust ball or roller bearing

C_{a2k} basic dynamic axial load rating as a row k of the stationary ring of thrust ball or roller bearing

C_e basic dynamic load rating for outer ring

C_i basic dynamic load rating for inner ring

C_r basic dynamic radial load rating for radial ball or roller bearing

D_{pw} pitch diameter of ball or roller set

D_w ball diameter

D_{we} mean roller diameter

E_o modified modulus of elasticity

F_a axial load

F_r radial load

J_1 factor relating mean equivalent load on a rotating ring to Q_{max}

J_2 factor relating mean equivalent load on a stationary ring to Q_{max}

J_a axial load integral

J_r radial load integral

L bearing life

L_{10} basic rating life

L_{we} effective contact length of roller

L_{wek} L_{we} per row k

N number of stress applications to a point on the raceway

P_a dynamic equivalent axial load for thrust bearing

P_r dynamic equivalent radial load for radial bearing

P_{r1} dynamic equivalent radial load for the rotating ring

P_{r2} dynamic equivalent radial load for the stationary ring

Q normal force between a rolling element and the raceways

Q_C rolling element load for the basic dynamic load rating of the bearing

Q_{C1} rolling element load for the basic dynamic load rating of a ring rotating relative to the applied load

Q_{C2} rolling element load for the basic dynamic load rating of a ring stationary relative to the applied load

Q_{max} maximum rolling element load

S probability of survival, reliability

V volume representative of the stress concentration

V_f rotation factor

X radial load factor for radial bearing

X_a radial load factor for thrust bearing

Y axial load factor for radial bearing

Y_a axial load factor for thrust bearing

Z number of balls or rollers per row

Z_k number of balls or rollers per row k

a semimajor axis of the projected contact ellipse

a_1 life adjustment factor for reliability

b semiminor axis of the projected contact ellipse

c exponent determined experimentally

c_c compression constant

e measure of life scatter, i.e. Weibull slope determined experimentally

e limiting value of F_a / F_r for the applicability of different values of factors X and Y in the new edition

f_c factor which depends on the geometry of the bearing components, the accuracy to which the various components are made, and the material

h exponent determined experimentally

i number of rows of balls or rollers

l circumference of the raceway

r cross-sectional raceway groove radius

r_e cross-sectional raceway groove radius of outer ring or housing washer

r_i cross-sectional raceway groove radius of inner ring or shaft washer

t auxiliary parameter

z_o depth of the maximum orthogonal subsurface shear stress

α nominal contact angle

α' actual contact angle

γ $D_w \cos \alpha / D_{pw}$ for ball bearings with $\alpha \neq 90°$

    D_w / D_{pw} for ball bearings with $\alpha = 90°$

    $D_{we} \cos \alpha / D_{pw}$ for roller bearings with $\alpha \neq 90°$

    D_{we} / D_{pw} for roller bearings with $\alpha = 90°$

ε parameter indicating the width of the loaded zone in the bearing

η reduction factor

λ reduction factor

    3

μ	factor introduced by Hertz
ν	factor introduced by Hertz, or adjustment factor for exponent variation
σ_{max}	maximum contact stress
$\Sigma\rho$	curvature sum
τ_o	maximum orthogonal subsurface shear stress
φ_o	one half of the loaded arc

5   General

The derivation of the basic dynamic load ratings is described in Formulae (1) to (46). The dynamic equivalent load and the radial and axial load factors are covered in Formulae (47) to (82), while basic rating life is described in Formulae (83) to (89).

6   Basic dynamic load rating

6.1   General

The background to basic dynamic load ratings of rolling bearings according to ISO 281 appears in References [5] and [6].

The expressions for calculation of basic dynamic load ratings of rolling bearings develop from a power formula that can be written as follows:

$$\ln\frac{1}{S} \propto \frac{\tau_o^c N^e V}{z_o^h} \tag{1}$$

where

S	is the probability of survival;
τ_o	is the maximum orthogonal subsurface shear stress;
N	is the number of stress applications to a point on the raceway;
V	is the volume representative of the stress concentration;
z_o	is the depth of the maximum orthogonal subsurface shear stress;
c, h	are experimentally determined exponents;
e	is the measure of life scatter, i.e. the Weibull slope determined experimentally.

For "point" contact conditions (ball bearings) it is assumed that the volume, V, representative of the stress concentration in Formula (1) is proportional to the major axis of the projected contact ellipse,

$2a$, the circumference of the raceway, l, and the depth, z_o, of the maximum orthogonal subsurface shear stress, τ_o:

$$V \propto 2a \, z_o \, l \tag{2}$$

Substituting Formula (2) into Formula (1):

$$\ln \frac{1}{S} \propto \frac{\tau_o^c N^e a l}{z_o^{h-1}} \tag{3}$$

"Line" contact was considered in References [5] and [6] to be approached under conditions where the major axis of the calculated Hertz contact ellipse is 1,5 times the effective roller contact length:

$$2a = 1,5 L_{we} \tag{4}$$

In addition, b/a should be small enough to permit the introduction of the limit value of ab^2 as b/a approaches 0:

$$ab^2 = \frac{2}{\pi} \frac{3Q}{E_o \sum \rho} \tag{5}$$

(for variable definitions, see 6.2).

6.2 Basic dynamic radial load rating, C_r, for radial ball bearings

From the theory of Hertz, the maximum orthogonal subsurface shear stress, τ_o, and the depth, z_o, can be expressed in terms of a radial load F_r, i.e. a maximum rolling element load, Q_{max}, or a maximum contact stress, σ_{max}, and dimensions for the contact area between a rolling element and the raceways. The relationships are:

$$\tau_o = T \sigma_{max}$$

$$z_o = \zeta b$$

$$T = \frac{(2t-1)^{1/2}}{2t(t+1)}$$

$$\zeta = \frac{1}{(t+1)(2t-1)^{1/2}}$$

$$a = \mu \left(\frac{3Q}{E_o \sum \rho} \right)^{1/3}$$

$$b = v \left(\frac{3Q}{E_o \sum \rho} \right)^{1/3}$$

where

σ_{max} is the maximum contact stress;

t is the auxiliary parameter;

a is the semimajor axis of the projected contact ellipse;

b is the semiminor axis of the projected contact ellipse;

Q is the normal force between a rolling element and the raceways;

E_o is the modified modulus of elasticity;

$\Sigma\rho$ is the curvature sum;

μ, v are factors introduced by Hertz.

Consequently, for a given rolling bearing, τ_0, a, l and z_0 can be expressed in terms of bearing geometry, load and revolutions. Formula (3) is changed to a formula by inserting a constant of proportionality. Inserting a specific number of revolutions (e.g. 10^6) and a specific reliability (e.g. 0,9), the formula is solved for a rolling element load for basic dynamic load rating which is designated to point contact rolling bearings introducing a constant of proportionality, A_1:

$$Q_C = \frac{1,3}{4^{(2c+h-2)/(c-h+2)}0,5^{3e/(c-h+2)}} A_1 \left(\frac{2r}{2r-D_w}\right)^{0,41} \frac{(1\mp\gamma)^{(1,59c+1,41h-5,82)/(c-h+2)}}{(1\pm\gamma)^{3e/(c-h+2)}} \times$$
$$\left(\frac{\gamma}{\cos\alpha}\right)^{3/(c-h+2)} D_w^{(2c+h-5)/(c-h+2)} Z^{-3e/(c-h+2)}$$
(6)

where

Q_C is the rolling element load for the basic dynamic load rating of the bearing;

D_w is the ball diameter;

γ is $D_w \cos\alpha/D_{pw}$;

in which

D_{pw} is the pitch diameter of the ball set;

α is the nominal contact angle;

Z is the number of balls per row.

The basic dynamic radial load rating, C_1, of a rotating ring is given by:

$$C_1 = Q_{C1}\ Z \cos\alpha\ \frac{J_r}{J_1} = 0,407\ Q_{C1}\ Z \cos\alpha$$
(7)

The basic dynamic radial load rating, C_2, of a stationary ring is given by:

$$C_2 = Q_{C2}\ Z \cos\alpha\ \frac{J_r}{J_2} = 0,389\ Q_{C2}\ Z \cos\alpha$$
(8)

where

Q_{C1} is the rolling element load for the basic dynamic load rating of a ring rotating relative to the applied load;

Q_{C2} is the rolling element load for the basic dynamic load rating of a ring stationary relative to the applied load;

$J_r = J_r\,(0,5)$ is the radial load integral for zero diametral clearance (see Table 3);

$J_1 = J_1\,(0,5)$ is the factor relating mean equivalent load on a rotating ring to $Q_{\max}$ for zero diametral clearance (see Table 3);

$J_2 = J_2\,(0,5)$ is the factor relating mean equivalent load on a stationary ring to $Q_{\max}$ for zero diametral clearance (see Table 3).

The relationship between C_r for an entire radial ball bearing, and C_1 and C_2, is expressed in terms of the product law of probability as:

$$C_r = C_1\left[1+\left(\frac{C_1}{C_2}\right)^{(c-h+2)/3}\right]^{-3/(c-h+2)} \tag{9}$$

Substituting Formulae (6), (7) and (8) into Formula (9), the basic dynamic radial load rating, C_r, for an entire ball bearing is expressed as:

$$C_r = 0,41\frac{1,3}{4^{(2c+h-2)/(c-h+2)}\ 0,5^{3e/(c-h+2)}}A_1\left(\frac{2r_i}{2r_i-D_w}\right)^{0,41}\frac{(1-\gamma)^{(1,59c+1,41h-5,82)/(c-h+2)}}{(1+\gamma)^{3e/(c-h+2)}}\gamma^{3/(c-h+2)}\times$$

$$\left[1+\left\{1,04\left[\frac{r_i}{r_e}\left(\frac{2r_e-D_w}{2r_i-D_w}\right)\right]^{0,41}\left(\frac{1-\gamma}{1+\gamma}\right)^{(1,59c+1,41h+3e-5,82)/(c-h+2)}\right\}^{(c-h+2)/3}\right]^{-3/(c-h+2)}\times$$

$$(i\,\cos\alpha)^{(c-h-1)/(c-h+2)}\ Z^{(c-h-3e+2)/(c-h+2)}D_w^{(2c+h-5)/(c-h+2)} \tag{10}$$

where

A_1 is the experimentally determined proportionality constant;

r_i is the cross-sectional raceway groove radius of the inner ring;

r_e is the cross-sectional raceway groove radius of the outer ring;

i is the number of rows of balls.

