TECHNICAL REPORT

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Explanatory notes on ISO 76

Notes explicatives sur l'ISO 76

Reference number
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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).

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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 second edition cancels and replaces the first edition (ISO 10657:1991), which has been technically revised.

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

— New subclause 0.4 and 0.5 included with explanations concerning the 2006 edition of ISO 76:2006 and ISO 76/Amd.1:2017;

— Inclusion of Clause 3 for symbols;

— Table 16 and Table 18 amended according to additional values in ISO 76:2006 (values of X_0 and Y_0 at contact angles 5° and 10° of angular contact ball bearings).

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

0.1 ISO/R 76:1958

ISO/R 76, *Ball and Roller Bearings — Methods of Evaluating Static Load Ratings,* was drawn up by Technical Committee ISO/TC 4, *Ball and Roller Bearings*.

ISO/R 76 was based on the studies of A. Palmgren et al[2],[3]. The basic static load ratings were defined to correspond to a total permanent deformation of rolling element and raceway at the most heavily stressed rolling element/raceway contact of 0,000 1 of the rolling element diameter. Then the standard values confined to the basic static load ratings for special inner design rolling bearings were laid down.

ISO/R 76:1958 was approved by 28 (out of a total of 38) member bodies and was then submitted to the ISO Council, which decided, in December 1958, to accept it as an ISO Recommendation.

0.2 ISO 76:1978

ISO/TC 4 decided to include the revision of ISO/R 76 in its programme of work and ISO/TC 4/SC 8 secretariat was requested to prepare a draft proposal. As a result, the secretariat submitted a draft proposal[3] in January 1976.

The draft proposal was accepted by 6 of the 8 members of ISO/TC 4/SC 8. Of the remaining two, Japan preferred further study and USA, its counter proposal, document ISO/TC 4/SC 8 N 64[4]. The draft was then submitted to the ISO Central Secretariat. After the draft had been approved by the ISO member bodies, the ISO Council decided in June 1978 to accept it as an International Standard.

ISO 76:1978 adopted the SI unit newton and was revised in total, but without essential changes of substance. However, values of X_0 and Y_0 for the nominal contact angles 15° and 45° for angular contact groove ball bearings were added to the table to calculate the static equivalent radial loads of radial ball bearings (see ISO 76:1978, Table 2).

0.3 ISO 76:1987

During the revision of ISO/R 76:1958, USA had in 1975 submitted a counter proposal[4] for the basic static load ratings based on a calculated contact stress.

The secretariat requested a vote on the revision of the static load ratings based on a contact stress level in January 1978 and afterward circulated the voted results in June 1978, and the item No. of revision work had become No. 157 of the programme of work of TC 4.

ISO/TC 4/SC 8, considering the proposals made in the documents TC 4/SC 8 N 75[5] and TC 4 N 865[6], as well as the comments made by TC 4 members and that several SC 8 members expressed a need for updating ISO 76, agreed to continue its study taking into account the possibility of using either permanent deformation or stress level as a basis for static load ratings, and ISO/TC 4/SC 8 requested its secretariat to prepare a new draft. The new draft was intended to be prepared with the principles and formulae of the document TC 4/SC 8 N 75, and to include levels of contact stress for various rolling element contact stated to be generally corresponding to a permanent deformation of 0,000 1 of the rolling element diameter at the centre of the most heavily stressed rolling element/raceway contact. For roller bearings a stress level of 4 000 MPa was agreed and then ISO/TC 4/SC 8 agreed, by a majority vote, that static load ratings should correspond to calculated contact stresses of

4 000 MPa for roller bearings,

4 600 MPa for self-aligning ball bearings, and

4 200 MPa for all other ball bearings to which the standard applies.

For these calculated contact stresses, a total permanent deformation occurs at the centre of the most heavily stressed rolling element/raceway contact, and its deformation is approximately 0,000 1 of the rolling element diameter.

ISO 76 was submitted to the ISO Central Secretariat in 1985, and after it had been approved by the ISO members, the ISO Council decided in February 1987 to accept it as an International Standard.

Furthermore, ISO/TC 4/SC 8 decided that supplementary background information, regarding the derivation of formulae and factors given in ISO 76, should be published as a Technical Report. This Technical report was published as ISO/TR 10657:1991.

An Amendment to ISO 76:1987 that explains the discontinuities in load ratings between radial- and axial bearings was published as ISO 76:1987/Amd.1:1999.

0.4 ISO 76:2006

A systematic review of ISO 76:1987 was agreed in 2003, based on the prior held balloting process and documents TC 4/SC 8 N 233 and N 235.

ISO 76:2006 includes editorial adaptations and updates as well as an extension by the static safety factor S_0. Furthermore, ISO 76:1987/Amd.1:1999 was integrated and became the informative Annex A "Discontinuities in the calculation of basic static load ratings".

0.5 ISO 76:2006/Amd.1:2017

ISO 76:2006/Amd.1:2017 includes the following items:

— graphs for the factors f_0, X_0 and Y_0 taken from draft ÖNORM M 6320 to be included in an informative annex;

— formulae for the calculation of the load rating factor f_0 from ISO/TR 10657 to be introduced in the normative part of the standard;

— the tables for the load rating factor f_0 will stay in the normative part of the standard, however a sentence will be introduced stating that the results obtained from formulae are preferred.

