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Rolling bearings — Methods for calculating the modified reference rating life for universally loaded rolling bearings

Roulements — Méthodes de calcul de la durée nominale de référence corrigée pour les roulements chargés universellement

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Foreword

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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 document 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).

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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 first edition of ISO 16281 cancels and replaces the first edition of ISO/TS 16281:2008, which has been technically revised. It also incorporates the Technical Corrigendum ISO/TS 16281:2008/Cor 1:2009.

The main changes are as follows:

— the coordinate system used in drawings and derivation of formulae has been changed to a right-handed coordinate system;

— the calculation of load distribution of cylindrical and tapered bearings has been described in greater detail and provisions for the calculation of load distribution and rating life of spherical roller bearings have been added;

— additional formulae have been given for the calculation of load distribution of hybrid bearings;

— reference geometries and the description of static equilibrium calculation for different bearing types have been moved to an informative annex.

This document is intended to be used in conjunction with ISO 281.

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

Since publication of the first edition of ISO 281:1990, additional knowledge has been gained regarding the influence on bearing life of contamination, lubrication, internal stresses from mounting, stresses from hardening, fatigue load limit of the material etc. It is therefore now possible to consider factors that have influence on bearing life in a more complete way in the life calculation.

ISO 281 provides a method to put into practice this new knowledge in a consistent way when the modified rating life of a bearing is calculated. However, the calculation method given in ISO 281 cannot consider the influence on life of tilted or misaligned bearings and the influence on life of bearing clearance during operation. ISO/TS 16281:2008 already describes an advanced calculation method, which makes it possible to consider these influences, and in addition provides the most accurate method for estimating the influence of contamination and other factors.

In addition to the content of ISO/TS 16281:2008, this document also addresses the analysis of hybrid bearings with rolling elements made of silicon nitride.

The primary purpose of this document is to provide a unified and manufacturer-independent advanced calculation method that allows for the consideration of actual operating conditions, thus enabling the end user to compare different bearing solutions on the same calculation basis. It is also intended to serve as a manufacturer-independent neutral basis for certification purposes, e.g. as required per IEC 61400-4[1] for bearings in wind turbine gearboxes.

This document is intended to be used for computer programs and together with ISO 281 covers the information needed for life calculations. For accurate life calculations under the operating conditions specified above, this document or advanced computer calculations should be used for determining the dynamic equivalent reference load under different loading conditions.

This document is not intended to supersede other advanced bearing analysis methods that are currently used in the design process as the primary tool for bearing design and selection.

Rolling bearings — Methods for calculating the modified reference rating life for universally loaded rolling bearings

1 Scope

This document defines the calculation of the modified reference rating life taking into consideration lubrication, contamination, and fatigue load limit of the bearing material, as well as tilting or misalignment, operating clearance of the bearing and internal load distribution on rolling elements. The calculation method provided in this document covers influencing parameters additional to those described in ISO 281.

The general directions and limitations given in ISO 281 apply to this document. The calculation methods pertain to the fatigue life of the bearings. Other mechanisms of failure, like wear or microspalling (gray-staining), lie outside the scope of this document.

This document applies to single- and multi-row radial and thrust ball bearings, subjected to radial and axial load and with radial clearance and tilt taken into account. It also applies to single- and multi-row radial and thrust roller bearings, subjected to radial and axial load and with radial clearance, edge stress and tilt taken into account. References to methods for the analysis of the internal load distribution under general load are given.

The calculation of load distribution and basic reference rating life is also applicable to hybrid bearings, using the dynamic load ratings per ISO 20056-1[2]. The calculation of the modified reference rating life is not applicable to hybrid bearings.

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*

ISO 5593, *Rolling bearings — Vocabulary*

ISO 15241, *Rolling bearings — Symbols for physical quantities*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 281, ISO 5593 and the following apply.

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

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

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

3.1
basic reference rating life
rating life associated with 90 % reliability for rolling bearings manufactured with commonly used high quality material, of good manufacturing quality, under consideration of actual load distribution in the bearing

3.2
modified reference rating life
rating life for 90 % or other reliability, for rolling bearings manufactured with commonly used high quality material, of good manufacturing quality, under consideration of actual load distribution, lubrication conditions, lubricant contamination and bearing fatigue load limit

Note 1 to entry: Life values for reliabilities higher than 90 % are denoted by the index n, where n = 100 – percentage of reliability.

3.3
deflection
change in position caused by elastic deformation, e.g. in a contact

3.4
displacement
change in position caused by rigid body motion, e.g. axial movement of rolling elements caused by tilting of the inner ring

3.5
initial contact angle
free contact angle
contact angle at initial contact between rolling element and both raceways, when an infinitesimal small axial load on bearing is applied

Note 1 to entry: The initial contact angle, α_0, is generally not identical to the nominal contact angle, α, in ISO 281.

4 Symbols

For the purpose of this document, the symbols given in ISO 15241 and the following apply.

A distance between raceway groove curvature centres of ball bearing having no clearance and having an initial contact angle, in millimetres

a semi-major axis of the contact ellipse of ball bearings, in millimetres

a_{ISO} life modification factor based on a systems approach of life calculation

$a_{\mathrm{ISO}k}$ life modification factor for lamina k of a roller bearing, based on a systems approach of life calculation

$a_{\mathrm{ISO}m}$ life modification factor for row m of a multi-row ball bearing, based on a systems approach of life calculation

a_1 life modification factor for reliability

b semi-minor axis of the contact ellipse of a ball bearing, in millimetres

C_{a} basic dynamic axial load rating according to ISO 281 or ISO 20056-1, in newtons

C_{r} basic dynamic radial load rating according to ISO 281 or ISO 20056-1, in newtons

C_{u} fatigue load limit, in newtons

c_{L} spring constant of a rolling element with line contact, in newtons per millimetre to the power of 10/9

c_P spring constant of a rolling element with point contact, in newtons per millimetre to the power of 3/2

c_{s} spring constant of a roller lamina, in newtons per millimetre to the power of 10/9

c_{T} spring constant of a tapered roller, in newtons per millimetre to the power of 10/9

D_{pw} pitch diameter of ball or roller set, in millimetres

D_{pwk} pitch diameter at lamina k for bearings having rollers with non-constant diameter, in millimetres

D_w nominal ball diameter, in millimetres

D_{we} roller diameter applicable in the calculation of load ratings, in millimetres

D_{wk} roller diameter at lamina k for rollers with non-constant diameter, in millimetres

d distance to the centre of contact of a tapered roller bearing, measured from the axial locating face of the outer ring, in millimetres

E modulus of elasticity, in megapascals (1 MPa = 1 N/mm^2)

NOTE 1 Elasticity constants used in this standard are based on E_{st} = 207 000 MPa for steel and E_{ce} = 300 000 MPa for Si_3N_4.

$E(\chi)$ complete elliptic integral of the second kind

$F(\rho)$ relative curvature difference for point contact

$F_e(\rho)$ relative curvature difference for point contact at outer ring

$F_i(\rho)$ relative curvature difference for point contact at inner ring

e subscript for outer ring or housing washer

e_C contamination factor

F_a bearing axial load (axial component of actual bearing load) acting at bearing rotation axis, in newtons

F_r bearing radial load (radial component of actual bearing load) acting at centre of bearing, in newtons

$f[j,k]$ load correction function for consideration of edge load

$f_e[j,k]$ load correction function for consideration of edge load at outer ring contact

$f_i[j,k]$ load correction function for consideration of edge load at inner ring contact

$G_{r\,op}$ radial operating clearance of bearing, in millimetres

i subscript for inner ring or shaft washer

i number of rows of rolling elements

j subscript for individual rolling element

$K(\chi)$ complete elliptic integral of the first kind

$K(\chi_e)$ complete elliptic integral of the first kind for point contact at outer ring

$K(\chi_i)$ complete elliptic integral of the first kind for point contact at inner ring

k subscript for individual lamina of a roller

L_{nmr} modified reference rating life, in million revolutions

NOTE 2 The subscript r in L_{nmr} denotes “reference”.

L_{we} effective contact length of a roller applicable in the calculation of load ratings, as defined in ISO 281, in millimetres

NOTE 3 For rollers of cylindrical and spherical roller bearings, L_{we} is defined along the roller axis. For tapered roller bearings, L_{we} is defined along the roller contact line.

L_{10r} basic reference rating life, in million revolutions

M_z moment acting on tilted bearing, in newton millimetres

m subscript for individual row of a multi-row bearing

NOTE 4 Subscripts are used in the order j,k,m, separated by commas, e.g. $q_{j,k,m}$ denotes the load on lamina k of roller j of row m.

n speed of rotation, in revolutions per minute

n_s number of laminae per roller

P dynamic equivalent load per ISO 281, in newtons

$P(x_k)$ profile function, in millimetres

$P_{ref\,a}$ dynamic equivalent reference axial load, in newtons

$P_{ref\,am}$ dynamic equivalent reference axial load of row m, in newtons

$P_{ref\,r}$ dynamic equivalent reference radial load, in newtons

$P_{ref\,rm}$ dynamic equivalent reference radial load of row m, in newtons

P_{sk} dynamic equivalent load of a bearing lamina k, in newtons

$P_{sk,m}$ dynamic equivalent load of a bearing lamina k of row m, in newtons

p_{He} contact stress at the contact of outer ring and rolling element, in megapascals

p_{Hi} contact stress at the contact of inner ring and rolling element, in megapascals

Q nominal force between a rolling element and the raceways, in newtons

Q_c rolling element load for the basic dynamic load rating of the bearing per ISO/TR 1281-1[3], in newtons

Q_{ce} equivalent nominal force between a rolling element and the raceways for the basic dynamic load rating of outer ring or housing washer, in newtons

Q_{ci} equivalent nominal force between a rolling element and the raceways for the basic dynamic load rating of inner ring or shaft washer, in newtons

Q_{ee} dynamic equivalent rolling element load on outer ring or housing washer, in newtons

$Q_{ee\,m}$ dynamic equivalent rolling element load on outer ring or housing washer of row m, in newtons

Q_{ei} dynamic equivalent rolling element load on inner ring or shaft washer, in newtons

$Q_{ei\,m}$ dynamic equivalent rolling element load on inner ring or shaft washer of row m, in newtons