Here, the contact angle α, the number of rolling elements (balls), Z, and the ball diameter, D_w, depend on bearing design. On the other hand, the ratios of raceway groove radii, r_i and r_e, to a half-diameter of a rolling element (ball), $D_w/2$ and $\gamma = D_w\cos\alpha/D_{pw}$, are not dimensional, therefore it is convenient in practice that the value for the initial terms on the right-hand side of Formula (10) to be designated as a factor, f_c:

$$C_r = f_c(i\,\cos\alpha)^{(c-h-1)/(c-h+2)}Z^{(c-h-3e+2)/(c-h+2)}D_w^{(2c+h-5)/(c-h+2)} \tag{11}$$

With radial ball bearings, the faults in bearings resulting from manufacturing need to be taken into consideration, and a reduction factor, λ, is introduced to reduce the value for a basic dynamic radial load rating for radial ball bearings from its theoretical value. It is convenient to include λ in the factor, fc. The value of λ is determined experimentally.

$$f_c = 0,41\ \lambda\ \frac{1,3}{4^{(2c+h-2)/(c-h+2)}0,5^{3e/(c-h+2)}}\ A_1\left(\frac{2r_i}{2r_i-D_w}\right)^{0,41}\ \frac{(1-\gamma)^{(1,59c+1,41h-5,82)/(c-h+2)}}{(1+\gamma)^{3e/(c-h+2)}}\ \gamma^{3/(c-h+2)}\times$$

$$\left[1+\left\{1,04\left[\frac{r_i}{r_e}\left(\frac{2r_e-D_w}{2r_i-D_w}\right)\right]^{0,41}\left(\frac{1-\gamma}{1+\gamma}\right)^{(1,59c+1,41h+3e-5,82)/(c-h+2)}\right\}^{(c-h+2)/3}\right]^{-3/(c-h+2)} \tag{12}$$

Based on References [5] and, [6] the following values were assigned to the experimental constants in the load rating formulae for ball bearings:

$$e = \frac{10}{9}$$

$$c = \frac{31}{3}$$

$$h = \frac{7}{3}$$

Substituting the numerical values into Formula (11) gives the following, however, a sufficient number of test results are only available for small balls, i.e. up to a diameter of 25,4 mm (1 inch), and these show that the load rating may be taken as being proportional to $D_w^{1,8}$. In the case of larger balls, the load rating appears to increase even more slowly in relation to the ball diameter, and $D_w^{1,4}$ can be assumed where $D_w > 25,4$ mm:

$$C_r = f_c\ (i\cos\alpha)^{0,7}\ Z^{2/3}\ D_w^{1,8}$$
for $D_w \leq 25,4$ mm $\tag{13}$

$$C_r = 3,647\ f_c\ (i\cos\alpha)^{0,7}\ Z^{2/3}\ D_w^{1,4}$$
for $D_w > 25,4$ mm $\tag{14}$

$$f_c = 0,089\ A_1\ 0,41\ \lambda\ \left(\frac{2r_i}{2r_i-D_w}\right)^{0,41}\ \frac{\gamma^{0,3}(1-\gamma)^{1,39}}{(1+\gamma)^{1/3}}\times$$

$$\left[1+\left\{1,04\left[\frac{r_i}{r_e}\left(\frac{2r_e-D_w}{2r_i-D_w}\right)\right]^{0,41}\left(\frac{1-\gamma}{1+\gamma}\right)^{1,72}\right\}^{10/3}\right]^{-3/10} \tag{15}$$

Values of f_c in ISO 281:2007, Table 2, are calculated by substituting raceway groove radii and reduction factors given in Table 1 into Formula (15).

The value for $0,089A_1$ is 98,066 5 to calculate C_r in newtons.

6.3 Basic dynamic axial load rating, C_a, for single row thrust ball bearings

6.3.1 Thrust ball bearings with contact angle $\alpha \neq 90°$

As in 6.2, for thrust ball bearings with contact angle $\alpha \neq 90°$:

$$C_a = f_c (\cos \alpha)^{(c-h-1)/(c-h+2)} \tan \alpha \; Z^{(c-h-3e+2)/(c-h+2)} D_w^{(2c+h-5)/(c-h+2)} \tag{16}$$

For most thrust ball bearings, the theoretical value of a basic dynamic axial load rating has to be reduced on the basis of unequal distribution of load among the rolling elements in addition to the reduction factor, λ, which is introduced in to radial ball bearing load ratings. This reduction factor is designated as η.

Consequently, the factor f_c is given by:

$$f_c = \lambda \; \eta \; \frac{1{,}3}{4^{(2c+h-2)/(c-h+2)} \; 0{,}5^{3e/(c-h+2)}} A_1 \left(\frac{2r_i}{2r_i - D_w} \right)^{0{,}41} \frac{(1-\gamma)^{(1{,}59c \, + \, 1{,}41h \, + \, 5{,}82)/(c-h+2)}}{(1+\gamma)^{3e/(c-h+2)}} \gamma^{3/(c-h+2)} \times$$

$$\left[1 + \left\{ \left[\frac{r_i}{r_e} \left(\frac{2r_e - D_w}{2r_i - D_w} \right) \right]^{0{,}41} \left(\frac{1-\gamma}{1+\gamma} \right)^{(1{,}59c \, + \, 1{,}41h \, +3e- \, 5{,}82)/(c-h+2)} \right\}^{(c-h+2)/3} \right]^{-3/(c-h+2)} \tag{17}$$

Similarly, to take the effect of ball size into account, substitute experimental constants $e = \frac{10}{9}$, $c = \frac{31}{3}$, and $h = \frac{7}{3}$ into Formulae (16) and (17) to give:

$$C_a = f_c (\cos \alpha)^{0{,}7} \tan \alpha \; Z^{2/3} D_w^{1{,}8}$$
for $D_w \leq 25{,}4$ mm
$$\tag{18}$$

$$C_a = 3{,}647 \, f_c (\cos \alpha)^{0{,}7} \tan \alpha \; Z^{2/3} D_w^{1{,}4}$$
for $D_w > 25{,}4$ mm
$$\tag{19}$$

$$f_c = 0{,}089 \, A_1 \; \lambda \eta \left(\frac{2r_i}{2r_i - D_w} \right)^{0{,}41} \frac{\gamma^{0{,}3}(1-\gamma)^{1{,}39}}{(1+\gamma)^{1/3}} \times$$

$$\left[1 + \left\{ \left[\frac{r_i}{r_e} \left(\frac{2r_e - D_w}{2r_i - D_w} \right) \right]^{0{,}41} \left(\frac{1-\gamma}{1+\gamma} \right)^{1{,}72} \right\}^{10/3} \right]^{-3/10} \tag{20}$$

The value for $0{,}089 A_1$ is 98,066 5 to calculate C_a in newtons. Values of f_c in ISO 281:2007, Table 4, given in the column for $\alpha = 45°$, 60° and 70°, are calculated by substituting raceway groove radii and reduction factors given in Table 1 into Formula (20).

6.3.2 Thrust ball bearings with contact angle $\alpha = 90°$

As in 6.2, for thrust ball bearings with contact angle $\alpha = 90°$:

$$C_a = f_c Z^{(c-h-3e+2)/(c-h+2)} D_w^{(2c+h-5)/(c-h+2)} \tag{21}$$

$$f_c = \lambda \eta \frac{1.3}{4^{(2c+h-2)/(c-h+2)} 0.5^{3e/(c-h+2)}} A_1 \left(\frac{2r_i}{2r_i - D_w} \right)^{0,41} \gamma^{3/(c-h+2)} \times$$

$$\left[1 + \left\{ \left[\frac{r_i}{r_e} \left(\frac{2r_e - D_w}{2r_i - D_w} \right) \right]^{0,41} \right\}^{(c-h+2)/3} \right]^{-3/(c-h+2)} \tag{22}$$

in which $\gamma = D_w/D_{pw}$.

Similarly, to take the effect of ball size into account, substitute experimental constants $e = \frac{10}{9}$, $c = \frac{31}{3}$, and $h = \frac{7}{3}$ into Formulae (21) and (22), to give:

$$C_a = f_c\ Z^{2/3}\ D_w^{1,8}$$
for $D_w \leq 25,4$ mm
$$\tag{23}$$

$$C_a = 3{,}647\ f_c\ Z^{2/3}\ D_w^{1,4}$$
for $D_w > 25,4$ mm
$$\tag{24}$$

$$f_c = 0{,}089\ A_1\ \lambda\ \eta \left(\frac{2r_i}{2r_i - D_w} \right)^{0,41} \gamma^{0,3} \left[1 + \left\{ \left[\frac{r_i}{r_e} \left(\frac{2r_e - D_w}{2r_i - D_w} \right) \right]^{0,41} \right\}^{10/3} \right]^{-3/10} \tag{25}$$

The value for $0{,}089A_1$ is $98{,}066\ 5$ to calculate C_a in newtons. Values of f_c in ISO 281:2007, Table 4, second column from left, are calculated by substituting raceway groove radii and reduction factors which are given in Table 1 into Formula (25).

6.4 Basic dynamic axial load rating, C_a, for thrust ball bearings with two or more rows of balls

According to the product law of probability, relationships between the basic axial load rating of an entire thrust ball bearing and of both the rotating and stationary rings are given as:

$$C_{ak} = \left[C_{a1k}^{-(c-h+2)/3} + C_{a2k}^{-(c-h+2)/3} \right]^{-3/(c-h+2)} \tag{26}$$

in which

$$\left. \begin{array}{l} C_{a1k} = Q_{C1}\ \sin \alpha\ Z_k \\ C_{a2k} = Q_{C2}\ \sin \alpha\ Z_k \end{array} \right\}$$

$$C_a = \left[C_{a1}^{-(c-h+2)/3} + C_{a2}^{-(c-h+2)/3} \right]^{-3/(c-h+2)} \tag{27}$$

where

C_{ak} is the basic dynamic axial load rating as a row k of an entire thrust ball bearing;

C_{a1k} is the basic dynamic axial load rating as a row k of the rotating ring of an entire thrust ball bearing;

C_{a2k} is the basic dynamic axial load rating as a row k of the stationary ring of an entire thrust ball bearing;

C_a is the basic dynamic axial load rating of an entire thrust ball bearing;

C_{a1} is the basic dynamic axial load rating of the rotating ring of an entire thrust ball bearing;

C_{a2} is the basic dynamic axial load rating of the stationary ring of an entire thrust ball bearing.

Substituting Formula (26) into Formula (27), and assuming that the load on a single row is proportional to the number of balls in each row, yields:

$$C_a = \left[\sum_{k=1}^{n} \left(C_{ak} \frac{\sum_{j=1}^{n} Z_j}{Z_k} \right)^{-(c-h+2)/3} \right]^{-3/(c-h+2)}$$

$$= \sum_{j=1}^{n} Z_j \left[\sum_{k=1}^{n} \left(\frac{Z_k}{C_{ak}} \right)^{(c-h+2)/3} \right]^{-3/(c-h+2)} \tag{28}$$

where

Z_k (Z_j) is the number of balls per row k (j).