Explanatory notes on ISO 76

1 Scope

This document specifies supplementary background information regarding the derivation of formulae and factors given in ISO 76:2006.

2 Normative references

There are no normative references in this document.

3 Terms, definitions and symbols

3.1 Terms and definitions

No terms and definitions are listed in this document.

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/

3.2 Symbols

C_{0a} basic static axial load rating, in newtons

C_{0r} basic static radial load rating, in newtons

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

D_w nominal ball diameter, in millimetres

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

E modulus of elasticity (Young' s modulus), in megapascals

E_1, E_2 modulus of elasticity of body 1 (rolling element) and of body 2 (raceway), in megapascals

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

E_0 $E/(1 - \nu^2)$

F_a bearing axial load (axial component of actual bearing load), in newtons

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

$F(\rho)$ relative curvature difference

$J_a(\varepsilon)$ axial load integral

$J_r(\varepsilon)$ radial load integral

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

L_{we} length of roller applicable in the calculation of load ratings, in millimetres

P_{0a} theoretical static equivalent axial load for thrust bearing, general speaking, called static equivalent axial load, in newtons

P_{0r} theoretical static equivalent radial load for radial bearing, general speaking, called static equivalent radial load, in newtons

Q normal force between rolling element and raceway, in newtons

Q_{max} maximum normal force between rolling element and raceway, in newtons

S Stribeck number

X_0 static radial load factor

Y_0 static axial load factor

Z number of balls carrying load in one direction, number of balls or rollers per row, or number of rolling elements per row

a semi-major axis of the projected contact ellipse, semilength of the contact surface

b semi-minor axis of the projected contact ellipse, semi-width of the contact surface

c compression constant, in 1/megapascals$^{2/3}$

f osculation = r/D_w

f_e osculation at the outer ring = r_e/D_w

f_i osculation at the inner ring = r_i/D_w

f_0 factor which depends on the geometry of the bearing components and on applicable stress level

i number of rows of balls or rollers in a bearing

k_0 load distribution parameter

r curvature radius of a raceway cross-section, in millimetres

r_e outer ring groove radius, in millimetres

r_i inner ring groove radius, in millimetres

t exponent in load–deflection formula

x distance in direction of the semi-major axis, in millimetres

y distance in direction of the semi-minor axis, in millimetres

α nominal contact angle, in degrees

α' actual contact angle, in degrees

γ	auxiliary parameter, $\gamma = D_w \cos\alpha / D_{pw}$	for ball bearings with $\alpha \neq 90°$
	$\gamma = D_w / D_{pw}$	for ball bearings with $\alpha = 90°$
	$\gamma = D_{we} \cos\alpha / D_{pw}$	for roller bearings with $\alpha \neq 90°$
	$\gamma = D_{we} / D_{pw}$	for roller bearings with $\alpha = 90°$
ε	parameter indicating the width of the loaded zone	
κ	ratio of semi-major to semi-minor axis = a/b	
ν	Poisson's ratio	
ν_1	Poisson's ratio of body 1 (rolling element)	
ν_2	Poisson's ratio of body 2 (raceway)	
$\Sigma\rho$	curvature sum	
ρ_{11}, ρ_{12}	principal curvature of body 1 (rolling element)	
ρ_{21}, ρ_{22}	principal curvature of body 2 (raceway)	
σ	calculated contact stress, in megapascals	
σ_{max}	maximum calculated contact stress, in megapascals	
ϕ	auxiliary angle, in radians	
ψ_0	one half of the loaded arc	

4 Basic static load ratings

4.1 General

4.1.1 Basic formula for point contact

The relationship between a calculated contact stress and a rolling element load within an elliptical contact area is given in Reference [8] as Formula (1),

$$\sigma = \frac{3Q}{2\pi ab}\left[1-\left(\frac{x}{a}\right)^2-\left(\frac{y}{b}\right)^2\right]^{1/2} \tag{1}$$

It is concluded that the maximum calculated contact stress (σ_{max}) occurs at the point of $x = 0$ and $y = 0$,

$$\sigma_{max} = \frac{3Q}{2\pi ab} \text{ or } Q = \frac{2\pi ab}{3}\sigma_{max} \tag{2}$$

According to the Hertz's theory,

$$a = \left(\frac{2\kappa^2 E(\kappa)}{\pi}\right)^{1/3}\left[\frac{3Q}{2\Sigma\rho}\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)\right]^{1/3} \tag{3}$$

$$b = \left(\frac{2E(\kappa)}{\pi\kappa}\right)^{1/3}\left[\frac{3Q}{2\Sigma\rho}\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)\right]^{1/3} \tag{4}$$

where

κ $= a/b$

$E(\kappa)$ $= \int_0^{\pi/2} \left[1 - \left(1 - \frac{1}{\kappa^2} \right) \sin^2 \phi \right]^{1/2} \mathrm{d}\phi$