Q_j rolling element load of rolling element j, in newtons

q_{ce} basic dynamic load rating of a bearing lamina at the outer ring or housing washer contact, in newtons

q_{ci} basic dynamic load rating of a bearing lamina at the inner ring or shaft washer contact, in newtons

$q_{ee\,k}$ dynamic equivalent rolling element lamina load of lamina k at outer ring or housing washer, in newtons

$q_{ee\,k,m}$ dynamic equivalent rolling element lamina load of lamina k of row m at outer ring or housing washer, in newtons

$q_{ei\,k}$ dynamic equivalent rolling element lamina load of lamina k at inner ring or shaft washer, in newtons

$q_{ei\,k,m}$ dynamic equivalent rolling element lamina load of lamina k of row m at inner ring or shaft washer, in newtons

$q_{j,k}$ load on the lamina k of roller j, in newtons

$q'_{j,k}$ corrected load on the lamina k of roller j, in newtons

R_i distance between the centre of curvature of the inner raceway groove and the axis of rotation, in millimetres

R_p convex curvature radius of spherical rollers, in millimetres

r_e cross-sectional raceway groove or spherical raceway curvature radius of outer ring or housing washer, in millimetres

r_i cross-sectional raceway groove or raceway curvature radius of inner ring or shaft washer, in millimetres

T total width of a tapered roller bearing, in millimetres

x, y, z axes of a right-handed coordinate system, where x is defined along the rotation axis of the bearing

x_k distance between centre of lamina k and roller centre, in millimetres

NOTE 5 x_k is measured along the rolling element axis for cylindrical and spherical roller bearings, and along the lateral surface for tapered roller bearings.

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

α nominal contact angle, in degrees

α_j operating contact angle of the rolling element j, in degrees

α_0 initial contact angle, in degrees

β half cone angle of a tapered roller, in degrees

γ auxiliary parameter, $\gamma = D_w \cos\alpha / D_{pw}$ for ball bearings, $\gamma = D_{we} \cos\alpha / D_{pw}$ for roller bearings

γ_k auxiliary parameter for the load correction function, $\gamma_k = D_{wk} \cos\alpha / D_{pwk}$ for rollers with non-constant diameter

δ total elastic deflection of both contacts of a rolling element, in millimetres

δ_a relative axial displacement of centre points on axis of both bearing rings, in millimetres

δ_e elastic deflection at outer ring point contact of a rolling element, in millimetres

δ_i elastic deflection at inner ring point contact of a rolling element, in millimetres

δ_j elastic deflection of both contacts of the rolling element j, in millimetres

$\delta_{j,k}$ elastic deflection of both contacts of the lamina k of the roller j, in millimetres

δ_L total elastic deflection of both contacts of a rolling element with line contact, in millimetres

δ_r relative radial displacement of centre points on axis of the bearing, in millimetres

κ viscosity ratio per ISO 281

λ reduction factor for the consideration of stress concentrations

ν adjustment factor for exponent variation

ν_E Poisson's ratio

NOTE 6 Elasticity constants used in this standard are based on ν_{Est} = 0,3 for steel and ν_{Ece} = 0,26 for Si_3N_4.

ρ curvature of the contact surface, in reciprocal millimetres

$\Sigma\rho_e$ curvature sum of point contact at outer ring, in reciprocal millimetres

$\Sigma\rho_i$ curvature sum of point contact at inner ring, in reciprocal millimetres

ϕ auxiliary angle for integration over contact ellipse, in radians

φ_j angular position of rolling element j, in degrees

χ_e ratio of semi-major to semi-minor axis of the contact ellipse at outer ring of ball bearings, a/b

χ_i ratio of semi-major to semi-minor axis of the contact ellipse at inner ring of ball bearings, a/b

ψ total misalignment between inner raceway and outer raceway, in degrees

ψ_j total misalignment between inner raceway and outer raceway in the plane of rolling element j, in degrees

5 Rating life analysis

5.1 General

This clause describes the analysis of the basic and modified reference rating life for ball and roller bearings.

This life analysis is based on the calculation of each rolling element load for the basic dynamic load rating and each force between rolling element and raceway. Methods for the calculation of each force between rolling element and raceway are described in Annex A.

Calculation methods concerning the analysis of bearings of different geometry or for more complex load cases can be derived from the formulae in Annex A.

In the life analysis for multi-row bearings, it is assumed that all rows are symmetrical and have identical rolling element sets. Formulae for multi-row bearings with deviating geometry can be derived from the formulae given in this clause and in ISO/TR 1281-1[3].

Four-point-contact ball bearings can be approximated as double-row angular contact ball bearings, if these are mounted radially free to take predominantly thrust load, i.e. having only two-point contacts at every ball.

The analysis of internal load distribution and modified reference rating life for bearings of a more complex geometry can be derived from the formulae given in this document. For these bearings, the load distribution for each individual row shall be considered.

There is a discontinuity in the load ratings per ISO 281, which leads to a discontinuity of calculated life results, which also appears in the basic and modified reference rating life. Details are given in Annex F.

5.2 Ball bearings

5.2.1 Rolling element load for the basic dynamic load rating

5.2.1.1 General

The equivalent nominal rolling element load for the basic dynamic load ratings for inner rings and outer rings, Q_{ci} and Q_{ce}, are derived from the rolling element load rating Q_c in ISO/TR 1281-1[3]. For all types of ball bearings, the values of the curvature radii r_i and r_e used in Formulae (1) to (4) shall be the same as the values used in the calculation of the basic dynamic load rating.

5.2.1.2 Radial ball bearings

For the inner ring, Q_{ci} shall be calculated by means of the basic dynamic radial load rating, C_r, for single-row and multi-row bearings.

$$Q_{ci} = \frac{C_r}{0{,}407\, Z\, (\cos\alpha)\, i^{7/10}} \left(1 + \left\{ 1{,}044 \left(\frac{1-\gamma}{1+\gamma} \right)^{1{,}72} \left[\frac{r_i}{r_e} \left(\frac{2\, r_e - D_w}{2\, r_i - D_w} \right) \right]^{0{,}41} \right\}^{\frac{10}{3}} \right)^{\frac{3}{10}} \tag{1}$$

For the outer ring, Q_{ce} shall be calculated by means of the basic dynamic radial load rating, C_r for single-row and multi-row bearings.

$$Q_{ce} = \frac{C_r}{0{,}389\, Z\, (\cos\alpha)\, i^{7/10}} \left(1 + \left\{ 1{,}044 \left(\frac{1-\gamma}{1+\gamma} \right)^{1{,}72} \left[\frac{r_i}{r_e} \left(\frac{2\, r_e - D_w}{2\, r_i - D_w} \right) \right]^{0{,}41} \right\}^{-\frac{10}{3}} \right)^{\frac{3}{10}} \tag{2}$$

5.2.1.3 Thrust ball bearings

For the inner ring or shaft washer, Q_{ci} shall be calculated using the basic dynamic axial load rating, C_a.

$$Q_{ci} = \frac{C_a}{Z \sin\alpha} \left(1 + \left\{ \left(\frac{1-\gamma}{1+\gamma} \right)^{1{,}72} \left[\frac{r_i}{r_e} \left(\frac{2\, r_e - D_w}{2\, r_i - D_w} \right) \right]^{0{,}41} \right\}^{\frac{10}{3}} \right)^{\frac{3}{10}} \tag{3}$$

For the outer ring or housing washer, Q_{ce} shall be calculated using the basic dynamic axial load rating, C_a.

$$Q_{ce} = \frac{C_a}{Z \sin\alpha} \left(1 + \left\{ \left(\frac{1-\gamma}{1+\gamma} \right)^{1{,}72} \left[\frac{r_i}{r_e} \left(\frac{2\, r_e - D_w}{2\, r_i - D_w} \right) \right]^{0{,}41} \right\}^{-\frac{10}{3}} \right)^{\frac{3}{10}} \tag{4}$$

5.2.2 Dynamic equivalent rolling element load

The calculation of the dynamic equivalent rolling element load is based on rolling element loads Q_j obtained from the static equilibrium analysis. The basic principle of static equilibrium analysis is described in Annex A.

The dynamic equivalent rolling element load on a single-row inner ring or a shaft washer, Q_{ei}, which is rotating relative to the bearing load is given by Formula (5):

$$Q_{ei} = \left(\frac{1}{Z} \sum_{j=1}^{Z} Q_j^{\,3} \right)^{\frac{1}{3}} \tag{5}$$

and on a single-row inner ring or a shaft washer which is stationary relative to the bearing load is given by Formula (6):

$$Q_{ei} = \left(\frac{1}{Z} \sum_{j=1}^{Z} Q_j^{\,\frac{10}{3}} \right)^{\frac{3}{10}} \tag{6}$$

The dynamic equivalent rolling element load on a single-row outer ring or a housing washer, Q_{ee}, which is stationary relative to the bearing load is given by Formula (7):

$$Q_{ee} = \left(\frac{1}{Z} \sum_{j=1}^{Z} Q_j^{\,\frac{10}{3}} \right)^{\frac{3}{10}} \tag{7}$$

and on a single-row outer ring or a housing washer which is rotating relative to the bearing load is given by Formula (8):

$$Q_{ee} = \left(\frac{1}{Z} \sum_{j=1}^{Z} Q_j^{\,3} \right)^{\frac{1}{3}} \tag{8}$$

For a normal load distribution, i.e. no significant elastic deformation of the outer ring, the difference between the dynamic equivalent rolling element loads for a rotating and a stationary inner ring is less than 2 %. This can generally be neglected, especially as the deviation of dynamic equivalent rolling element loads on inner ring and outer ring partially compensate each other.

When calculations are carried out, the inner ring is generally considered to be rotating and the outer ring to be stationary.