Substituting experimental constants $c = \dfrac{31}{3}$ and $h = \dfrac{7}{3}$ gives:

$$C_a = (Z_1 + Z_2 + Z_3 + \cdots + Z_n) \left[\left(\frac{Z_1}{C_{a1}} \right)^{10/3} + \left(\frac{Z_2}{C_{a2}} \right)^{10/3} + \left(\frac{Z_3}{C_{a3}} \right)^{10/3} + \cdots + \left(\frac{Z_n}{C_{an}} \right)^{10/3} \right]^{-3/10} \tag{29}$$

The load ratings C_{a1}, C_{a2}, C_{a3} … C_{an} for the rows with Z_1, Z_2, Z_3 … Z_n balls are calculated from the appropriate single row thrust ball bearing formula in 6.3.

6.5   Basic dynamic radial load rating, C_r, for radial roller bearings

By a procedure similar to that used to obtain Formula (10) for point contact in 6.2, but applying Formulae (4) and (5), the basic dynamic radial load rating of radial roller bearings (line contact) is obtained:

$$C_r = 0{,}377 \frac{1}{2^{(c+h-1)/(c-h+1)} \ 0{,}5^{2e/(c-h+1)}} \ B_1 \ \frac{(1-\gamma)^{(c+h-3)/(c-h+1)}}{(1+\gamma)^{2e/(c-h+1)}} \ \gamma^{2/(c-h+1)} \times$$

$$\left\{ 1 + \left[1{,}04 \left(\frac{1-\gamma}{1+\gamma} \right)^{(c+h+2e-3)/(c-h+1)} \right]^{(c-h+1)/2} \right\}^{-2/(c-h+1)} \left(i \, L_{we} \, \cos\alpha \right)^{(c-h-1)/(c-h+1)} \times$$

$$Z^{(c-h-2e+1)/(c-h+1)} \ D_{we}^{(c+h-3)/(c-h+1)} \tag{30}$$

where

B_1 is an experimentally determined proportionality constant;

D_{we} is the mean roller diameter;

γ is $D_{we} \cos\alpha / D_{pw}$;

in which

D_{pw} is the pitch diameter of roller set;

α is the nominal contact angle;

L_{we} is the effective contact length of roller;

i is the number of rows of rollers;

Z is the number of rollers per row.

Here, the contact angle, α, the number of rollers, Z, the mean roller diameter, D_{we}, and the effective contact length, L_{we}, depend on bearing design. On the other hand, $\gamma = D_{we} \cos \alpha / D_{pw}$ is not dimensional, therefore it is convenient in practice that the terms up to "$i L_{we}$..." on the right-hand side of Formula (30) to be designated as a factor, f_c.

Consequently,

$$C_r = f_c \ (i \ L_{we} \ \cos \alpha)^{(c-h-1)/(c-h+1)} \ Z^{(c-h-2e+1)/(c-h+1)} \ D_{we}^{(c+h-3)/(c-h+1)} \tag{31}$$

For the basic dynamic radial load rating for radial roller bearings, adjustments are made to take account of stress concentration (e.g. edge loading) and of the use of a constant instead of a varying life formula exponent (see Clause 8). Adjustment for stress concentration is a reduction factor, λ, and for exponent variation a factor, ν. It is convenient to include both factors — which are determined experimentally — in the factor, f_c, which is consequently given by:

$$f_c = 0{,}377 \ \lambda \ \nu \ \frac{1}{2^{(c+h-1)/(c-h+1)} \ 0{,}5^{\ 2e/(c-h+1)}} \ B_1 \ \frac{(1-\gamma)^{(c+h-3)/(c-h+1)}}{(1+\gamma)^{2e/(c-h+1)}} \ \gamma^{2/(c-h+1)} \times$$

$$\left\{ 1+\left[1{,}04 \ \left(\frac{1-\gamma}{1+\gamma}\right)^{(c+h+2e-3)/(c-h+1)} \right]^{(c-h+1)/2} \right\}^{-2/(c-h+1)} \tag{32}$$

The Weibull slope, e, and the constants, c and h, are determined experimentally. Based on References [5] and [6] and subsequent verification tests with spherical, cylindrical, and tapered roller bearings, the following values were assigned to the experimental constants in the rating formulae for roller bearings:

$e = \dfrac{9}{8}$

$c = \dfrac{31}{3}$

$h = \dfrac{7}{3}$

Substituting experimental constants $e = \dfrac{9}{8}$, $c = \dfrac{31}{3}$, and $h = \dfrac{7}{3}$ into Formulae (31) and (32),

$$C_r = f_c \ (i \ L_{we} \ \cos \alpha)^{7/9} \ Z^{3/4} D_{we}^{29/27} \tag{33}$$

$$f_c = 0{,}483 B_1 \ 0{,}377 \ \lambda \ \nu \ \frac{\gamma^{2/9}(1-\gamma)^{29/27}}{(1+\gamma)^{1/4}} \left\{ 1+\left[1{,}04\left(\frac{1-\gamma}{1+\gamma}\right)^{143/108} \right]^{9/2} \right\}^{-2/9} \tag{34}$$

The value for $0{,}483B_1$ is 551,133 73 to calculate C_r in newtons. Values of f_c in ISO 281:2007, Table 7, are calculated by substituting the reduction factor given in Table 2 into Formula (34).

6.6 Basic dynamic axial load rating, C_a, for single row thrust roller bearings

6.6.1 Thrust roller bearings with contact $\alpha \neq 90°$

Extension of 6.2 gives:

$$C_a = f_c (L_{we} \cos \alpha)^{(c-h-1)/(c-h+1)} \tan \alpha \; Z^{(c-h-2e+1)/(c-h+1)} D_{we}^{(c+h-3)/(c-h+1)} \tag{35}$$

For thrust roller bearings, the theoretical value of a basic dynamic axial load rating has to be reduced on the basis of unequal distribution of load among the rolling elements in addition to the reduction factor, λ, which is introduced in radial roller bearing load ratings. This reduction factor is designated as η.

Consequently, the factor f_c is given by:

$$f_c = \lambda \; v \; \eta \; \frac{1}{2^{(c+h-1)/(c-h+1)} 0{,}5^{2e/(c-h+1)}} \; B_1 \; \frac{(1-\gamma)^{(c+h-3)/(c-h+1)}}{(1+\gamma)^{2e/(c-h+1)}} \; \gamma^{2/(c-h+1)} \times$$
$$\left\{ 1 + \left[\left(\frac{1-\gamma}{1+\gamma} \right)^{(c+h+2e-3)/(c-h+1)} \right]^{(c-h+1)/2} \right\}^{-2/(c-h+1)} \tag{36}$$

Substituting experimental constants $e = \frac{9}{8}$, $c = \frac{31}{3}$, and $h = \frac{7}{3}$,

$$C_a = f_c (L_{we} \cos \alpha)^{7/9} \tan \alpha \; Z^{3/4} D_{we}^{29/27} \tag{37}$$

$$f_c = 0{,}483 B_1 \; \lambda \; v \; \eta \; \frac{\gamma^{2/9} (1-\gamma)^{29/27}}{(1+\gamma)^{1/4}} \left\{ 1 + \left[\left(\frac{1-\gamma}{1+\gamma} \right)^{143/108} \right]^{9/2} \right\}^{-2/9} \tag{38}$$

The value of $0{,}483 B_1$ is 551,133 73 to calculate C_a in newtons. Values for f_c in ISO 281:2007, Table 10, given in the columns for α = 50°, 65° and 80°, are calculated by substituting reduction factors given in Table 2 into Formula (38).

6.6.2 Thrust roller bearings with contact angle α = 90°

Extension of 6.2 gives:

$$C_a = f_c L_{we}^{(c-h-1)/(c-h+1)} Z^{(c-h-2e+1)/(c-h+1)} D_{we}^{(c+h-3)/(c-h+1)} \tag{39}$$

$$f_c = \lambda \; v \; \eta \; \frac{1}{2^{(c+h-1)/(c-h+1)} 0{,}5^{2e/(c-h+1)}} \; B_1 \gamma^{2/(c-h+1)} 2^{-2/(c-h+1)} \tag{40}$$

Substituting experimental constants $e = \frac{9}{8}$, $c = \frac{31}{3}$, and $h = \frac{7}{3}$,

$$C_a = f_c \; L_{we}^{7/9} \; Z^{3/4} \; D_{we}^{29/27} \tag{41}$$

$$f_c = 0{,}41 B_1 \; \lambda \; v \; \eta \; \gamma^{2/9} \tag{42}$$

The value for $0{,}41 B_1$ is 472,453 88 to calculate C_a in newtons. Values of f_c in ISO 281:2007, Table 10, second column from left, are calculated by substituting reduction factors given in Table 2 into Formula (42).

6.7 Basic dynamic axial load rating, C_a, for thrust roller bearings with two or more rows of rollers

According to the product law of probability, relationships between the basic dynamic axial load rating of an entire thrust roller bearing and of both the rotating and stationary rings are given as follows:

$$C_{ak} = \left[C_{a1k}^{-(c-h+1)/2} + C_{a2k}^{-(c-h+1)/2} \right]^{-2/(c-h+1)} \tag{43}$$

in which

$$\left. \begin{aligned} C_{a1k} &= Q_{C1} \, \sin \alpha \, Z_k L_{wek} \\ C_{a2k} &= Q_{C2} \, \sin \alpha \, Z_k L_{wek} \end{aligned} \right\}$$

$$C_a = \left[C_{a1}^{-(c-h+1)/2} + C_{a2}^{-(c-h+1)/2} \right]^{-2/(c-h+1)} \tag{44}$$

where

C_{ak} is the basic dynamic axial load rating as a row k of an entire thrust roller bearing;

C_{a1k} is the basic dynamic axial load rating as a row k of the rotating ring of an entire thrust roller bearing;

C_{a2k} is the basic dynamic axial load rating as a row k of the stationary ring of an entire thrust roller bearing;

C_a is the basic dynamic axial load rating of an entire thrust roller bearing;

C_{a1} is the basic dynamic axial load rating of the rotating ring of an entire thrust roller bearing;

C_{a2} is the basic dynamic axial load rating of the stationary ring of an entire thrust roller bearing;

Z_k is the number of rollers per row k.