$\Sigma\rho$ $= \rho_{11} + \rho_{12} + \rho_{21} + \rho_{22}$

$\rho_{11} = \rho_{12}$ $= \frac{2}{D_\mathrm{w}}$

Substituting Formula (3) and Formula (4) into Formula (2) for the case of $E_1 = E_2 = E$ and $\nu_1 = \nu_2 = \nu$,

$$Q = \sigma_{\max}^3 \frac{32\pi}{3E_0^2} \kappa \left(\frac{E(\kappa)}{\Sigma\rho} \right)^2 \tag{5}$$

and

$$1 - \frac{2}{\kappa^2 - 1} \left(\frac{K(\kappa)}{E(\kappa)} - 1 \right) - F(\rho) = 0 \tag{6}$$

where

E_0 $= \frac{E}{1 - \nu^2}$

E $= 2{,}07 \times 10^5$ MPa

ν $= 0{,}3$

$K(\kappa)$ $= \int_0^{\pi/2} \left[1 - \left(1 - \frac{1}{\kappa^2} \right) \sin^2 \phi \right]^{-1/2} \mathrm{d}\phi$

$F(\rho) = \frac{\rho_{11} - \rho_{12} + \rho_{21} - \rho_{22}}{\rho_{11} + \rho_{12} + \rho_{21} + \rho_{22}}$

Consequently, from Formula (5),

$$Q = 6{,}476\,206\,5 \times 10^{-10} \kappa \left(\frac{E(\kappa)}{\Sigma\rho} \right)^2 \sigma_{\max}^3 \tag{7}$$

4.1.2 Basic formula for line contact

The relationship between a calculated contact stress and a rolling element load for a line contact is given in Reference [9] as follows,

$$\sigma = \frac{2Q}{\pi L_{we} b}\left[1-\left(\frac{y}{b}\right)^2\right]^{1/2} \tag{8}$$

It is concluded that the maximum calculated contact stress (σ_{max}) from Formula (8) occurs at the line of $y = 0$,

$$\sigma_{max} = \frac{2Q}{\pi L_{we}\, b} \text{ or } Q = \frac{\pi L_{we}\, b}{2}\sigma_{max} \tag{9}$$

And also b is given by the following formula,

$$b = \left[\frac{4Q}{\pi L_{we} \Sigma\rho}\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right)\right]^{1/2} \tag{10}$$

where

$\Sigma\rho = \rho_{11} + \rho_{12} + \rho_{21} + \rho_{22}$

$\rho_{11} = \dfrac{2}{D_{we}}$

$\rho_{21} = \pm\dfrac{2}{D_{we}}\dfrac{\gamma}{1\mp\gamma}$; the upper sign applies to inner ring contact and the lower to outer ring contact;

$\rho_{12} = 0$

$\rho_{22} = 0$

$\gamma = \dfrac{D_{we}\cos\alpha}{D_{pw}}$

Substituting Formula (10) into Formula (9) for the case of $E_1 = E_2 = E$ and $\nu_1 = \nu_2 = \nu$,

$$Q = 2\pi\sigma_{max}^2 \frac{L_{we}}{E_0 \Sigma\rho}$$

where

$E_0 = \dfrac{E}{1-\nu^2}$

$E = 2{,}07 \times 10^5$ MPa

$\nu = 0{,}3$

Consequently,

$$Q = 2{,}762\,173\,2 \times 10^{-5}\frac{L_{we}}{\Sigma\rho}\sigma_{max}^2 \tag{11}$$

4.2 Basic static radial load rating C_{0r} for radial ball bearings

4.2.1 Radial and angular contact groove ball bearings

The curvature sum $\Sigma\rho$ and the relative curvature difference $F(\rho)$ of radial and angular contact groove ball bearings is given by the following formulae,

$$\Sigma\rho = \frac{2}{D_\mathrm{w}}\left(2 \pm \frac{\gamma}{1 \mp \gamma} - \frac{1}{2 f_{\mathrm{i(e)}}}\right) \tag{12}$$

$$F(\rho) = \frac{\pm \dfrac{\gamma}{1 \mp \gamma} + \dfrac{1}{2 f_{\mathrm{i(e)}}}}{2 \pm \dfrac{\gamma}{1 \mp \gamma} - \dfrac{1}{2 f_{\mathrm{i(e)}}}} \tag{13}$$

where

the upper sign applies to inner ring contact and the lower to outer ring contact;

$\gamma \quad = \dfrac{D_\mathrm{w} \cos\alpha}{D_\mathrm{pw}}$

$f_{\mathrm{i(e)}}$ denotes

$f_\mathrm{i} \quad = \dfrac{r_\mathrm{i}}{D_\mathrm{w}}$ for inner ring contact, and

$f_\mathrm{e} \quad = \dfrac{r_\mathrm{e}}{D_\mathrm{w}}$ for outer ring contact

Substituting Formula (12) into Formula (7),

$$Q = 6{,}476\,206\,5 \times 10^{-10}\,\kappa\left(\frac{D_\mathrm{w}}{2}\,\frac{E(\kappa)}{2 \pm \dfrac{\gamma}{1 \mp \gamma} - \dfrac{1}{2 f_{\mathrm{i(e)}}}}\right)^2 \sigma_{\max}^3 \tag{14}$$