5.2.3 Basic reference rating life

Using the equivalent nominal rolling element load for the basic dynamic load ratings and the dynamic equivalent rolling element loads, the basic reference rating life, L_{10r}, of a single-row bearing is calculated as shown in Formula (9):

$$L_{10r} = \left[\left(\frac{Q_{ci}}{Q_{ei}} \right)^{-\frac{10}{3}} + \left(\frac{Q_{ce}}{Q_{ee}} \right)^{-\frac{10}{3}} \right]^{-\frac{9}{10}} \tag{9}$$

The basic reference life L_{10r} for a multi-row bearing is given by Formula (10):

$$L_{10r}=\left(\sum_{m=1}^{i}\left[\left(\frac{Q_{ci}}{Q_{eim}}\right)^{-\frac{10}{3}}+\left(\frac{Q_{ce}}{Q_{eem}}\right)^{-\frac{10}{3}}\right]\right)^{-\frac{9}{10}} \tag{10}$$

5.2.4 Dynamic equivalent reference load

The dynamic equivalent reference load for radial ball bearings, $P_{ref\,r}$, is given by Formula (11):

$$P_{ref\,r}=\frac{C_r}{L_{10r}^{\frac{1}{3}}} \tag{11}$$

and for thrust (axial) ball bearings, $P_{ref\,a}$, is given by Formula (12):

$$P_{ref\,a}=\frac{C_a}{L_{10r}^{\frac{1}{3}}} \tag{12}$$

5.2.5 Modified reference rating life

5.2.5.1 General

The calculation of the modified reference rating life is only applicable for bearings with balls and raceway made of steel. For details on the calculation of a_{ISO}, see Annex D. The life modification factor a_{ISO} per ISO 281 has not been defined for hybrid bearings.

5.2.5.2 Single-row ball bearings

The modified reference rating life, L_{nmr}, for single-row ball bearings is calculated by means of the life modification factor a_{ISO} as per Formula (13):

$$L_{nmr}=a_1\, a_{ISO}\, L_{10r} \tag{13}$$

where a_{ISO} shall be calculated in function of $(\frac{e_C\, C_u}{P_{ref\,r}},\kappa)$ per ISO 281:2007, Formulae (31) to (33) for radial ball bearings, and in function of $(\frac{e_C\, C_u}{P_{ref\,a}},\kappa)$ per ISO 281:2007, Formulae (37) to (39) for thrust ball bearings.

It should be noted that in ISO 281:2007, a_{ISO} is defined in function of $\left(\frac{e_C\, C_u}{P},\kappa\right)$. For the calculation of a_{ISO} for the modified reference rating life, the dynamic equivalent load P shall be substituted by the reference load, $P_{ref\,r}$ or $P_{ref\,a}$, respectively.

5.2.5.3 Multi-row ball bearings

The modified reference rating life, L_{nmr}, for multi-row ball bearings is calculated by Formula (14):

$$L_{nmr}=a_1\left(\sum_{m=1}^{i}\left\{a_{ISOm}^{-\frac{10}{9}}\left[\left(\frac{Q_{ci}}{Q_{eim}}\right)^{-\frac{10}{3}}+\left(\frac{Q_{ce}}{Q_{eem}}\right)^{-\frac{10}{3}}\right]\right\}\right)^{-\frac{9}{10}} \tag{14}$$

where the life modification factor a_{ISOm} for row m of a multi-row ball bearing shall be calculated in function of $\left(\frac{e_C\, C_u}{i\, P_{ref\,rm}},\kappa\right)$ per ISO 281:2007, Formulae (31) to (33) for multi-row radial ball bearings, and in function of $\left(\frac{e_C\, C_u}{i\, P_{ref\, am}}, \kappa\right)$ per ISO 281:2007, Formulae (37) to (39) for multi-row thrust ball bearings. It should be noted that in ISO 281:2007, a_{ISO} is defined in function of $\left(\frac{e_C\, C_u}{P}, \kappa\right)$. For the calculation of a_{ISOm} for the modified reference rating life, the dynamic equivalent load P shall be substituted by the term, $(i\, P_{ref\, rm})$ or $(i\, P_{ref\, am})$, respectively.

The dynamic equivalent reference load of row m of a multi-row radial ball bearing is calculated by Formula (15):

$$P_{ref\, rm} = \frac{C_r}{i^{\frac{7}{10}} \left[\left(\frac{Q_{ci}}{Q_{eim}} \right)^{-\frac{10}{3}} + \left(\frac{Q_{ce}}{Q_{eem}} \right)^{-\frac{10}{3}} \right]^{-\frac{3}{10}}} \quad (15)$$

and for row m of a multi-row thrust ball bearing calculated by Formula (16):

$$P_{ref\, am} = \frac{C_a}{i^{\frac{7}{10}} \left[\left(\frac{Q_{ci}}{Q_{eim}} \right)^{-\frac{10}{3}} + \left(\frac{Q_{ce}}{Q_{eem}} \right)^{-\frac{10}{3}} \right]^{-\frac{3}{10}}} \quad (16)$$

5.3 Roller bearings

5.3.1 General

The rating life of roller bearings is calculated on a per-lamina basis to account for the effects of roller profile and uneven load distribution within the bearing. The analysis of bearing internal load distribution using a laminum model is described in Annex A. The formulae given in this subclause assume the use of a laminum model with constant lamina width. Formulae for the use of a variable lamina width can be derived accordingly.

The number of laminae per roller, n_s, which can be even or odd, shall not be less than 30. If a variable lamina width is used, the maximum lamina width shall not be larger than $\frac{1}{30} L_{we}$.

5.3.2 Rolling element load for the basic dynamic load rating

5.3.2.1 General

The rolling element load for the basic dynamic load ratings for inner rings and outer rings, Q_{ci} and Q_{ce}, are derived from the rolling element load rating Q_c defined in ISO/TR 1281-1[3].

5.3.2.2 Radial roller bearings

The equivalent nominal rolling element load for the basic dynamic load ratings for inner rings, Q_{ci}, and outer rings, Q_{ce}, shall be calculated, respectively by Formulae (17) and (18), using the basic dynamic radial load rating, C_r, for single-row and multi-row bearings:

$$Q_{ci} = \frac{1}{\lambda \nu} \frac{C_r}{0{,}378\, Z (\cos\alpha)\, i^{\frac{7}{9}}} \left\{ 1 + \left[1{,}038 \left(\frac{1-\gamma}{1+\gamma} \right)^{\frac{143}{108}} \right]^{\frac{9}{2}} \right\}^{\frac{2}{9}} \tag{17}$$

$$Q_{ce} = \frac{1}{\lambda \nu} \frac{C_r}{0{,}364\, Z (\cos\alpha)\, i^{\frac{7}{9}}} \left\{ 1 + \left[1{,}038 \left(\frac{1-\gamma}{1+\gamma} \right)^{\frac{143}{108}} \right]^{-\frac{9}{2}} \right\}^{\frac{2}{9}} \tag{18}$$

with

$$\lambda \nu = 0{,}83 \tag{19}$$

according to ISO/TR 1281-1[3].

This value of $\lambda \nu$ requires a detailed analysis of the contact stress as described in C.1, or by applying the approximate function for the stress concentration given in C.2.

5.3.2.3 Thrust roller bearings

The equivalent nominal rolling element load for the basic dynamic load ratings for inner rings or shaft washers, Q_{ci}, and outer rings or housing washers, Q_{ce}, shall be calculated, respectively by Formulae (20) and (21), using the basic dynamic axial load rating, C_a:

$$Q_{ci} = \frac{1}{\lambda \nu} \frac{C_a}{Z \sin\alpha} \left\{ 1 + \left[\left(\frac{1-\gamma}{1+\gamma} \right)^{\frac{143}{108}} \right]^{\frac{9}{2}} \right\}^{\frac{2}{9}} \tag{20}$$

$$Q_{ce} = \frac{1}{\lambda \nu} \frac{C_a}{Z \sin\alpha} \left\{ 1 + \left[\left(\frac{1-\gamma}{1+\gamma} \right)^{\frac{143}{108}} \right]^{-\frac{9}{2}} \right\}^{\frac{2}{9}} \tag{21}$$

with

$$\lambda \nu = 0{,}73 \tag{22}$$

according to ISO/TR 1281-1[3].

This value of $\lambda \nu$ requires a detailed analysis of the contact stress as described in C.1 or by applying the approximate function for the stress concentration given in C.2.

5.3.3 Basic dynamic load rating of a bearing lamina

The basic dynamic load rating of a bearing lamina at the inner ring, q_{ci}, is calculated by Formula (23):

$$q_{ci} = Q_{ci} \left(\frac{1}{n_s} \right)^{\frac{7}{9}} \tag{23}$$

The basic dynamic load rating of a bearing lamina at the outer ring, q_{ce}, is calculated by Formula (24):

$$q_{ce} = Q_{ce} \left(\frac{1}{n_s} \right)^{\frac{7}{9}} \tag{24}$$

5.3.4 Concentration of edge stress

In cases where the rollers are excessively loaded, or insufficiently profiled for the operating load, or severely misaligned, edge stresses can arise which shall be taken into account in the rating life calculation.

The distribution of actual non-Hertzian contact stress over the length of the rollers can be calculated by means of References [4], [5] or [6] or similar advanced computer programs for non-Hertzian contact analysis, see C.1.

The lamina load for each loaded lamina of each roller shall be corrected by a load correction function, $f[j,k]$, so that the nominal Hertzian contact stress, calculated for a cylinder of diameter and width of each lamina, matches the highest actual contact stress over the width of that lamina, as obtained by the methods given above.

As a first approximation, a load correction function, $f[j,k]$, given in C.2, can be used. An actual analysis of the stress concentration as described in C.1 is preferred.

5.3.5 Dynamic equivalent load on a lamina

To consider the actual non-Hertzian contact stress distribution, the dynamic equivalent rolling element lamina load of k at inner ring, q_{eik}, which is rotating relative to the load is calculated from the lamina loads, corrected for actual contact stress, $q'_{j,k}$ (see C.1 and C.2) given by Formula (25):

$$q_{eik} = \left[\frac{1}{Z} \sum_{j=1}^{Z} \left(q'_{j,k} \right)^4 \right]^{\frac{1}{4}} \tag{25}$$

The dynamic equivalent rolling element lamina load of k at inner ring, q_{eik}, which is stationary relative to the load is calculated by Formula (26):

$$q_{eik} = \left[\frac{1}{Z} \sum_{j=1}^{Z} \left(q'_{j,k} \right)^{\frac{9}{2}} \right]^{\frac{2}{9}} \tag{26}$$

The dynamic equivalent rolling element lamina load of k at inner ring, q_{eek}, which is stationary relative to the load is calculated by Formula (27):

$$q_{eek} = \left[\frac{1}{Z} \sum_{j=1}^{Z} \left(q'_{j,k} \right)^{\frac{9}{2}} \right]^{\frac{2}{9}} \tag{27}$$

The dynamic equivalent rolling element lamina load of k at inner ring, q_{eek}, which is rotating relative to the load is calculated by Formula (28):

$$q_{eek} = \left[\frac{1}{Z} \sum_{j=1}^{Z} \left(q'_{j,k} \right)^4 \right]^{\frac{1}{4}} \tag{28}$$

For a normal load distribution, i.e. no significant elastic deformation of the outer ring, the difference between the dynamic equivalent rolling element loads for a rotating and a stationary inner ring is less than 2 %. This can generally be neglected, especially as the deviation of dynamic equivalent rolling element loads on inner ring and outer ring partially compensate each other.