Substituting Formula (43) into Formula (44), and assuming that the load on a single row is proportional to the product of roller length and number of rollers in each row, gives:

$$C_a = \left[\sum_{k=1}^{n} \left(C_{ak} \frac{\sum_{j=1}^{n} \left(Z_j L_{wej} \right)}{Z_k L_{wek}} \right)^{-(c-h+1)/2} \right]^{-2/(c-h+1)}$$

$$= \sum_{j=1}^{n} \left(Z_j L_{wej} \right) \left[\sum_{k=1}^{n} \left(\frac{Z_k L_{wek}}{C_{ak}} \right)^{(c-h+1)/2} \right]^{-2/(c-h+1)} \tag{45}$$

where

$Z_k (Z_j)$ is the number of rollers per row k (j);

$L_{wek} (L_{wej})$ is the effective contact length of roller in row k (j).

Substituting experimental constants $c=\dfrac{31}{3}$ and $h=\dfrac{7}{3}$,

$$C_a = \left(Z_1 L_{we1} + Z_2 L_{we2} + Z_3 L_{we3} + \cdots + Z_n L_{wen}\right) \times \left[\left(\frac{Z_1 L_{we1}}{C_{a1}}\right)^{9/2} + \left(\frac{Z_2 L_{we2}}{C_{a2}}\right)^{9/2} + \left(\frac{Z_3 L_{we3}}{C_{a3}}\right)^{9/2} + \cdots + \left(\frac{Z_n L_{wen}}{C_{an}}\right)^{9/2}\right]^{-2/9}$$

(46)

The load ratings, C_{a1}, C_{a2}, C_{a3} … C_{an} for the rows with Z_1, Z_2, Z_3 … Z_n rollers of lengths L_{we1}, L_{we2}, L_{we3} … L_{wen}, are calculated from the appropriate single row thrust roller bearing formula in 6.6.

Table 1 — Raceway groove radius and reduction factor for ball bearings

Table No. in ISO 281:2007	Bearing type	Raceway groove radius		Reduction factor	
		r_i	r_e	λ	η
2	Single row radial contact groove ball bearings Single and double row angular contact groove ball bearings	0,52 D_w		0,95	—
	Double row radial contact groove ball bearings	0,52 D_w		0,90	—
	Single and double row self-aligning ball bearings	0,53 D_w	$0,5\left(\dfrac{1}{\gamma}+1\right)D_w$	1	—
	Single row radial contact separable ball bearings (magneto bearings)	0,52 D_w	∞	0,95	—
4	Thrust ball bearings	0,535 D_w		0,90	$1-\dfrac{\sin\alpha}{3}$
NOTE Values of f_c in ISO 281:2007, Tables 2 and 4, are calculated by substituting raceway groove radii and reduction factors in this table into Formulae (15), (20) and (25), respectively.					

Table 2 — Reduction factor for roller bearings

Table No. in ISO 281:2007	Bearing type	Reduction factor	
		λv	η
7	Radial roller bearings	0,83	—
10	Thrust roller bearings	0,73	$1 - 0,15 \sin \alpha$
NOTE Values of f_c in ISO 281:2007, Tables 7 and 10, are calculated by substituting reduction factors in this table into Formulae (34), (38) and (42), respectively.			

7 Dynamic equivalent load

7.1 Expressions for dynamic equivalent load

7.1.1 Theoretical dynamic equivalent radial load, P_r, for single row radial bearings

If the indices 1 and 2 are assigned to the ring which rotates relative to the direction of load and the stationary ring respectively, then the mean values of the rolling element loads which are decisive for a single row radial bearing ring's life are given by:

$$
\left.
\begin{aligned}
Q_{C1} &= Q_{\max} J_1 = \frac{F_r}{Z \cos \alpha} \frac{J_1}{J_r} = \frac{F_a}{Z \sin \alpha} \frac{J_1}{J_a} \\
Q_{C2} &= Q_{\max} J_2 = \frac{F_r}{Z \cos \alpha} \frac{J_2}{J_r} = \frac{F_a}{Z \sin \alpha} \frac{J_2}{J_a}
\end{aligned}
\right\}
\tag{47}
$$

where

$Q_{\max}$ is the maximum rolling element load;

J_1 is the factor relating Q_{C1} to $Q_{\max}$;

J_2 is the factor relating Q_{C2} to $Q_{\max}$;

F_r is the radial load;

F_a is the axial load;

J_r is the radial load integral;

J_a is the axial load integral;

Z is the number of rolling elements;

α is the nominal contact angle.

Radial and axial load integrals are given by:

$$
\left.
\begin{aligned}
J_r &= J_r(\varepsilon) = \frac{1}{2\pi} \int_{-\phi_o}^{+\phi_o} \left[1 - \frac{1}{2\varepsilon}(1 - \cos \phi) \right]^t \cos \phi \, \mathrm{d}\phi \\
J_a &= J_a(\varepsilon) = \frac{1}{2\pi} \int_{-\phi_o}^{+\phi_o} \left[1 - \frac{1}{2\varepsilon}(1 - \cos \phi) \right]^t \mathrm{d}\phi
\end{aligned}
\right\}
\tag{48}
$$

where

t is 3/2 for point contact;

t is 1,1 for line contact;

ϕ_o is one half of the loaded arc;

ε is a parameter indicating the width of the loaded zone in the bearing.

Introducing the notation

$$J(t;s) = \left\{ \frac{1}{2\pi} \int\limits_{-\phi_o}^{+\phi_o} \left[1 - \frac{1}{2\varepsilon}(1 - \cos\phi) \right]^t d\phi \right\}^{1/s} \tag{49}$$

$$\left. \begin{array}{l} J_1 = J_1(\varepsilon) = J\left(\frac{9}{2};3\right); J_2 = J_2(\varepsilon) = J\left(5;\frac{10}{3}\right) \\[2mm] J_1 = J_1(\varepsilon) = J\left(\frac{9}{2};4\right); J_2 = J_2(\varepsilon) = J\left(5;\frac{9}{2}\right) \end{array} \right\} \tag{50}$$

for point and line contact respectively.

If P_{r1} and P_{r2} are the dynamic equivalent radial loads for the respective rings, then with radial displacement of the rings ($\varepsilon = 0{,}5$)

$$Q_{C1} = \frac{P_{r1}}{Z\cos\alpha} \frac{J_1(0{,}5)}{J_r(0{,}5)}; \quad Q_{C2} = \frac{P_{r2}}{Z\cos\alpha} \frac{J_2(0{,}5)}{J_r(0{,}5)} \tag{51}$$

where the values of $J_1(0{,}5)$, $J_2(0{,}5)$ and $J_r(0{,}5)$ are given in Table 3.

From Formulae (47), (51) and

$$\left(\frac{P_r}{C_r}\right)^w = \left(\frac{P_{r1}}{C_1}\right)^w + \left(\frac{P_{r2}}{C_2}\right)^w$$

is obtained

$$\left. \begin{array}{l} \dfrac{P_r}{F_r} = \left[\left(\dfrac{C_r}{C_1} \dfrac{J_{r(0{,}5)}}{J_1(0{,}5)} \dfrac{J_1}{J_r} \right)^w + \left(\dfrac{C_r}{C_2} \dfrac{J_{r(0{,}5)}}{J_2(0{,}5)} \dfrac{J_2}{J_r} \right)^w \right]^{1/w} \\[4mm] \dfrac{P_r}{F_a \cot\alpha} = \left[\left(\dfrac{C_r}{C_1} \dfrac{J_1}{J_1(0{,}5)} \right)^w + \left(\dfrac{C_r}{C_2} \dfrac{J_2}{J_2(0{,}5)} \right)^w \right]^{1/w} \dfrac{J_{r(0{,}5)}}{J_a} \end{array} \right\} \tag{52}$$

where

C_r is the basic dynamic radial load rating;

C_1 is the basic dynamic radial load rating of a rotating ring;

C_2 is the basic dynamic radial load rating of a stationary ring;

w is equal to pe, where p is the exponent on life formula and e is the Weibull slope.

Table 3 — Values of $J_r(0{,}5)$, $J_a(0{,}5)$, $J_1(0{,}5)$, $J_2(0{,}5)$ and w

Quantity	Point contact		Line contact		Point and line contact	
	Single row bearing	Double row bearing	Single row bearing	Double row bearing	Single row bearing	Double row bearing
$J_r(0{,}5)$	0,228 8	0,457 7	0,245 3	0,490 6	0,236 9	0,473 9
$J_a(0{,}5)$	0,278 2	0	0,309 0	0	0,293 2	0
$J_1(0{,}5)$	0,562 5	0,692 5	0,649 5	0,757 7	0,604 4	0,724 4
$J_2(0{,}5)$	0,587 5	0,723 3	0,674 4	0,786 7	0,629 5	0,754 3

Table 3 *(continued)*

Quantity	Point contact		Line contact		Point and line contact	
	Single	Double	Single	Double	Single	Double
	row bearing		row bearing		row bearing	
$J_r(0,5)/J_a(0,5)$	0,822	—	0,794	—	0,808	—
$J_r(0,5)/J_1(0,5)$	0,407	0,661	0,378	0,648	0,392	0,654
$J_r(0,5)/J_2(0,5)$	0,389	0,633	0,364	0,623	0,376	0,628
$J_2(0,5)/J_1(0,5)$	1,044		1,038		1,041	
$\dfrac{J_r(0,5)}{\sqrt{J_1(0,5)J_2(0,5)}}$	0,398 ($\approx 0,40$)	0,647 ($\approx 0,65$)	0,371	0,635	0,384	0,641
w	$\dfrac{10}{3}$		$\dfrac{9}{2}$		$\dfrac{180}{47}$	
$2^{1-(1/w)}$	1,625		1,714		1,669	

For radial displacement of the bearing rings ($\varepsilon = 0,5$) and fixed outer ring load ($C_1 = C_i$, basic dynamic load rating for inner ring; $C_2 = C_e$, basic dynamic load rating for outer ring) from [Formula (52)](#) is found

$$
\left.
\begin{aligned}
P_r = F_r = \frac{J_r(0,5)}{J_a(0,5)} F_a \cot\alpha &= 0,822\ F_a\ \cot\alpha \ \text{(point contact)} \\
P_r = F_r = \frac{J_r(0,5)}{J_a(0,5)} F_a \cot\alpha &= 0,794\ F_a\ \cot\alpha \ \text{(line contact)}
\end{aligned}
\right\}
\tag{53}
$$

for point and line contact respectively.

For $\varepsilon = 0,5$ and fixed inner ring load ($C_1 = V_f C_e$; $C_2 = C_i/V_f$), is found

$$
P_r = V_f F_r
\tag{54}
$$

where V_f is the rotation factor.

The factor V_f varies between $1 \pm 0,044$ and $1 \pm 0,038$ for point and line contact respectively. In ISO 281:2007, the rotation factor V_f has been deleted.

NOTE   The value of 1,2 for the rotation factor V_f was given in ISO/R281 for radial bearings, except self-aligning ball bearings, as safety factor.