Substituting Formula (12) and Formula (14) into Formula (15) (see Reference [10]), and furthermore exchanging Q for $Q_{\max}$, gives

$$C_{0r} = \frac{1}{S}\,Z\,Q_{\max} \cos \tag{15}$$

where S is a function of the loaded zone parameter ε. If one half of the balls are loaded then S = 4,37 applies. A common value used in general bearing calculations is S = 5, which leads to a rather conservative estimate of the maximum ball load.

$$C_{0r} = 0{,}2\,Z\,Q_{\max} \cos\alpha \tag{16}$$

Consequently,

$$C_{0r} = 0{,}2 \times 6{,}476\,206\,5 \times 10^{-10} \times (4\,000)^3 \left(\frac{\sigma_{max}}{4\,000}\right)^3 \kappa \times \frac{1}{4} \left(\frac{E(\kappa)}{2 \pm \frac{\gamma}{1 \mp \gamma} - \frac{1}{2 f_{i(e)}}}\right)^2 Z\, D_w^2 \cos\alpha$$

where the upper sign refers to the inner ring and the lower sign refers to the outer ring. Therefore, introducing the number of rows, i, of balls gives Formula (17):

$$C_{0r} = f_0\, i\, Z\, D_w^2 \cos \tag{17}$$

where f_0 is the factor which depends on the geometry of the bearing components and on applicable stress level:

$$f_0 = 2{,}072 \left(\frac{\sigma_{max}}{4\,000}\right)^3 \kappa \left(\frac{E(\kappa)}{2 \pm \frac{\gamma}{1 \mp \gamma} - \frac{1}{2 f_{i(e)}}}\right)^2 \tag{18}$$

For an inner ring with f_i = 0,52, Formula (18) becomes,

$$f_0 = 2{,}072 \left(\frac{\sigma_{max}}{4\,000}\right)^3 \kappa \left(\frac{E(\kappa)}{2 + \frac{\gamma}{1 - \gamma} - \frac{1}{1{,}04}}\right)^2 \tag{19}$$

and for an outer ring with f_e = 0,53,

$$f_0 = 2{,}072 \left(\frac{\sigma_{max}}{4\,000}\right)^3 \kappa \left(\frac{E(\kappa)}{2 - \frac{\gamma}{1 + \gamma} - \frac{1}{1{,}06}}\right)^2 \tag{20}$$

The smaller value between the f_0 values calculated from Formula (19) and Formula (20) is used in the calculation of static load ratings.

The values of factor f_0 in Table 1 of ISO 76:2006 are calculated from substituting the values for κ, $E(\kappa)$ and $\gamma = D_w \cos\alpha / D_{pw}$ shown in Table A.1, and σ_{max} = 4 200 MPa into the above formula.

These values apply to bearings with a cross-sectional raceway groove radius not larger than 0,52 D_w in radial and angular contact groove ball bearing inner rings, and 0,53 D_w in radial and angular contact groove ball bearing outer rings and self-aligning ball bearing inner rings

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

4.2.2 Self-aligning ball bearings

The curvature sum $\Sigma\rho$ of self-aligning ball bearings is given by the following formula for an outer ring:

$$\Sigma\rho = \frac{4}{D_w}\left(\frac{1}{1+\gamma}\right) \tag{21}$$

Substituting Formula (21) into Formula (7),

$$Q = 6{,}476\,206\,5 \times 10^{-10}\,\kappa\left(\frac{D_w}{4}(1+\gamma)E(\kappa)\right)^2 \sigma_{max}^3 \tag{22}$$

In general, $\kappa = a/b = 1$ for the case of contact between an outer ring raceway and balls of self-aligning ball bearings. Consequently,

$$E(\kappa) = \int_0^{\pi/2}\left[1-\left(1-\frac{1}{\kappa^2}\right)\sin^2\phi\right]^{1/2} d\phi = \int_0^{\pi/2} d\phi = \frac{\pi}{2}$$

Therefore, Formula (22) is obtained

$$Q = 6{,}476\,206\,5 \times 10^{-10}\,\kappa\left(\frac{D_w}{8}(1+\gamma)\pi\right)^2 \sigma_{max}^3 \tag{23}$$

Substituting Formula (23) into Formula (16) and moreover exchanging Q for Q_{max},

$$C_{0r} = 2{,}072\left(\frac{\sigma_{max}}{4\,000}\right)^3\left[\frac{\pi}{4}(1+\gamma)\right]^2 Z\,D_w^2 \cos\alpha$$

Introducing the number of rows of balls i yields Formula (24)

$$C_{0r} = f_0\, i\, Z\, D_w^2 \cos \tag{24}$$

where

$$f_0 = 2{,}072\left(\frac{\sigma_{max}}{4\,000}\right)^3\left[\frac{\pi}{4}(1+\gamma)\right]^2 \tag{25}$$

The values of factor f_0 in Table 1 of ISO 76:2006 are calculated from substituting σ_{max} = 4 600 MPa and values of $\gamma = D_w \cos\alpha / D_{pw}$ shown in the Table 1 of ISO 76:2006 into Formula (25).