When calculations are carried out, the inner ring is generally considered to be rotating and the outer ring to be stationary.

For multi-row roller bearings, the dynamic equivalent load is calculated on a per row basis, using the syntax $q_{eik,m} = q_{eik}$ for lamina k of row m, and for $q_{eek,m}$ accordingly.

5.3.6 Basic reference rating life

The basic reference rating life, L_{10r}, for single-row roller bearings is calculated by Formula (29):

$$L_{10r} = \left\{ \sum_{k=1}^{n_s} \left[\left(\frac{q_{ci}}{q_{eik}} \right)^{-\frac{9}{2}} + \left(\frac{q_{ce}}{q_{eek}} \right)^{-\frac{9}{2}} \right] \right\}^{-\frac{8}{9}} \tag{29}$$

The basic reference rating life, L_{10r}, for multi-row roller bearings is calculated by Formula (30):

$$L_{10r} = \left(\sum_{m=1}^{i} \left\{ \sum_{k=1}^{n_s} \left[\left(\frac{q_{ci}}{q_{eik,m}} \right)^{-\frac{9}{2}} + \left(\frac{q_{ce}}{q_{eek,m}} \right)^{-\frac{9}{2}} \right] \right\} \right)^{-\frac{8}{9}} \tag{30}$$

5.3.7 Dynamic equivalent reference load of a bearing lamina

The dynamic equivalent load, $P_{sk,m}$, of the bearing lamina k of row m of a radial roller bearing is calculated by Formula (31):

$$P_{sk,m} = 0{,}323\, Z \left(\cos\alpha\right) n_s \left\{ \frac{\left[q_{eik,m}^{\frac{9}{2}} + \left(1{,}038 \frac{q_{ci}}{q_{ce}} q_{eek,m} \right)^{\frac{9}{2}} \right]}{\left[1 + \left(1{,}038 \frac{q_{ci}}{q_{ce}} \right)^{\frac{9}{2}} \right]} \right\}^{\frac{2}{9}} \quad (31)$$

For single-row thrust roller bearings, the dynamic equivalent load, P_{sk}, of the bearing lamina k is calculated by Formula (32):

$$P_{sk} = Z \left(\sin\alpha\right) n_s \left(\frac{q_{eik}^{\frac{9}{2}} + q_{eek}^{\frac{9}{2}}}{2} \right)^{\frac{2}{9}} \quad (32)$$

5.3.8 Modified reference rating life

For single-row radial roller bearings, the modified reference rating life, L_{nmr}, is calculated by Formula (33):

$$L_{nmr} = a_1 \left(\sum_{k=1}^{n_s} \left\{ a_{ISOk}^{-\frac{9}{8}} \left[\left(\frac{q_{ci}}{q_{eik}} \right)^{-\frac{9}{2}} + \left(\frac{q_{ce}}{q_{eek}} \right)^{-\frac{9}{2}} \right] \right\} \right)^{-\frac{8}{9}} \quad (33)$$

where the life modification factor a_{ISOk} is calculated on a per-lamina basis in function of $\left(\frac{e_C C_u}{P_{sk}}, \kappa \right)$ per ISO 281:2007, Formulae (34) to (36).

For multi-row radial roller bearings, the modified reference rating life, L_{nmr}, is calculated by Formula (34):

$$L_{nmr} = a_1 \left[\sum_{m=1}^{i} \left(\sum_{k=1}^{n_s} \left\{ a_{ISOk,m}^{-\frac{9}{8}} \left[\left(\frac{q_{ci}}{q_{eik,m}} \right)^{-\frac{9}{2}} + \left(\frac{q_{ce}}{q_{eek,m}} \right)^{-\frac{9}{2}} \right] \right\} \right) \right]^{-\frac{8}{9}} \quad (34)$$

where the life modification factor $a_{ISOk,m}$ for row m is calculated on a per lamina basis in function of $\left(\frac{e_C C_u}{i\, P_{sk,m}}, \kappa \right)$ per ISO 281:2007, Formulae (34) to (36).

For thrust roller bearings, the modified reference rating life, L_{nmr}, is calculated by Formula (35):

$$L_{nmr} = a_1 \left(\sum_{k=1}^{n_s} \left\{ a_{ISOk}^{-\frac{9}{8}} \left[\left(\frac{q_{ci}}{q_{eik}} \right)^{-\frac{9}{2}} + \left(\frac{q_{ce}}{q_{eek}} \right)^{-\frac{9}{2}} \right] \right\} \right)^{-\frac{8}{9}} \quad (35)$$

where the life modification factor a_{ISOk} is calculated on a per lamina basis in function of $\left(\frac{e_C C_u}{P_{sk}}, \kappa \right)$ per ISO 281:2007, Formulae (40) to (42).

It should be noted that in ISO 281:2007 a_{ISO} is defined in function of $\left(\frac{e_{\mathrm{C}}\,C_{\mathrm{u}}}{P},\kappa\right)$. For the calculation of a_{ISO} for the modified reference rating life, the dynamic equivalent load, P, shall be substituted by the reference load, $P_{sk,m}$ or P_{sk}, respectively.

6 Elastic deflection of point- and line contact

6.1 General

The calculated bearing internal load distribution is significantly affected by the formula set used for the calculation of the elastic deflection in the rolling contacts.

The formulae sets given in 6.2 and 6.3 for the elastic deflection shall be used in the analysis of the reference rating life to ensure comparability of results.

NOTE These formulae sets define the total elastic deflection between the raceways, i.e. the sum of the deflections of both contacts, see Figure 1.

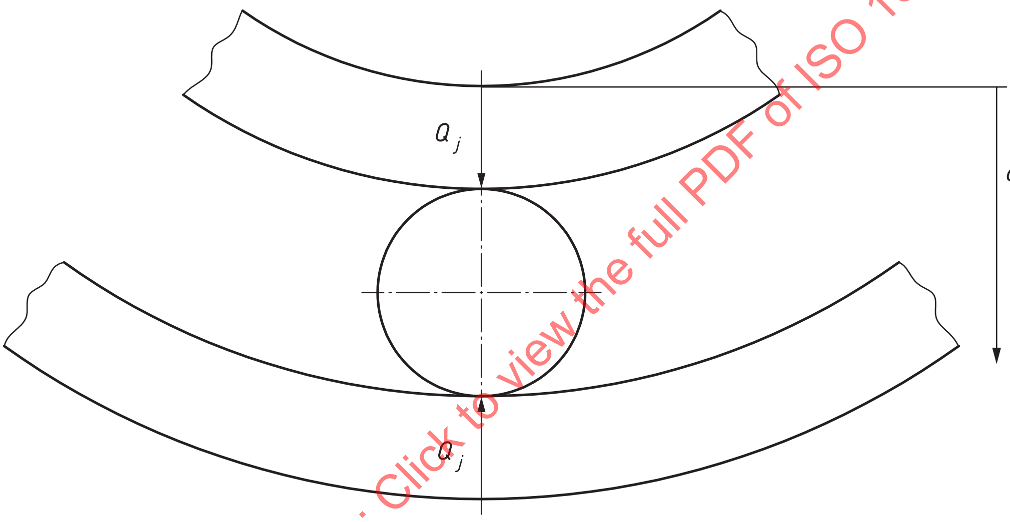


Figure 1 — Total deflection of rolling element contacts

6.2 Elastic deflection of point contact

The elastic deflection of a point contact can be calculated from Hertzian theory. The elastic deflection of a single point contact is given by Formula (36) at the inner ring contact:

$$\delta_\mathrm{i} = \sqrt[3]{4{,}5\left(\frac{1-\nu_E{}^2}{\pi E}\right)^2} K(\chi_\mathrm{i}) \sqrt[3]{\frac{\Sigma\rho_\mathrm{i}}{\chi_\mathrm{i}^2 E(\chi_\mathrm{i})}} Q^{\frac{2}{3}} \tag{36}$$

and by Formula (37) for the outer ring contact:

$$\delta_\mathrm{e} = \sqrt[3]{4{,}5\left(\frac{1-\nu_E{}^2}{\pi E}\right)^2} K(\chi_\mathrm{e}) \sqrt[3]{\frac{\Sigma\rho_\mathrm{e}}{\chi_\mathrm{e}{}^2 E(\chi_\mathrm{e})}} Q^{\frac{2}{3}} \tag{37}$$

The total elastic deflection of both contacts at inner ring and outer ring, δ, is given by Formula (38):

$$\delta = \sqrt[3]{4{,}5\left(\frac{1-\nu_E{}^2}{\pi E}\right)^2} \left[K(\chi_\mathrm{i}) \sqrt[3]{\frac{\Sigma\rho_\mathrm{i}}{\chi_\mathrm{i}^2 E(\chi_\mathrm{i})}} + K(\chi_\mathrm{e}) \sqrt[3]{\frac{\Sigma\rho_\mathrm{e}}{\chi_\mathrm{e}{}^2 E(\chi_\mathrm{e})}}\right] Q^{\frac{2}{3}} \tag{38}$$

This leads to Formula (39) for load-deflection:

$$Q = c_\mathrm{P}\, \delta^{3/2} \tag{39}$$

with the spring constant, c_P given Formula (40):

$$c_P = 1{,}48 \frac{E_\mathrm{st}}{1-\nu_\mathrm{Est}{}^2} \left[K(\chi_\mathrm{i}) \sqrt[3]{\frac{\Sigma\rho_\mathrm{i}}{\chi_\mathrm{i}^2 E(\chi_\mathrm{i})}} + K(\chi_\mathrm{e}) \sqrt[3]{\frac{\Sigma\rho_\mathrm{e}}{\chi_\mathrm{e}{}^2 E(\chi_\mathrm{e})}}\right]^{-\frac{3}{2}} \tag{40}$$

For hybrid ball bearings, i.e. bearings with raceways made of steel and balls made of silicon-nitride (Si_3N_4), the spring constant is calculated by Formula (41):

$$c_P = 1{,}48 \frac{2}{\dfrac{1-\nu_\mathrm{Est}{}^2}{E_\mathrm{st}} + \dfrac{1-\nu_\mathrm{Ece}{}^2}{E_\mathrm{ce}}} \left[K(\chi_\mathrm{i}) \sqrt[3]{\frac{\Sigma\rho_\mathrm{i}}{\chi_\mathrm{i}^2 E(\chi_\mathrm{i})}} + K(\chi_\mathrm{e}) \sqrt[3]{\frac{\Sigma\rho_\mathrm{e}}{\chi_\mathrm{e}{}^2 E(\chi_\mathrm{e})}}\right]^{-\frac{3}{2}} \tag{41}$$

The calculation of the Hertzian parameters used in the formulae above is described in Annex E.