For axial displacement of the bearing rings ($\varepsilon = \infty$) and fixed outer ring load ($C_1 = C_i$, $C_2 = C_e$),

$$
\left.
\begin{aligned}
P_r &= Y F_a \\
Y &= f_1 \frac{C_i}{C_e} \frac{J_r(0,5)}{J_1(0,5)} \cot\alpha
\end{aligned}
\right\}
\tag{55}
$$

The factor $f_1(C_i/C_e)$ varies between 1 and $1/V_f = J_1(0,5)/J_2(0,5)$. Introducing as a good approximation the geometric mean value $1/\sqrt{V_f}$ between these two values (see [Table 3](#)),

$$
Y = \frac{J_r(0,5)}{\sqrt{J_1(0,5)J_2(0,5)}} \cot\alpha
\tag{56}
$$

For non-self-aligning bearings, consideration has to be given to the effect of the manufacturing precision on the factor, Y.

The value of Y given in Formula (56) is corrected by the reduction factor η.

$$Y_1 = \frac{Y}{\eta} \tag{57}$$

For combined loads, Formula (52) gives related values of F_r/P_r and $F_a \cot \alpha/P_r$ corresponding to the curves given in Figure 1 for the limiting cases $C_1/C_2 \approx 0$ and $C_2/C_1 \approx 0$.

The points A represent $\varepsilon = 0,5$, i.e. radial displacement of the bearing rings. For these points,

$$\left.\begin{array}{l} F_a = 1{,}22 F_r \tan \alpha \;(\text{point contact}) \\ F_a = 1{,}26 F_r \tan \alpha \;(\text{line contact}) \end{array}\right\} \tag{58}$$

for point and line contact, respectively.
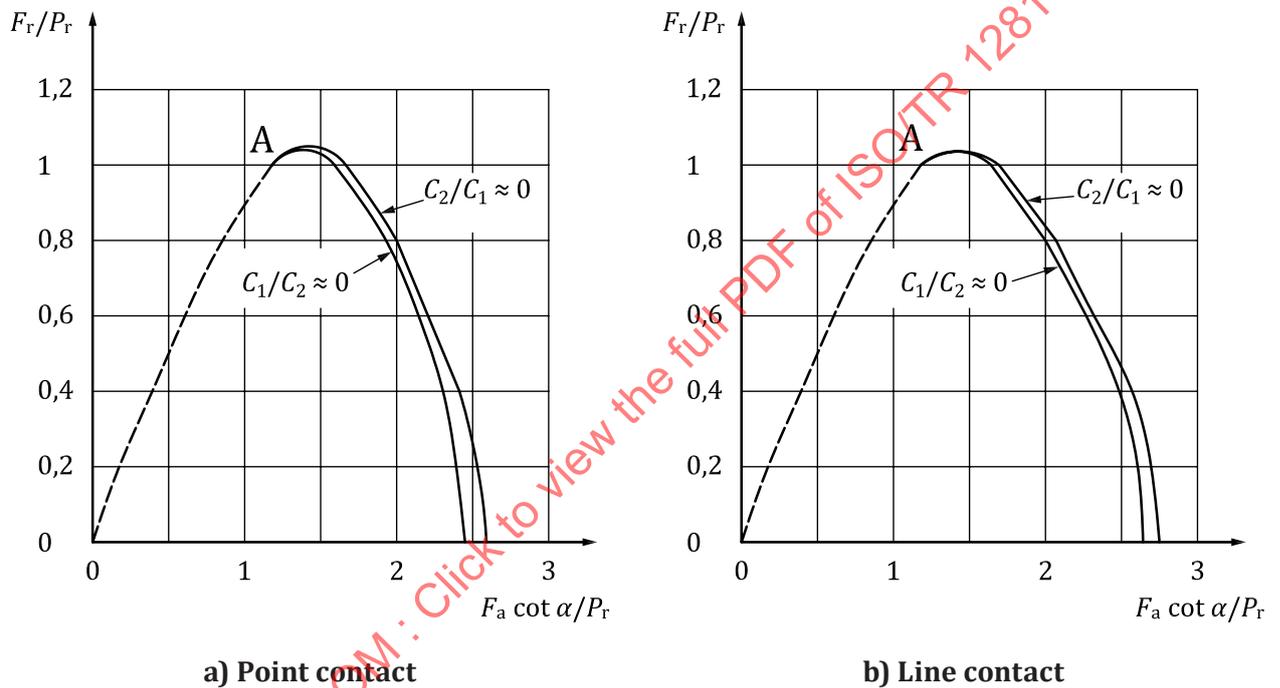
a) Point contact   **b) Line contact**

Key

A   points A
C_1 basic dynamic radial load rating of a rotating ring
C_2 basic dynamic radial load rating of a stationary ring
F_a axial load
F_r radial load
P_r dynamic equivalent radial load for radial bearing
α nominal contact angle

Figure 1 — Dynamic equivalent radial load, P_r, for single row radial bearings with constant contact angle, α

7.1.2   Theoretical dynamic equivalent radial load, P_r, for double row radial bearings

For double row radial bearings, the indices I and II are assigned to the respective rows. The determining factors for life of the rotating and stationary rings are the mean values:

$$\left.\begin{aligned} Q_{C1} &= Q_{\text{maxI}} J_1 \\ Q_{C2} &= Q_{\text{maxII}} J_2 \end{aligned}\right\} \tag{59}$$

where

$$\left.\begin{aligned} J_1 &= \left[J_1(\varepsilon_I)^w + \left(\frac{Q_{\text{maxII}}}{Q_{\text{maxI}}}\right)^w J_1(\varepsilon_{II})^w \right]^{1/w} \\ J_2 &= \left[J_2(\varepsilon_I)^w + \left(\frac{Q_{\text{maxII}}}{Q_{\text{maxI}}}\right)^w J_2(\varepsilon_{II})^w \right]^{1/w} \end{aligned}\right\} \tag{60}$$

For a bearing without internal clearances,

$$\left.\begin{aligned} \varepsilon_I + \varepsilon_{II} &= 1 \text{ for } \varepsilon_I \le 1 \\ \varepsilon_{II} &= 0 \text{ for } \varepsilon_I > 1 \end{aligned}\right\} \tag{61}$$

If the values of J_r, J_a, J_1 and J_2 for double row bearings are introduced, then the equivalent bearing load is obtained from Formula (52), as for single row bearings. $J_r(0,5), J_a(0,5), J_1(0,5)$ are here the values valid for $\varepsilon_I = \varepsilon_{II} = 0,5$ (see Table 3).

The bent curves given in Figure 2 are found for the limiting cases $C_1/C_2 \approx 0$ and $C_2/C_1 \approx 0$.

Both rows are loaded if $\varepsilon_I < 1$, i.e. if

$$\left.\begin{aligned} F_a &< 1,67 F_r \tan \alpha \text{ (point contact)} \\ F_a &< 1,91 F_r \tan \alpha \text{ (line contact)} \end{aligned}\right\} \tag{62}$$

for point and line contact, respectively.

Only one row is loaded if F_a is greater than that value. In that case, the life for double row bearings can be calculated from the theory of single row bearings as well as from the theory of double row bearings.

If P_{rI} is the equivalent radial load for the loaded row considered as a single row bearing and P_r is the equivalent load for the double row bearing,

$$\frac{P_r}{P_{rI}} = \frac{C_r}{C_1} = 2^{1-(1/w)} \tag{63}$$

Figure 1 and Figure 2 are calculated on the assumption of a constant contact angle. Figure 1 a) and Figure 2 a) are also approximately applicable to angular contact groove ball bearings, if cot α' is determined from Formula (64):
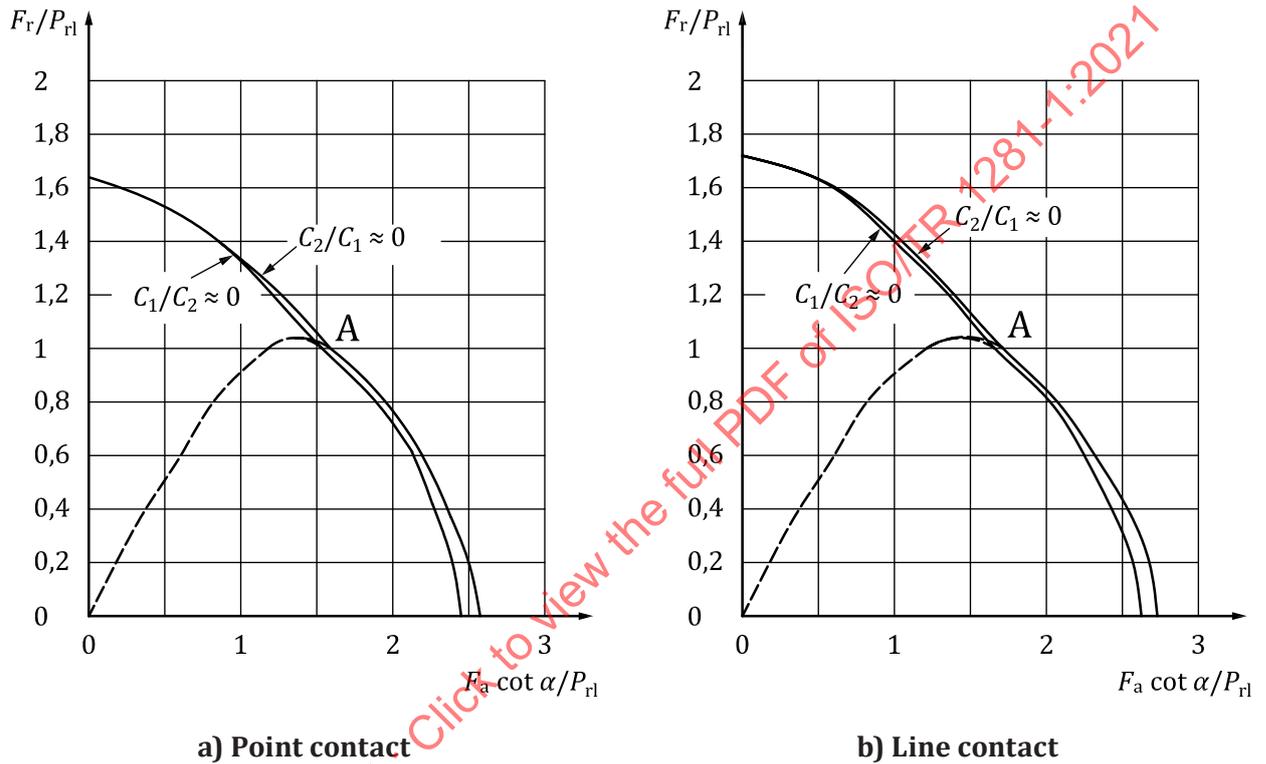
$$\left(\frac{\cos \alpha}{\cos \alpha'} - 1\right)^{3/2} \sin \alpha' = \left[\frac{c_c}{(2r/D_w) - 1}\right]^{3/2} \frac{F_a}{Z D_w^2} \tag{64}$$

where

c_c is the compression constant, which depends on the modulus of elasticity and the conformity $2r/D_w$;

r is the cross-sectional raceway groove radius;