4.3 Basic static axial load rating C_{0a} for thrust ball bearings

The curvature sum $\Sigma\rho$ and the relative curvature difference $F(\rho)$ of thrust ball bearings is given by the following formulae:

$$\Sigma\rho = \frac{2}{D_w}\left(2 \pm \frac{\gamma}{1\mp\gamma} - \frac{1}{2f}\right) \tag{26}$$

$$F(\rho)=\frac{\pm\dfrac{\gamma}{1\mp\gamma}+\dfrac{1}{2f}}{2\pm\dfrac{\gamma}{1\mp\gamma}-\dfrac{1}{2f}} \tag{27}$$

where the upper sign refers to the inner ring and the lower sign refers to the outer ring and

$f = r/D_w$

Substituting Formula (26) into Formula (7),

$$Q=6{,}476\,206\,5\times10^{-10}\,\kappa\left(\frac{D_w}{2}\,\frac{E(\kappa)}{2\pm\dfrac{\gamma}{1\mp\gamma}-\dfrac{1}{2f}}\right)^2\sigma_{max}^3 \tag{28}$$

Substituting Formula (28) into the following Formula (29),

$$C_{0a} = Z\,Q_{max}\sin\alpha \tag{29}$$

Therefore,

$$C_{0a}=10{,}362\left(\frac{\sigma_{max}}{4\,000}\right)^3\kappa\left(\frac{E(\kappa)}{2\pm\dfrac{\gamma}{1\mp\gamma}-\dfrac{1}{2f}}\right)^2 Z\,D_w^2\,\sin\alpha \tag{30}$$

The smaller value C_{0a} calculated from Formula (30) is adopted. For washers with f = 0,54, using the upper sign gives Formula (31),

$$C_{0a} = f_0\,Z\,D_w^2\,\sin \tag{31}$$

where

$$f_0=10{,}362\left(\frac{\sigma_{max}}{4\,000}\right)^3\kappa\left(\frac{E(\kappa)}{2+\dfrac{\gamma}{1-\gamma}-\dfrac{1}{1{,}08}}\right)^2 \tag{32}$$

The values of factor f_0 in Table 1 of ISO 76:2006 are calculated from substituting the values for κ, $E(\kappa)$ and $\gamma = D_w \cos\alpha/D_{pw}$ shown in Table A.2, and σ_{max} = 4 200 MPa into Formula (32).

4.4 Basic static radial load rating C_{0r} for radial roller bearings

The curvature sum $\Sigma\rho$ for radial roller bearings is given by the following formula,

$$\Sigma\rho = \frac{2}{D_{we}} \frac{1}{1 \mp \gamma} \tag{33}$$

Substituting Formula (33) into Formula (11) and adopting the smaller Q,

$$Q = 1{,}381\,086\,7 \times 10^{-5}\,(1-\gamma) L_{we}\, D_{we}\, \sigma_{max}^2 \tag{34}$$

Substituting Formula (34) into the following formula gives Formula (35),

$$C_{0r} = \frac{1}{S} Z\, Q_{max} \cos \tag{35}$$

where S is a function of the loaded zone parameter ε. If one half of the rollers are loaded then S = 4,08 applies. A common value used in general bearing calculations is S = 5, which leads to a rather conservative estimate of the maximum roller load.

$$C_{0r} = 44{,}194\,774 \left(\frac{\sigma_{max}}{4\,000} \right)^2 (1-\gamma)\, Z\, L_{we}\, D_{we} \cos\alpha$$

Consequently, adopting σ_{max} = 4 000 MPa and introducing the number of rows, i, of rollers gives Formula (36),

$$C_{0r} = 44 \left(1 - \frac{D_{we} \cos\alpha}{D_{pw}} \right) i\, Z\, L_{we}\, D_{we} \cos \tag{36}$$

NOTE The value has been rounded for use in the final document.

4.5 Basic static axial load rating C_{0a} for thrust roller bearings

The curvature sum $\Sigma\rho$ of thrust roller bearings is given by Formula (33) and Q is given by Formula (34).

Substituting Formula (33) and Formula (34) into Formula (29),

$$C_{0a} = 220{,}973\,87 \left(\frac{\sigma_{max}}{4\,000} \right)^2 (1-\gamma)\, Z\, L_{we}\, D_{we} \sin\alpha$$

Consequently, adopting σ_{max} = 4 000 MPa, gives Formula (37),

$$C_{0a} = 220 \left(1 - \frac{D_{we} \cos\alpha}{D_{pw}} \right) Z\, L_{we}\, D_{we} \sin \tag{37}$$

NOTE The value has been rounded for use in the final document.

5 Static equivalent load

5.1 Theoretical static equivalent radial load P_{0r} for radial bearings

5.1.1 Single-row radial bearings and radial contact groove ball bearings (nominal contact angle α = 0°)

Assuming both the bearing rings will yield a parallel displacement when a radial and axial loads act simultaneously on single-row radial bearings, the maximum rolling element load, Q_{max}, is given by Formula (38):

$$Q_{max} = \frac{F_r}{Z \cos \alpha \, J_r} = \frac{F_a}{Z \sin \alpha \, J_a} \tag{38}$$

The radial and axial load integrals are given by Formula (39)

$$\left.\begin{aligned} J_r = J_r(\varepsilon) = \frac{1}{2\pi}\int_{-\psi_o}^{+\psi_o}\left[1-\frac{1}{2\varepsilon}(1-\cos\psi)\right]^t \cos\psi \, d\psi \\ J_a = J_a(\varepsilon) = \frac{1}{2\pi}\int_{-\psi_o}^{+\psi_o}\left[1-\frac{1}{2\varepsilon}(1-\cos\psi)\right]^t d\psi \end{aligned}\right\} \tag{39}$$

where t is the exponent in the deflection-load formula

t = 3/2 for point contact;

t = 1,1 for line contact.