6.3 Elastic deflection of line contact

6.3.1 General

The formulae for line contact describe the total elastic deflection of both inner ring and outer ring contact of a roller. The elastic deflection of one single inner ring or outer ring contact can be approximated as one half of the deflection given by the formulae below. Thus, the spring constant for a single sided line contact can be approximated as the spring constant for the complete rolling element, multiplied by $2^{10/9}$.

6.3.2 Cylindrical rollers

According to Reference [7], the total elastic deflection of a cylindrical roller can be described by Formula (42):

$$Q = c_L \ \delta_L^{10/9} \tag{42}$$

with the spring constant, c_L, for contacting parts made of steel given by Formula (43):

$$c_L = 35\,948\, L_{we}^{8/9} \tag{43}$$

For hybrid roller bearings, i.e. bearings with raceways made of steel and rollers made of silicon-nitride (Si_3N_4), the spring constant is calculated by Formula (44):

$$c_L = 42\,119\, L_{we}^{8/9} \tag{44}$$

The load-deflection formula, giving the load on lamina k of roller j, $q_{j,k}$, is the following Formula (45):

$$q_{j,k} = c_s \ \delta_{j,k}^{10/9} \tag{45}$$

with the spring constant, c_s, for bearings made of steel, given by Formula (46):

$$c_s = \frac{c_L}{n_s} \tag{46}$$

NOTE A laminum model for cylindrical roller bearings is depicted in Figure A.2.

6.3.3 Tapered rollers

The elastic deflection, perpendicular to the axis of a tapered roller (see Figure 2), can be described by Formula (47):

$$Q = c_T \ \delta_L^{10/9} \tag{47}$$

with the spring constant, c_T, for contacting parts made of steel, given by Formula (48):

$$c_T = 35\,948\,(L_{we} \cos\beta)^{8/9} \tag{48}$$

The load-deflection formula, giving the load on lamina k of roller j, $q_{j,k}$, is given by Formula (49):

$$q_{j,k} = c_s \ \delta_{j,k}^{10/9} \tag{49}$$

with the spring constant, c_s, given by Formula (50):

$$c_s = \frac{c_T}{n_s} \tag{50}$$

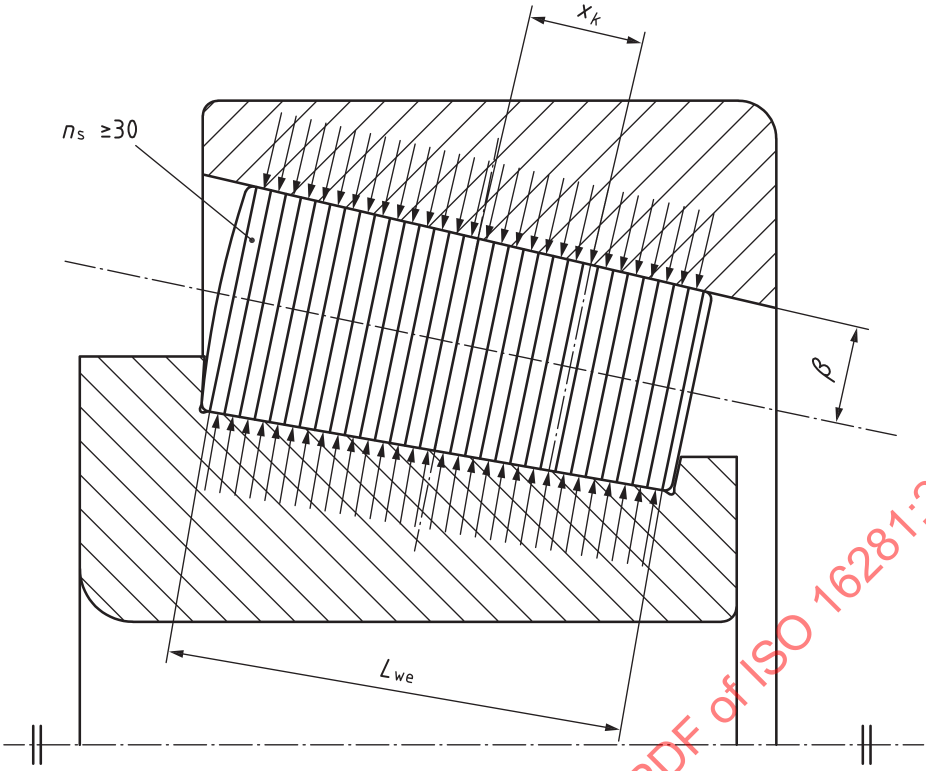


Figure 2 — Laminum model for tapered roller

6.3.4 Rollers of spherical roller bearings

For the calculation of elastic deflection, a roller of a spherical roller bearing can be approximated as a series of cylindrical roller laminae, see Figure 3. Thus, the elastic deflection of such a roller, perpendicular to the roller axis, can be described by Formula (51):

$$Q = c_L \, \delta_L^{10/9} \tag{51}$$

with the spring constant, c_L, for contacting parts made of steel, given by Formulae (52) and (53):

$$c_L = 35\,948 \, L_{we}^{8/9} \tag{52}$$

and

$$c_s = \frac{c_L}{n_s} \tag{53}$$

The load-deflection formula, giving the load on lamina k of roller j, $q_{j,k}$ is the following Formula (54):

$$q_{j,k} = c_s \, \delta_{j,k}^{10/9} \tag{54}$$

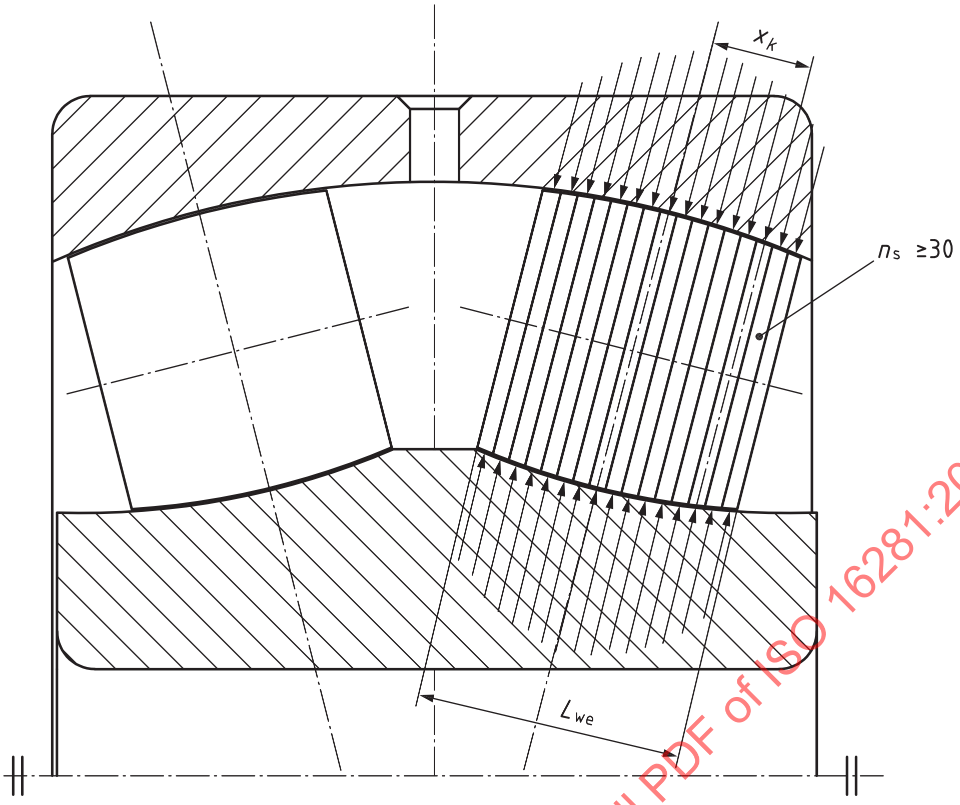


Figure 3 — Laminum model for rollers of a spherical roller bearing

Annex A
(informative)

Calculation of the bearing internal load distribution

A.1 General

This annex describes the analysis of the internal load distribution for single-row radial and thrust ball and roller bearings under radial and axial load, taking into account radial clearance and tilt. Calculation methods concerning the analysis of bearings of different geometry or for more complex load cases can be derived from the formulae given in this document.

The bearing internal load distribution is calculated for a static equilibrium; dynamic effects like centrifugal and gyroscopic forces are considered insignificant. This assumption is generally valid for low and moderate speeds, i.e. typically at $n\,D_{\mathrm{pw}}$ below 1 million mm/min. The consideration of centrifugal and gyroscopic forces is addressed in A.4.