D_w is the ball diameter.

a) Point contact   b) Line contact

Key

A   points A
C_1 basic dynamic radial load rating of a rotating ring
C_2 basic dynamic radial load rating of a stationary ring
F_a axial load
F_r radial load
P_{rl} dynamic equivalent radial load for the loaded row considered as a single row bearing
α nominal contact angle

Figure 2 — Dynamic equivalent radial load, P_{rl}, for double row radial bearings with constant contact angle, α

7.1.3 Theoretical dynamic equivalent radial load, P_r, for radial contact groove ball bearings

Figure 3 is applicable to radial contact groove ball bearings. The curve AC has been determined from Formula (52) and the approximate formula

$$\tan \alpha' \approx \left[\frac{2c}{(2r/D_w)-1} \right]^{3/8} \left(1 - \frac{1}{2\varepsilon} \right)^{3/8} \left(\frac{F_a}{J_a \ i \ Z \ D_w^2} \right)^{1/4} \tag{65}$$

and gives the functional relationship between F_r/P_r and $F_a \cot \alpha'/P_r$ where α' is the contact angle calculated from Formula (66) (see Reference [5])

$$\tan \alpha' \approx \left[\frac{2c}{(2r/D_w)-1} \right]^{3/8} \left(\frac{F_a}{i \ Z \ D_w^2} \right)^{1/4} \tag{66}$$

Formula (66) is obtained from Formula (65) for a centric axial load F_a, where $F_r = 0$, i.e. $\varepsilon = \infty$ and $J_a = 1$.
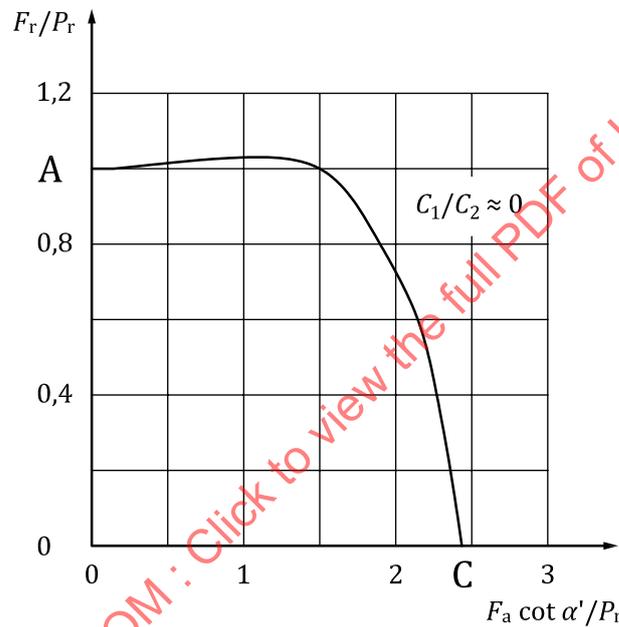

Key

A	point A
C	point C
C_1	basic dynamic radial load rating of a rotating ring
C_2	basic dynamic radial load rating of a stationary ring
F_a	axial load
F_r	radial load
P_r	dynamic equivalent radial load for radial bearing
α'	contact angle calculated from Formula (66)

Figure 3 — Dynamic equivalent radial load, P_r, for radial contact groove ball bearings

7.1.4 Practical expressions for dynamic equivalent radial load, P_r, for radial bearings with constant contact angle

From a practical standpoint, it is preferable to replace the theoretical curves in Figure 1 and Figure 2 by broken lines A_1BC for single row bearings and ABC for double row bearings, as in Figure 4.
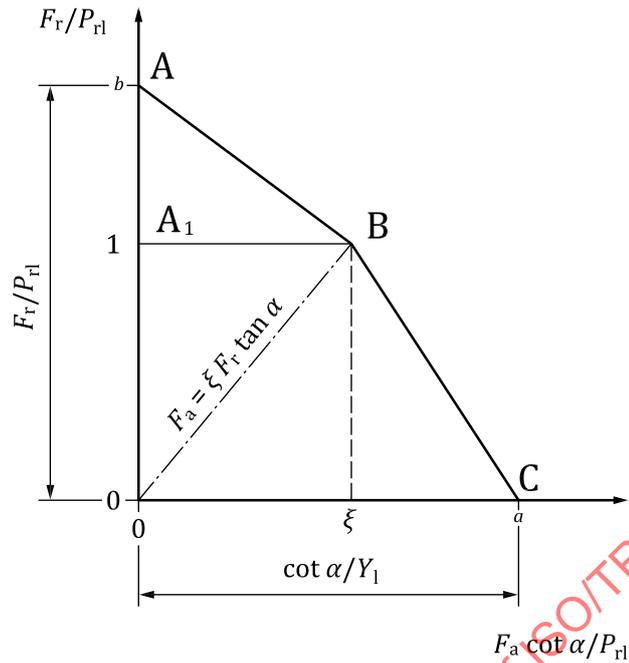
Key

A, A_1, B, C	points
A	intercept of line BC on abscissa (x-coordinate of point C)
B	intercept of line AB on ordinate (y-coordinate of point A)
F_a	axial load
F_r	radial load
P_{rI}	dynamic equivalent radial load for radial bearing, row I
Y_I	axial load factor for radial bearing, row I
A	nominal contact angle
Ξ	value of $F_a \cot \alpha / P_{rI}$ at the x-coordinate of point B

Figure 4 — Dynamic equivalent radial load, P_{rI}, for radial bearings with constant contact angle, α

The formula for the straight line A_1B in Figure 4 is

$$\frac{F_r}{P_{rI}} = 1$$

Therefore, for $F_a/F_r \le \xi \tan \alpha$:

$$P_{rI} = F_r \tag{67}$$

and the straight line passing through the points B $(\xi, 1)$ and C $(a, 0)$ is given by

$$\frac{(F_r/P_{rI}) - 1}{(F_a \cot \alpha / P_{rI}) - \xi} = \frac{-1}{a - \xi}$$

From this formula, for $F_a/F_r > \xi \tan \alpha$, it follows that

$$\left.
\begin{array}{l}
P_{rI} = \left(1 - \dfrac{\xi}{a}\right) F_r + \dfrac{1}{a} \cot \alpha \ F_a \equiv X_1 \ F_r + Y_1 \ F_a \\[2ex]
\text{where} \\[1ex]
X_1 = 1 - \dfrac{\xi}{a} = 1 - \xi \ Y_1 \ \tan \alpha \\[2ex]
\text{Therefore, from Formula (57)} \\[1ex]
X_1 = 1 - \dfrac{J_r(0{,}5)}{\sqrt{J_1(0{,}5)J_2(0{,}5)}} \dfrac{\xi}{\eta} \\[2ex]
Y_1 = \dfrac{J_r(0{,}5) \cot \alpha}{\sqrt{J_1(0{,}5)J_2(0{,}5)}} \dfrac{1}{\eta}
\end{array}
\right\} \tag{68}$$

For the double row bearings, the formula for the straight line AB is

$$\frac{(F_r/P_{rI}) - b}{F_a \cot \alpha / P_{rI}} = \frac{1 - b}{\xi}$$

From this,

$$P_{rI} = \frac{F_r}{b} + \left(\frac{b-1}{b}\right) \frac{F_a \cot \alpha}{\xi}$$

Therefore, for $F_a/F_r \le \xi \tan \alpha$, it follows that

$$\left.
\begin{array}{l}
P_r = 2^{1-(1/w)} P_{rI} = F_r + \left[2^{1-(1/w)} - 1\right] \dfrac{\cot \alpha}{\xi} F_a \equiv X_3 \ F_r + Y_3 \ F_a \\[2ex]
\text{where} \\[1ex]
X_3 = 1; \ Y_3 = \left[2^{1-(1/w)} - 1\right] \dfrac{1}{\xi} \cot a
\end{array}
\right\} \tag{69}$$

Further, from Formula (68), which represents straight line BC, for $F_a/F_r > \xi \tan \alpha$:

$$\left.
\begin{array}{l}
P_r = 2^{1-(1/w)} P_{rI} = 2^{1-(1/w)} X_1 F_r + 2^{1-(1/w)} Y_1 \ F_a \equiv X_2 \ F_r + Y_2 \ F_a \\[2ex]
\text{where} \\[1ex]
X_2 = 2^{1-(1/w)} X_1; \ Y_2 = 2^{1-(1/w)} Y_1
\end{array}
\right\} \tag{70}$$

Integrating the above, Table 4 shows expressions of dynamic equivalent radial load, P_r, and of factors X and Y, for radial bearings with constant contact angle, α.

Table 4 — Expressions for dynamic equivalent radial load, P_r, and factors X and Y for radial bearings with constant contact angle, α

|		Single row bearings	Double row bearings
Expressions	$\dfrac{F_a}{F_r} \leq e$	$P_r = F_r$	$P_r = X_3\,F_r + Y_3\,F_a$
	$\dfrac{F_a}{F_r} > e$	$P_r = X_1\,F_r + Y_1\,F_a$	$P_r = X_2\,F_r + Y_2\,F_a$
Radial load factor, X Axial load factor, Y		$X_1 = 1 - \dfrac{J_r(0,5)}{\sqrt{J_1(0,5)J_2(0,5)}}\dfrac{\xi}{\eta}$ $Y_1 = \dfrac{J_r(0,5)\cot\alpha}{\sqrt{J_1(0,5)J_2(0,5)}}\dfrac{1}{\eta}$	$\dfrac{X_2}{X_1} = \dfrac{Y_2}{Y_1} = 2^{1-(1/w)}$ $X_3 = 1$ $Y_3 = \dfrac{1}{\xi}\left[2^{1-(1/w)} - 1\right]\cot\alpha$
The symbol e is the limiting value of F_a/F_r for the applicability of different values of factors X and Y.		$e = \xi\,\tan\alpha$	

7.1.5 Practical expressions for dynamic equivalent radial load, P_r, for radial ball bearings

Generally, the contact angle of radial ball bearings varies with the load, but Table 4 can be approximately applicable to angular contact groove ball bearings, if α is replaced by contact angle α' under the axial load F_a given by Formula (64).