Assuming the bearing has no radial internal clearance under mounting, the static equivalent radial load becomes $P_{0r} = F_r$ when the rings displace in the radial direction (ε = 0,5). Consequently, the following formula can be obtained from Formula (38)

$$Q_{max} = \frac{P_{0r}}{Z \cos \alpha \, J_r(0,5)}$$

the following relationship yields

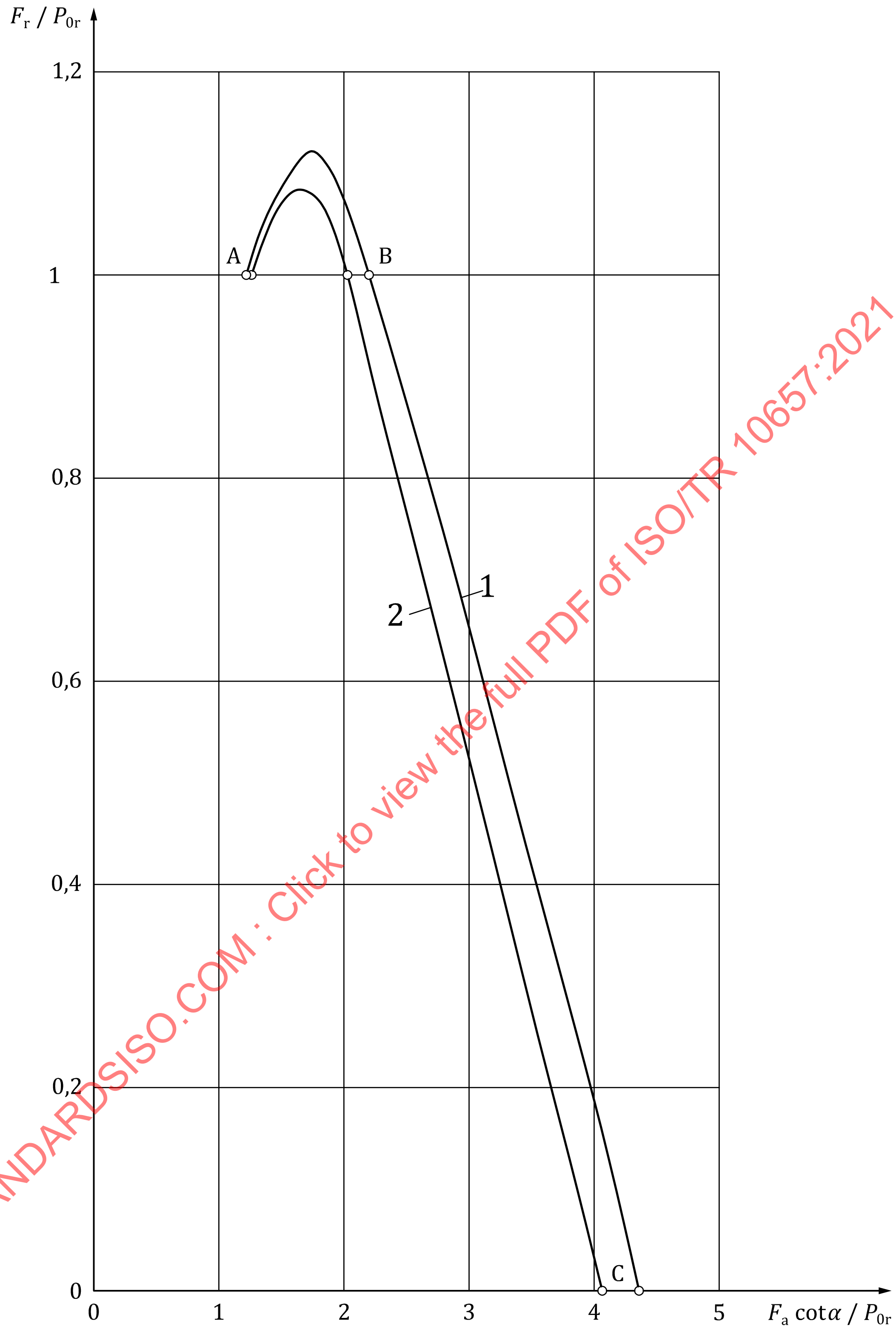
$$\frac{F_r}{P_{0r}} = \frac{J_r}{J_r(0,5)} \tag{40}$$

$$\frac{F_a \cot \alpha}{P_{0r}} = \frac{J_a}{J_r(0,5)} \tag{41}$$

The values calculated from Formula (40) and Formula (41) for a constant contact angle α are given in Table 1. In accordance with the functional relationship given in this table, the static equivalent radial load P_{0r} for the given values of F_r, F_a and α can be obtained. The relationship between F_r/P_{0r} and $F_a \cot \alpha/P_{0r}$ is also shown in Figure 1.