A.2 Bearing internal load distribution for ball bearings

A.2.1 General

For a radial ball bearing with diametrically measured radial operating clearance, $G_{\mathrm{r\,op}}$ having an initial contact angle is given by Formula (A.1):

$$\alpha_0 = \arccos\left(1 - \frac{G_{\mathrm{r\,op}}}{2\,A}\right) \tag{A.1}$$

and having the displacements δ_{r}, δ_{a} and total misalignment ψ as depicted in Figure A.1 b), the total elastic deflection of the ball j, δ_j, is given by Formula (A.2):

$$\delta_j = \left[\sqrt{\left(A\cos\alpha_0 + \delta_{\mathrm{r}}\cos\varphi_j\right)^2 + \left(A\sin\alpha_0 + \delta_{\mathrm{a}} - R_{\mathrm{i}}\sin\psi\cos\varphi_j\right)^2} - A\right] \tag{A.2}$$

The right-hand side of Formula (A.2) is set to zero if it is negative.

NOTE The initial contact angle, α_0, is generally not identical to the nominal contact angle, α, in ISO 281.

In Formula (A.2), A is the distance between the curvature centres of the raceway groove radii, r_i and r_e, see Formula (A.3) and Figure A.1 a).

$$A = r_i + r_e - D_w \tag{A.3}$$

The distance between the centre of curvature of the inner raceway groove and the axis of rotation, R_i, is given by Formula (A.4):

$$R_i = \frac{D_{pw}}{2} + \left(r_i - \frac{D_w}{2} \right) \cos \alpha_0 \tag{A.4}$$

The contact loads can be calculated from the load-deflection relationship for point contacts using Formula (39). These contact loads act in the direction of the operating contact angle of the ball, α_j, as per Formula (A.5):

$$\alpha_j = \arctan \left(\frac{A \sin \alpha_0 + \delta_a - R_i \sin \psi \cos \varphi_j}{A \cos \alpha_0 + \delta_r \cos \varphi_j} \right) \tag{A.5}$$

A.2.2 Static equilibrium

The static equilibrium conditions for the external forces and moment acting on the bearing rings and the reaction forces of the balls yield the equation system, which can be solved by iteration, of the sum of all forces given by Formulae (A.6) and (A.7):

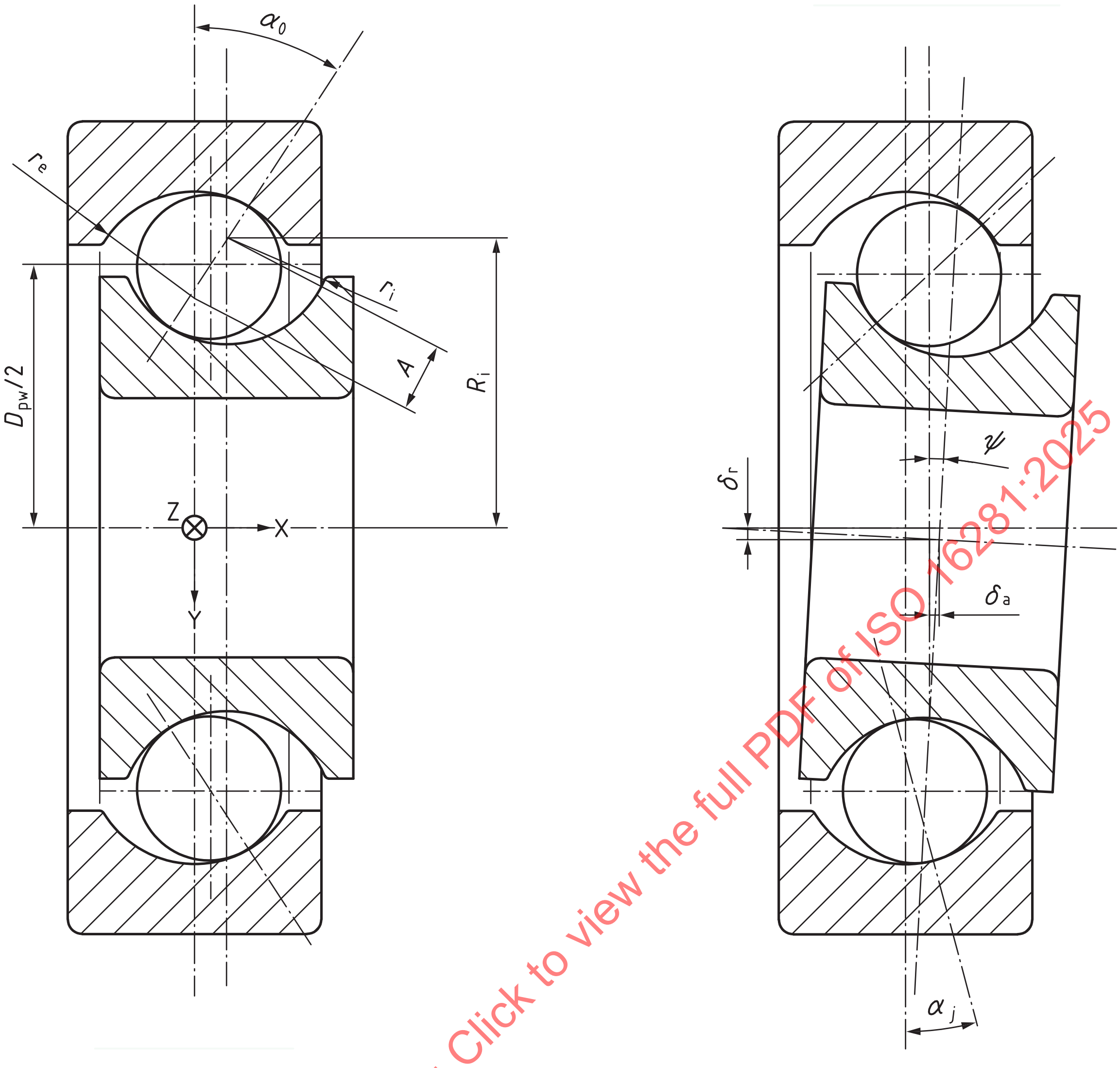
$$F_r - c_P \sum_{j=1}^{Z} \delta_j^{3/2} \cos \alpha_j \cos \varphi_j = 0 \tag{A.6}$$

$$F_a - c_P \sum_{j=1}^{Z} \delta_j^{3/2} \sin \alpha_j = 0 \tag{A.7}$$

and the sum of all moments is calculated by Formula (A.8):

$$M_z - \frac{D_{pw}}{2} c_P \sum_{j=1}^{Z} \delta_j^{3/2} \sin \alpha_j \cos \varphi_j = 0 \tag{A.8}$$

NOTE The formula set is based on a right-handed coordinate system, where the X-axis coincides with the nominal rotation axis of the bearing. In all figures depicted in this document, the Y-axis is oriented vertically towards the bottom of the page.



a) Initial contact angle

b) Displacement, misalignment of inner ring and operating contact angle

NOTE Bearing clearance is nullified by inner ring axial displacement a) and under combined axial and radial load on the inner ring b). Displacements are exaggerated.

Figure A.1 — Auxiliary geometry parameters for ball bearings

A.3 Bearing internal load distribution for roller bearings

A.3.1 Cylindrical roller bearings

For the case where the raceways are straight (i.e. unprofiled), the elastic deflection of a misaligned roller can be described by a lamina model, as depicted in Figure A.2.

For a relative radial displacement, δ_r, of the inner ring, the elastic deflection of the roller j, δ_j, is calculated by Formula (A.9):

$$\delta_j = \delta_r \cos\varphi_j - \frac{G_{r\,op}}{2} \quad (A.9)$$

The total misalignment angle between the raceways in the plane, shown in Figure A.3, of the roller j, ψ_j, is given by Formula (A.10):

$$\psi_j = \arctan\left(\tan\psi \cos\varphi_j\right) \quad (A.10)$$

This leads to the elastic deflection of the lamina k of a straight (i.e. unprofiled) roller j, given by Formula (A.11):

$$\delta_{j,k} = \delta_j + x_k \tan\psi_j \quad (A.11)$$

Further, the profile depth is subtracted from the deflection, giving the total deflection $\delta_{j,k}$:

$$\delta_{j,k} = \delta_j + x_k \tan\psi_j - 2P(x_k) \quad (A.12)$$

The right-hand side of Formula (A.12) is set to zero if it is negative.

NOTE 1 The assumptions in Formulae (A.11) and (A.12) are not exactly true, when axial rib loads act on the roller end face. However, for typical load cases, the consideration of axial rib loads for cylindrical roller bearings is not necessary.

NOTE 2 In case of a profiled raceway, Formulae (A.9) to (A.12) can be modified accordingly.

Reference profiles for typical roller bearing designs are given in Annex B. The use of actual profile data is preferred.