Therefore, according to Table 3,

$$\left.\begin{aligned}
&X_1 = 1 - 0,4\frac{\xi}{\eta}\,;\; Y_1 = \frac{0,4}{\eta}\cot\alpha' \\
&X_2 = 1,625\,X_1\,;\; Y_2 = 1,625\,Y_1 \\
&X_3 = 1\,;\; Y_3 = \frac{0,625}{\xi}\cot\alpha'
\end{aligned}\right\} \tag{71}$$

For single row and double row radial contact groove ball bearings, the theoretical curve in Figure 3 is replaced by the broken line A_1BC in Figure 5.
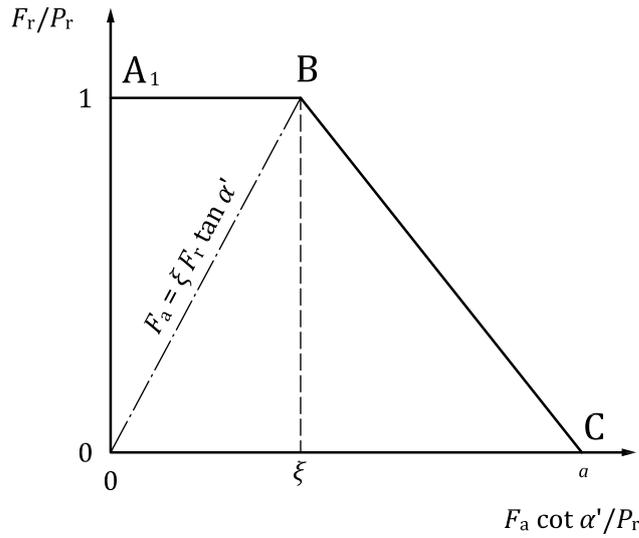
Key

A$_1$, B, C   points

a intercept of line BC on abscissa (x-coordinate of point C)

F_a axial load

F_r radial load

P_r dynamic equivalent radial load for radial bearing

α' contact angle calculated from Formula (66)

ξ value of $F_a \cot \alpha'/P_r$ at point B (and its x-coordinate)

Figure 5 — Dynamic equivalent radial load, P_r, for radial contact groove ball bearings

For this type of bearing,

$$\left.\begin{array}{l} X_1 = X_2 = 1 - 0{,}4\,\dfrac{\xi}{\eta} \\[2mm] Y_1 = Y_2 = 0{,}4\,\dfrac{\cot \alpha'}{\eta} \\[2mm] X_3 = 1;\ Y_3 = 0 \end{array}\right\} \tag{72}$$

For self-aligning ball bearings, the contact angle can be considered as independent of the load ($\alpha' = \alpha$); also η can be assumed to be unity.

7.1.6   Practical expressions for dynamic equivalent axial load, P_a, for thrust bearings

The radial and axial load factors, X_a and Y_a, for single and double direction bearings with $\alpha \neq 90°$ are obtained on the basis of the expressions for dynamic equivalent radial load, P_r, for single row and double row radial bearings, respectively.

That is, for single direction bearings, when $F_a/F_r > \xi \tan \alpha$

$$Y_1 \ P_a = P_r = X_1 \ F_r + Y_1 \ F_a$$

So

$$\left. \begin{aligned} P_a &= \frac{X_1}{Y_1} F_r + F_a \equiv X_{a1} \ F_r + Y_{a1} \ F_a \\ &\text{where} \\ X_{a1} &= \frac{X_1}{Y_1}; \ Y_{a1} = 1 \end{aligned} \right\} \quad (73)$$

and for double direction bearings, when $F_a/F_r > \xi \tan \alpha$,

$$\left. \begin{aligned} P_a &= \frac{X_2}{Y_2} F_r + F_a \equiv X_{a2} \ F_r + Y_{a2} \ F_a \\ &\text{where} \\ X_{a2} &= \frac{X_2}{Y_2}; \ Y_{a2} = 1 \end{aligned} \right\} \quad (74)$$

Further, when $F_a/F_r \leq \xi \tan \alpha$, approximately

$$Y_2 \ P_a = P_r = X_3 \ F_r + Y_3 \ F_a$$

Therefore

$$\left. \begin{aligned} P_a &= \frac{X_3}{Y_2} F_r + \frac{Y_3}{Y_2} F_a \equiv X_{a3} \ F_r + Y_{a3} \ F_a \\ &\text{where} \\ X_{a3} &= \frac{X_3}{Y_2}; \ Y_{a3} = \frac{Y_3}{Y_2} \end{aligned} \right\} \quad (75)$$

Integrating the above, Table 5 shows expressions for dynamic equivalent axial load, P_a, for thrust bearings and factors X_a and Y_a.

Table 5 — Expressions for dynamic equivalent axial load, P_a, and factors X_a and Y_a for thrust bearings

		Single direction bearings	Double direction bearings
Expressions	$\dfrac{F_a}{F_r} \leq e$	—	$P_a = X_{a3}\,F_r + Y_{a3}\,F_a$
	$\dfrac{F_a}{F_r} > e$	$P_a = X_{a1}\,F_r + Y_{a1}\,F_a$	$P_a = X_{a2}\,F_r + Y_{a2}\,F_a$
Radial load factor, X_a Axial load factor, Y_a		$X_{a1} = \dfrac{X_1}{Y_1}$ $Y_{a1} = 1$	$X_{a2} = \dfrac{X_2}{Y_2}$ $Y_{a2} = 1$ $X_{a3} = \dfrac{X_3}{Y_2}$ $Y_{a3} = \dfrac{Y_3}{Y_2}$
The symbol e is the Limiting value of F_a/F_r for the applicability of different values of factors X and Y are determined by the formula.			$e = \xi\,\tan\alpha$

7.2 Factors X, Y and e

7.2.1 Radial ball bearings

7.2.1.1 Values of ξ

For single row radial contact groove ball bearings, Reference [5] gives a value of $\xi = 1,2$ based on the results of tests, and for other bearings $\xi = 1,5$ which are close to the theoretical curves. However, based on later tests, ISO/R281 took values of $\xi = 1,05$ for radial contact groove ball bearings and single row angular contact groove types with $\alpha = 5$; $\xi = 1,25$ for other angular contact groove types; and $\xi = 1,5$ for self-aligning types (Reference [4]).

7.2.1.2 Values of η

The reduction factor, η, depends on the contact angle, α, and is given by

$$\eta = 1 - k\,\sin\alpha \tag{76}$$

Based on experience and preliminary tests, Reference [5] gives $k = 0,4$ and Reference [6] $k = 0,15$ to $0,33$. In ISO/R281, $k = 0,4$ ($= 1/2,5$) was used for radial contact groove bearings ($\alpha = 5°$) and angular contact groove bearings with $\alpha = 5°$, $10°$ and $15°$ and $k = (1/2,75)$ is used for angular contact groove bearings with $\alpha = 20°$ to $45°$ (see Reference [4]).

NOTE   ISO/R281 did not include factors for bearings with $\alpha = 45°$. Factors for this angle are specified in ISO 281:2007.

7.2.1.3 Values of contact angle α'

For radial contact groove ball bearings as well as angular contact groove bearings with nominal contact angle $\alpha \leq 15°$, the real contact angle varies considerably with the load. Consequently, ISO 281:2007, Table 3, gives all factors as functions of the relative axial load.

The values of contact angle, α', under an axial load, F_a, can be calculated from

$$\left(\frac{\cos 5°}{\cos\alpha'}-1\right)^{3/2}\sin\alpha'=\left[\frac{c}{(2r/D_w)-1}\right]^{3/2}\frac{F_a}{iZD_w^2} \tag{77}$$

for radial contact groove ball bearings (considering them as angular contact groove bearings with a nominal contact angle, $\alpha = 5°$), and from Formula (64) for angular contact groove bearings with a nominal contact angle, α.

For $2r/D_w = 1,035$, $c = 0,000\ 438\ 71$ is given, with units in newtons and millimetres.

Table 6 shows the values of contact angle α' calculated from Formulae (64) and (77) for $2r/D_w = 1,035$.

For angular contact groove ball bearings with $\alpha \geq 20°$, the influence of the axial load on the contact angle is comparatively small and therefore ISO 281:2007, Table 5, has only one set of X, Y and e factors for each α. With regard to the calculation rules applied to these bearings, see 7.2.2.3.

Table 6 — Values of contact angle α' for radial and angular contact groove ball bearings
($\alpha = 5°$, 10° and 15°)

F_a/ZD_w^2 [a]		$\alpha=5°$	$\alpha=10°$	$\alpha=15°$
lbf/in²	MPa[b]	α'		
25	0,172 37	10,230°	12,953°	16,781°
50	0,344 74	11,811°	14,177°	17,652°
100	0,689 48	13,734°	15,768°	18,866°
150	1,034 21	15,037°	16,893°	19,767°
200	1,378 95	16,048°	17,786°	20,503°
300	2,068 4	17,607°	19,187°	21,688°
500	3,447 4	19,809°	21,207°	23,488°
750	5,171 1	21,761°	23,028°	25,075°
1 000	6,894 8	23,263°	24,444°	26,360°

[a] For radial contact groove bearings F_a/iZD_w^2.

[b] 1 MPa = 1 N/mm²

7.2.2 Values of X, Y and e for each type of radial ball bearing

7.2.2.1 General

Integrating the above, methods of calculating values of X, Y and e are as follows (see Tables 10 and 11).

7.2.2.2 Radial contact groove ball bearings

$$X_1 = X_2 = 1-\frac{0,4 \times 1,05}{1-0,4\sin 5°}= 0,5648 \approx 0,56$$

$$Y_1 = Y_2 = \frac{0,4\cot\alpha'}{1-0,4\sin 5°}= 0,41445\cot\alpha'$$

$$e = 1,05\tan\alpha'$$

The calculated Y_1 value of $0,964\ 1 \approx 0,96$ for $F_a/iZD_w^2 = 6,89$ MPa is adjusted to 1,00 in consideration of the relationship with the value of Y_1 for angular contact groove type with $\alpha \geq 20°$ (see Figure 6); namely, the calculated contact angle, α', of 23,262° is adjusted to 22,512° ($\alpha' = \tan^{-1} 0,414\ 45$). Therefore, the

calculated e value of 0,451 42 ≈ 0,45 becomes 0,435 2 ≈ 0,44 [e = 1,05 tan 22,512° or 0,4 × 1,05/(1 − 0,4 sin 5°)].

7.2.2.3 Angular contact groove ball bearings with $\alpha \leq 15°$

For single row bearings with α = 5°, the values of X_1, Y_1 and e are the same as those for the radial contact groove type above.