Table 1 — Values for F_r/P_{0r} and $F_a \cot \alpha/P_{0r}$ versus $F_r \tan \alpha/F_a$ for single-row radial bearings

ε	Ball bearings			Roller bearings		
	$F_r \tan \alpha/F_a$	F_r/P_{0r}	$F_a \cot \alpha/P_{0r}$	$F_r \tan \alpha/F_a$	F_r/P_{0r}	$F_a \cot \alpha/P_{0r}$
0,5	0,822 5	1	1,215 8	0,794 0	1	1,259 5
0,6	0,783 5	1,055 8	1,347 5	0,748 2	1,046 9	1,399 3
0,7	0,742 7	1,094 9	1,474 3	0,700 0	1,074 6	1,535 3
0,8	0,699 5	1,118 3	1,598 8	0,648 4	1,083 4	1,670 9
0,9	0,652 9	1,125 5	1,723 9	0,591 7	1,071 1	1,810 2
1	0,600 0	1,112 8	1,854 7	0,523 8	1,028 6	1,963 8
1,25	0,453 8	1,000 3	2,204 3	0,360 0	0,847 4	2,354 1
1,67	0,308 0	0,816 5	2,651 2	0,233 3	0,646 4	2,770 3
2,5	0,185 0	0,585 2	3,163 7	0,137 2	0,438 2	3,194 8
5	0,083 1	0,310 8	3,740 0	0,061 1	0,221 8	3,631 7
∞	0	0	4,370 6	0	0	4,076 6



Key

1 ball bearing

2 roller bearing

NOTE A, B and C are used to define line segments.

Figure 1 — Theoretical relationship between radial and axial loads versus static equivalent load for single-row radial bearings

Table 1 and Figure 1 are calculated and plotted based on the assumption of a constant contact angle. However, the above relationship is also approximately applicable to ball bearings (e.g. angular contact groove ball bearings), a contact angle of which varies with the load, if cot α' given by the following Formula (42) is substituted for cot α[12]

$$\frac{\cos\alpha}{\cos\alpha'} = 1 + \frac{c}{\frac{2r}{D_w} - 1}\left(\frac{F_a}{Z\,D_w^2\,\sin\alpha'}\right)^{2/3} \tag{42}$$

where

c is the compression constant depending on the elastic modulus and conformity 2 r/D_w;

r is the curvature radius of a raceway cross-section;

D_w is the nominal ball diameter (see Table 2).

Table 2 — Values of c and $\frac{c}{2\frac{r}{D_w}-1}$

2 r/D_w	1,032 5	1,035	1,037 5	1,06
$c \times 10^4$	4,321 7	4,387 1	4,474 5	4,954 7
$\frac{c}{2\frac{r}{D_w}-1}$	0,013 23	0,012 53	0,011 93	0,008 258
NOTE The unit of c is "1/MPa$^{2/3}$".				

For an example of 2 r/D_w = 1,035, the values of cot α' for each value of $\frac{F_a}{ZD_w^2}$ for angular contact groove ball bearings with α = 15° ~ 45° are given in Table 3.

Table 3 — Example of values of cot α' for angular contact groove ball bearings

α	$\frac{F_a}{ZD_w^2}$				
	0,5	1	2	5	10
	cot α'				
15°	3,024	2,793	2,526	2,154	1,865
20°	2,450	2,322	2,164	1,905	1,691
25°	1,997	1,929	1,834	1,664	1,511
30°	1,651	1,613	1,552	1,444	1,337
35°	1,381	1,356	1,317	1,248	1,171
40°	1,163	1,146	1,122	1,072	1,018
45°	0,975	0,969	0,952	0,920	0,879
NOTE $F_a / ZD_w^2 = (F_a / C_{0r})\ f_0\ \cos\alpha$, since $C_{0r} = f_0\ ZD_w^2\ \cos\alpha$.					

Furthermore, for single and double-row radial contact groove ball bearings, Table 4 can be obtained from Formula (40), Formula (41) and the following Formula (43)[7]:

$$\sin\alpha' \approx \tan\alpha' \approx \left(\frac{2c}{2\frac{r}{D_\mathrm{w}}-1}\right)^{3/8}\left(1-\frac{1}{2\varepsilon}\right)^{3/8}\left(\frac{F_\mathrm{a}}{J_\mathrm{a}\, i\, Z\, D_\mathrm{w}^2}\right)^{1/4} \quad (43)$$

For given values of F_r and F_a a provisional value of α' is found using Formula (44). Next, Table 4 is used to find ε and $F_\mathrm{r}/P_\mathrm{0r}$ or $F_\mathrm{a}\cot\alpha'/P_\mathrm{0r}$ and then P_0r can be determined.