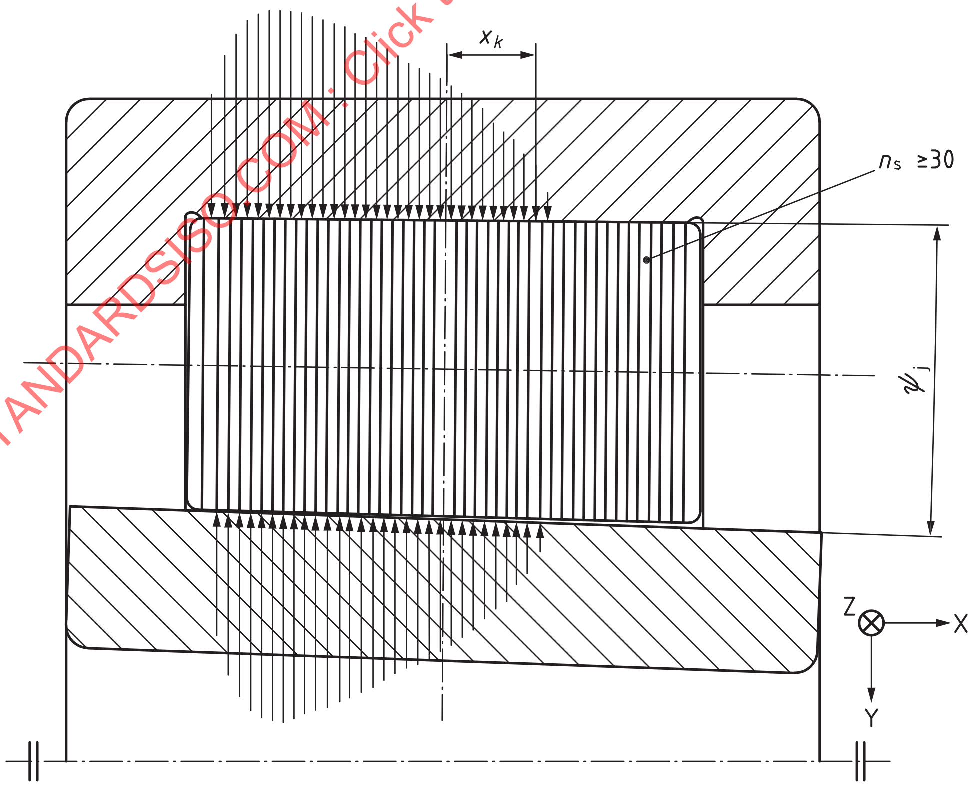


Figure A.2 — Lamina model for roller bearing

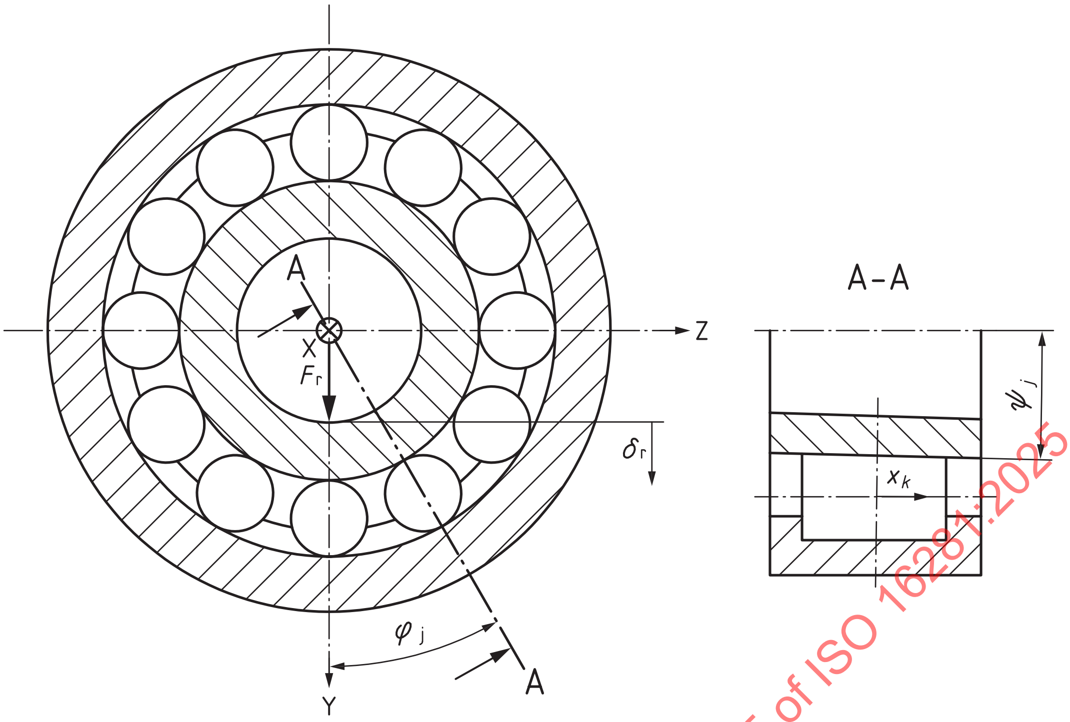


Figure A.3 — Misaligned roller bearing

The static equilibrium conditions for the external forces and moment acting on the bearing rings and the reaction forces of the rollers yield the equation system, which can be solved by iteration, of the sum of all forces as given in Formula (A.13):

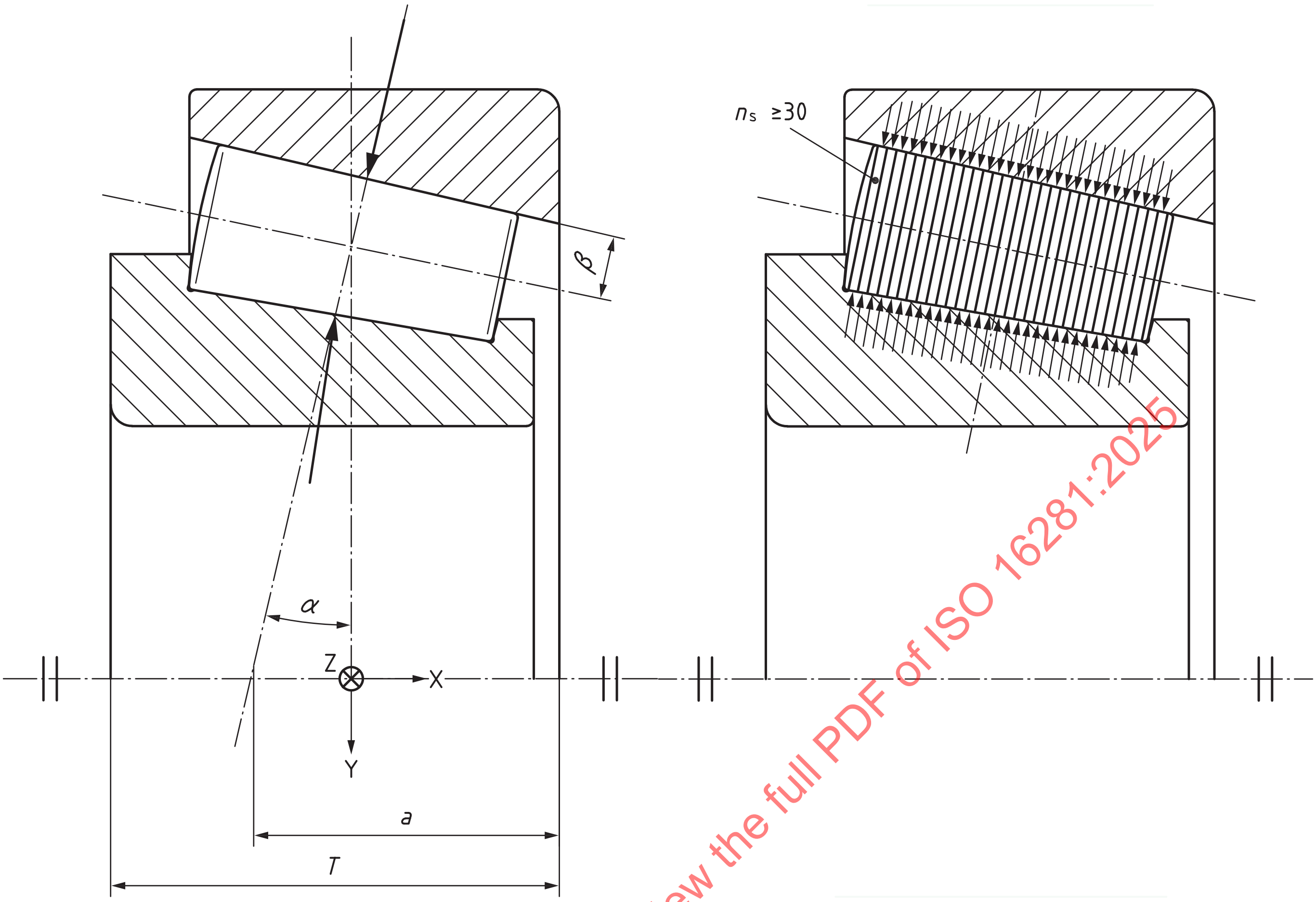
$$F_r - \frac{c_L}{n_s} \sum_{j=1}^{Z} \left(\cos\varphi_j \sum_{k=1}^{n_s} \delta_{j,k}^{\frac{10}{9}} \right) = 0 \tag{A.13}$$

and the sum of all moments as given in Formula (A.14):

$$M_z - \frac{c_L}{n_s} \sum_{j=1}^{Z} \left(\cos\varphi_j \sum_{k=1}^{n_s} x_k \, \delta_{j,k}^{\frac{10}{9}} \right) = 0 \tag{A.14}$$

A.3.2 Tapered roller bearings

For tapered roller bearings, the contact forces are acting perpendicular to each raceway, see Figure A.4. The axial force on the rib can be calculated from the half cone angle of a tapered roller, β . For typical half cone angles, this axial component is rather small compared to the cumulated loads on the raceway and can be neglected in the equilibrium analysis.



a) **Rolling element load**

b) **Lamina loads**

Figure A.4 — Resulting forces on a roller in a tapered roller bearing

For a tapered roller, the total elastic deflection of the lamina k of the roller j, perpendicular to the roller axis, $\delta_{j,k}$, is:

$$\delta_{j,k} = \left(\left[\delta_\mathrm{a} - \left(\frac{D_\mathrm{pw} + D_\mathrm{we} \cos\alpha}{2} - x_k \sin\alpha \right) \sin\psi \cos\varphi_j \right] \sin\alpha + \left\{ \left[\delta_\mathrm{r} + \left(\frac{T}{2} - a + \frac{D_\mathrm{pw}/\cos\alpha + D_\mathrm{we}}{2} \sin\alpha + x_k \cos\alpha \right) \sin\psi \right] \cos\varphi_j \right\} \cos\alpha \right) / \cos\beta - 2P(x_k) \quad \text{(A.15)}$$

The right-hand side of Formula (A.15) is set to zero if it is negative. Reference profiles for typical roller bearing designs are given in Annex B. The use of actual profile data is preferred.