For double row bearings with α = 5°

$$X_2 = 1,625 \times 0,48 = 0,78$$

because

$$X_1 = 1 - \frac{0,4 \times 1,25}{1 - 0,4 \sin 5°} = 0,481\ 9 \approx 0,48$$

$$Y_2 = 1,625\ Y_1$$

where

$$Y_1 = \frac{0,4 \cot \alpha'}{1 - 0,4 \sin 5°} = 0,414\ 45 \cot \alpha'$$

$$Y_3 = \frac{0,625 \cot \alpha'}{1,25} = 0,5 \cot \alpha'$$

$$e = 1,25 \tan \alpha'$$

For $F_a / Z D_w^2$ = 6,89 MPa, the contact angle, α', of 22,512° is used.

Therefore,

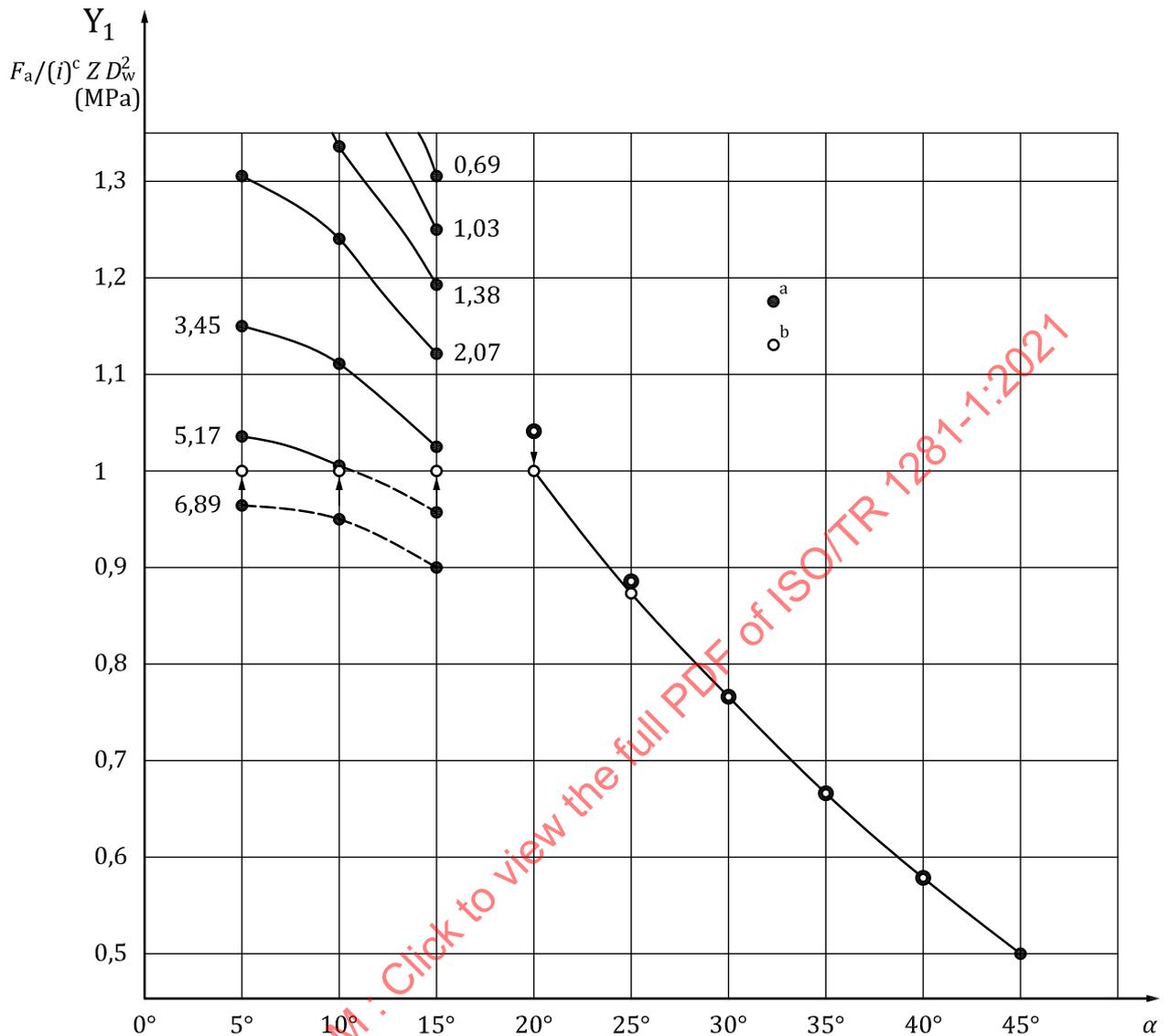$$Y_2 = 1,625 \times 1 = 1,625 \approx 1,63$$

$$Y_3 = 0,5 \cot 22,512°$$

or

$$Y_3 = \frac{1,562\ 5\ (1 - 0,4 \sin 5°)}{1,25} = 1,206\ 4 \approx 1,21$$

and

$$e = 1,25 \tan 22,512°$$

or

$$e = \frac{0,4 \times 1,25}{1 - 0,4 \sin 5°} = 0,518\ 1 \approx 0,52$$

Key

D_w	ball diameter		α	nominal contact angle
F_a	axial load		a	Calculated values.
i	number of rows of balls		b	Adjusted values.
Y_1	axial load factor		c	Include factor i for radial contact groove bearings.
Z	number of balls per row			

Figure 6 — Adjustment of Y_1 values for radial and angular contact groove ball bearings

For bearings with α = 10° and α = 15°,

$$X_1 = 1 - \frac{0,4 \times 1,25}{1 - 0,4 \sin \alpha}$$

$$X_2 = 1,625\, X_1$$

$$Y_1 = \frac{0,4 \cot \alpha'}{1 - 0,4 \sin \alpha}$$

$$Y_2 = 1,625\, Y_1$$

$$Y_3 = \frac{0,625}{1,25} \cot \alpha' = 0,5 \cot \alpha'$$

$$e = 1,25 \tan \alpha'$$

Namely, for $\alpha = 10°$, $X_1 = 0,462\ 7 \approx 0,46$, $Y_1 = 0,429\ 86 \cot \alpha'$, and for $\alpha = 15°$, $X_1 = 0,442\ 3 \approx 0,44$, $Y_1 = 0,446\ 19 \cot \alpha'$.

For the above-stated reason, in the case of the calculated value of Y_1 being less than 1, Y_1 has to be set equal to 1,00 (see Figure 6). Therefore, for $F_a / i\ Z\ D_w^2 = 6,89$ MPa

$$Y_2 = 1,625 \approx 1,63$$

$$Y_3 = 0,5 \cot 23,261°$$

or

$$Y_3 = 1,25\ (1 - 0,4 \sin 10°) = 1,163\ 2 \approx 1,16$$

and

$$e = 1,25 \tan 23,261°$$

or

$$e = \frac{0,4 \times 1,25}{1 - 0,4 \sin 10°} = 0,537\ 3 \approx 0,54$$

and also for $F_a / ZD_w^2 = 5,17$ MPa and $F_a / ZD_w^2 = 6,89$ MPa

$$Y_2 = 1,625 \times 1 = 1,625 \approx 1,63$$

$$Y_3 = 0,5 \cot 24,046°$$

or

$$Y_3 = 1,25\ (1 - 0,4 \sin 15°) = 1,120\ 6 \approx 1,12$$

and

$$e = 1,25 \tan 24,046°$$

or

$$e = \frac{0,4 \times 1,25}{1 - 0,4 \sin 15°} = 0,557\ 7 \approx 0,56$$

7.2.2.4 Angular contact groove ball bearings with α = 20° to α = 45°

$$X_1 = 1 - \frac{0,4 \times 1,25}{1 - (1/2,75)\sin\alpha}$$

See Table 7.

$$X_2 = 1,625 \, X_1$$

Table 7 — Values of X_1 for bearings with α = 20° to α = 45°

α	X_1
20°	0,429 0 ≈ 0,43
25°	0,409 2 ≈ 0,41
30°	0,388 9 ≈ 0,39
35°	0,368 2 ≈ 0,37
40°	0,347 5 ≈ 0,35
45°	0,326 9 ≈ 0,33

For values of Y_1, in principle, the values in Table 8, taken from document ISO/TC 4 N36 (= TC 4 N56 = TC 4 N110), are used (see Note), where the first and second values are adjusted, in consideration of the relationship with the values of Y_1 for $\alpha \leq 15°$ (see Figure 6).

Table 8 — Values of Y_1

α	Y_1
20°	1,04 adjusted to 1,00
25°	0,89 adjusted to 0,87
30°	0,76
35°	0,66
40°	0,57
(45°)	(0,50)

Then values of Y_2, e and Y_3 are calculated from Formula (78) which are obtained from Formula (71):

$$\left.\begin{aligned} Y_2 &= 1,625 \, Y_1 \\ e &= \frac{1 - X_1}{Y_1} \\ Y_3 &= \frac{0,625}{e} \end{aligned}\right\} \tag{78}$$

NOTE   The values of Y_1 are obtained from Formula (79):

$$Y_1 = \frac{0,4}{\eta} \cot \alpha' = \frac{0,4}{1 - (1/3) \sin \alpha'} \cot \alpha' \tag{79}$$

Where the formula $1 - (1/3) \sin \alpha'$ is given in Reference [5] and the values of contact angle α' are determined by the formula

$$\cos \alpha' = 0,972\,402 \cos \alpha$$

as in Table 9:

Table 9 — Values of α'

α	α'
20°	23,97°
25°	28,20°
30°	32,63°
35°	37,20°
40°	41,85°
(45°)	(46,56°)

The relation between α and α' is obtained from Formula (64) for

$$\frac{2r}{D_w} = \frac{r_i}{D_w} + \frac{r_e}{D_w} = 0,517\,5 + 0,53 = 1,047\,5$$

$$c_c = 0,000\,458\,35$$

and the rolling body load, in megapascals, is given by

$$\frac{F_a}{Z \, D_w^2 \, \sin \alpha'} = 4,903\,3 \text{ MPa}$$

This is equivalent to 0,5 kgf/mm².

Moreover, for bearings with $\alpha = 45°$, which were not included in ISO/R281, a Y_1 value of $0,498\,6 \approx 0,50$ is determined from the same Formula (79), and was specified first in ISO 281-1:1977 together with the values of remaining factors Y_2, Y_3 and e, which are obtained from the Formula (78).

7.2.2.5 Self-aligning ball bearings

Taking $\alpha' = \alpha$, $\eta = 1$, and $\xi = 1,5$,

$$X_1 = 1 - 0,4 \times 1,5 = 0,4$$

$$X_2 = 1,625 \times 0,4 = 0,65$$

$$Y_1 = 0,4 \cot \alpha$$

$$Y_2 = 1,625 \, Y_1 = 0,65 \cot \alpha$$

$$Y_3 = \frac{0,625}{1,5} \cot \alpha = 0,416\,7 \cot \alpha \approx 0,42 \cot \alpha$$

$$e = 1,5 \tan \alpha$$