The static equilibrium conditions for the external forces and moment acting on the bearing rings and the reaction forces of the rollers yield the equation system, which can be solved by iteration, of the sum of all forces given by Formulae (A.16) and (A.17):

$$F_\mathrm{r} - \frac{c_\mathrm{s} \cos\alpha}{\cos\beta} \sum_{j=1}^{Z} \left(\cos\varphi_j \sum_{k=1}^{n_\mathrm{s}} \delta_{j,k}^{\frac{10}{9}} \right) = 0 \quad \text{(A.16)}$$

$$F_a - \frac{c_s \sin\alpha}{\cos\beta} \sum_{j=1}^{Z} \left(\sum_{k=1}^{n_s} \delta_{j,k}^{\frac{10}{9}} \right) = 0 \tag{A.17}$$

and the sum of all moments given by Formula (A.18):

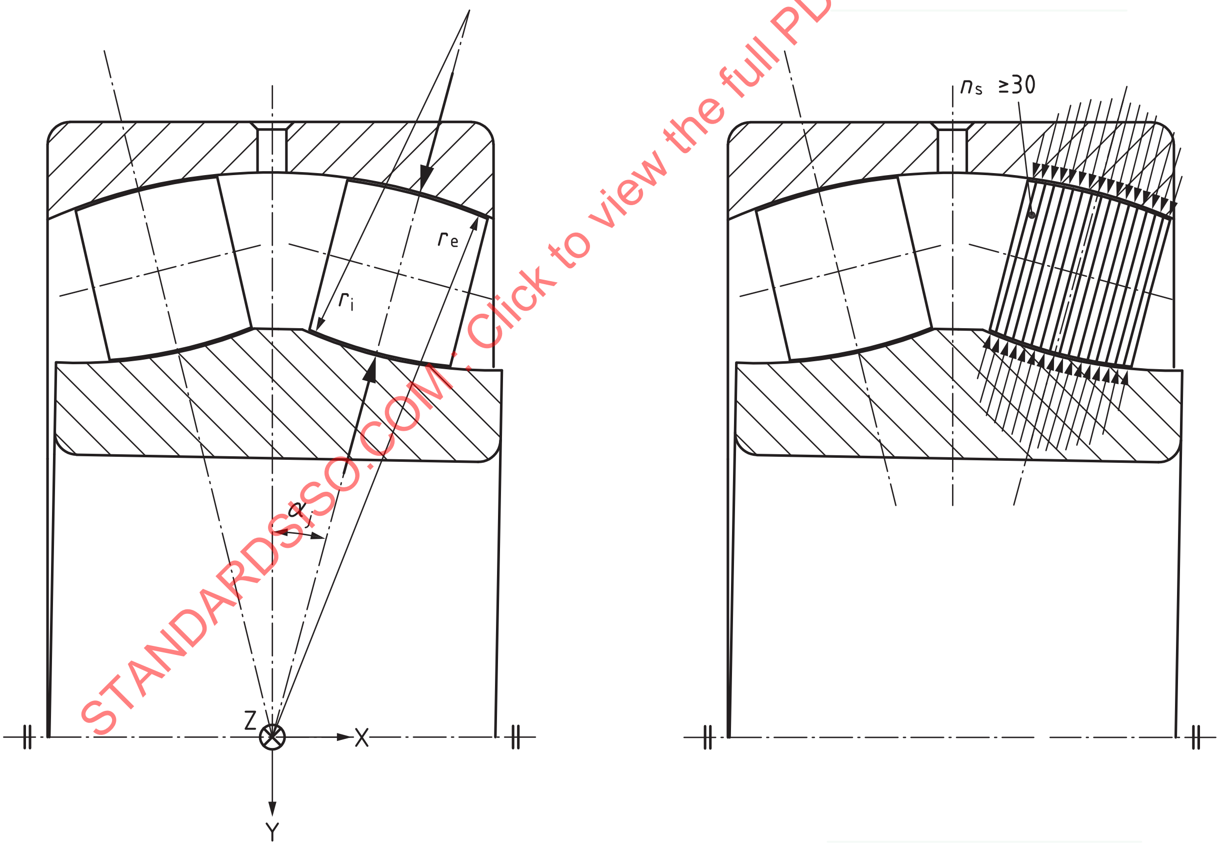
$$M_z - \frac{c_s \cos\alpha}{\cos\beta} \sum_{j=1}^{Z} \left[\cos\varphi_j \sum_{k=1}^{n_s} \left(\frac{T}{2} - a + \frac{x_k}{\cos\alpha} \right) \delta_{j,k}^{\frac{10}{9}} \right] = 0 \tag{A.18}$$

A.3.3 Spherical roller bearings

Spherical roller bearings can be used under misalignment condition between inner ring and outer ring, but cannot support moment loads.

For spherical roller bearings, the load distribution can be calculated from a combination of the calculation approaches for ball bearings and for roller bearings, as depicted in Figure A.5.

Typically, the operating contact angle α_j of roller j is calculated from the positions of centre of curvature for inner raceway and outer raceway, similar to the operating contact angle calculation for ball bearings. The contact forces between roller and raceways are then calculated using the roller contact lamina model per 6.3.4, where contact forces are assumed to be perpendicular to the roller axis. This approach is valid, since generally the contact force components parallel to the roller axis cancel out.



a) Rolling element load **b) Lamina loads**

NOTE Radii are not to scale. Exemplarily depicted is a spherical roller bearing that does not have ribs or guide rings.

Figure A.5 — Resulting forces on a roller in a spherical roller bearing

The static equilibrium conditions for the external forces acting on the bearing rings and the reaction forces of the rollers on both rows yield the equation system of the sum of all forces, which can be solved by iteration given by Formulae (A.19) and (A.20):

$$F_r - c_s \sum_{i=1}^{2} \left[\sum_{j=1}^{Z} \left(\cos\alpha_{i,j} \cos\varphi_j \sum_{k=1}^{n_s} \delta_{i,j,k}^{\frac{10}{9}} \right) \right] = 0 \tag{A.19}$$

$$F_a - c_s \sum_{i=1}^{2} \left[\sum_{j=1}^{Z} \left(\sin\alpha_{i,j} \sum_{k=1}^{n_s} \delta_{i,j,k}^{\frac{10}{9}} \right) \right] = 0 \tag{A.20}$$

NOTE The operating contact angle $\alpha_{i,j}$ can be positive or negative.

A.4 Consideration of centrifugal and gyroscopic forces

Generally, for standard bearings operating below the thermal reference speed per ISO 15312[8], centrifugal and gyroscopic forces have little effect on the bearing internal load distribution, and thus can be neglected.

However, at a very high speed, typically at $n\,D_{pw}$ over 1 million mm/min, the rolling element load distribution can be significantly altered by the effect of centrifugal forces acting on the rolling element.

In this case, the effect of centrifugal and gyroscopic forces on the rolling element should be considered by calculating the static equilibrium for each rolling element. Typically, this is done simultaneously with the calculation of static equilibrium for the bearing ring, which leads to a significant increase of degrees of freedom for the calculation model.

Annex B
(informative)

Reference geometries

B.1 General

From the calculation methods for radial ball bearings and radial roller bearings presented in this document, calculation methods for bearings with more complex designs can be derived. Reference geometries for the most common bearing design types are defined in this annex.

The geometry data given here are approximate values. Actual bearing designs, based on the expertise of the manufacturer, can deviate from these reference geometries. The use of actual bearing geometries and profiles in the analysis is preferred.

For roller bearings, in addition to the roller profile, any undercuts or additional raceway profile should be considered in the sum profile when calculating the actual contact stress.

B.2 Deep groove ball bearings, angular contact ball bearings and separable ball bearings (magneto bearings)

Cross-sectional raceway groove radius of outer ring, r_e, is given by Formula (B.1):

$$r_e = 0{,}53\, D_w \tag{B.1}$$

Cross-sectional raceway groove radius of inner ring, r_i, is given by Formula (B.2):

$$r_i = 0{,}52\, D_w \tag{B.2}$$

B.3 Self-aligning ball bearings

Cross-sectional spherical raceway curvature radius of outer ring, r_e, is given by Formula (B.3):

$$r_e = 0{,}5\left(1 + \frac{1}{\gamma}\right) D_w \tag{B.3}$$

Cross-sectional raceway groove radius of inner ring, r_i, is given by Formula (B.4):

$$r_i = 0{,}53\, D_w \tag{B.4}$$

B.4 Thrust ball bearings and thrust angular contact ball bearings

Cross-sectional raceway groove radius of housing washer, r_e, is given by Formula (B.5):

$$r_e = 0{,}54\, D_w \tag{B.5}$$

Cross-sectional raceway groove radius of shaft washer, r_i, is given by Formula (B.6):

$$r_i = 0{,}54\, D_w \tag{B.6}$$

B.5 Cylindrical roller bearings and needle roller bearings

For rollers having a length $L_{we} \leq 2{,}5\,D_{we}$:

$$P(x_k) = 0{,}000\,35\,D_{we}\ \ln\left[\frac{1}{1-\left(\frac{2\,x_k}{L_{we}}\right)^2}\right] \quad \text{(B.7)}$$

For rollers having a length $L_{we} > 2{,}5\,D_{we}$, a stepwise defined profile function applies:

$$P(x_k) = 0 \qquad \text{for} |x_k| \leq \frac{L_{we} - 2{,}5\,D_{we}}{2} \quad \text{(B.8)}$$

$$P(x_k) = 0{,}000\,50\,D_{we}\ \ln\left[\frac{1}{1-\left(\frac{[2|x_k| - (L_{we} - 2{,}5\,D_{we})]}{2{,}5\,D_{we}}\right)^2}\right] \qquad \text{for} |x_k| > \frac{L_{we} - 2{,}5\,D_{we}}{2} \quad \text{(B.9)}$$

B.6 Tapered roller bearings

Profile of tapered rollers, perpendicular to the axis of the roller is given by Formula (B.10):

$$P(x_k) = \frac{0{,}000\,45}{\cos\beta}\,D_{we}\ \ln\left[\frac{1}{1-\left(\frac{2\,x_k}{L_{we}}\right)^2}\right] \quad \text{(B.10)}$$

NOTE For typical taper angles of $\beta < 8{,}5°$, the rounded value of $0{,}000\,45 / \cos\beta \approx 0{,}000\,45$ can be applied.

B.7 Spherical roller bearings

Cross-sectional spherical raceway curvature radius of outer ring, r_e, is given by Formula (B.11):

$$r_e = \frac{D_{pw}}{2\cos\alpha} + \frac{D_{we}}{2} \quad \text{(B.11)}$$

Cross-sectional raceway curvature radius of inner ring, r_i, is given by Formula (B.12):

$$r_i = r_e \quad \text{(B.12)}$$

Convex curvature radius of spherical rollers, R_p, is given by Formula (B.13):

$$R_p = 0{,}97\,r_e \quad \text{(B.13)}$$

B.8 Thrust cylindrical roller bearings and thrust needle roller bearings

The roller profile is defined in Formulae (B.7), (B.8) and (B.9).