Table 4 — Values of $F_\mathrm{r}/P_\mathrm{0r}$ and $F_\mathrm{a}\cot\alpha'/P_\mathrm{0r}$ versus $F_\mathrm{r}\tan\alpha'/F_\mathrm{a}$ for radial contact groove ball bearings

ε	$F_\mathrm{r}\tan\alpha'$/Fa	F_r / P_0r	F_a cot α'/ P_0r
0,5	∞	1	0
0,6	1,143 2	1,055 8	0,923 8
0,7	0,905 5	1,094 9	1,209 6
0,8	0,785 9	1,118 3	1,423 1
0,9	0,701 3	1,125 5	1,605 1
1	0,628 0	1,112 8	1,772 1
1,25	0,463 2	1,000 3	2,160 0
1,67	0,310 5	0,816 5	2,603 5
2,5	0,185 5	0,585 2	3,154 8
5	0,083 1	0,310 8	3,737 7
∞	0	0	4,370 6

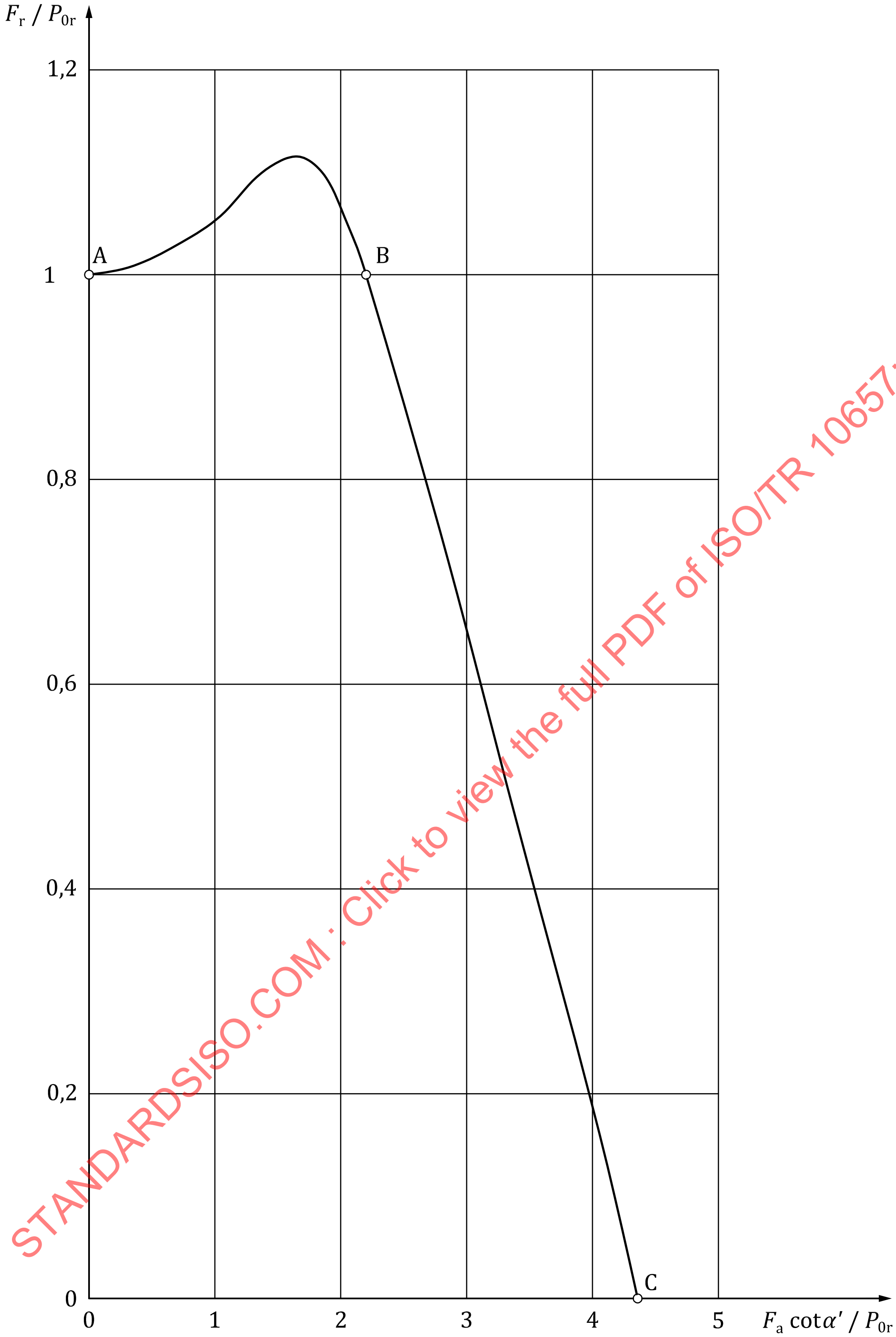
$$\tan\alpha' \approx \left(\frac{2c}{2\frac{r}{D_\mathrm{w}}-1}\right)^{3/8}\left(\frac{F_\mathrm{a}}{i\, Z\, D_\mathrm{w}^2}\right)^{1/4} \quad (44)$$

For $2\,r/D_\mathrm{w}$ = 1,035, for example, the values of $\tan\alpha'$ for each value of $F_\mathrm{a}/i\,Z\,D_\mathrm{w}^2$ are given in Table 5.

Table 5 — Example of values of contact angle for radial contact groove ball bearings

$F_\mathrm{a}/i\,Z\,D_\mathrm{w}^2$	0,5	1	2	5	10
$\tan\alpha'$	0,211 0	0,251 0	0,298 5	0,375 3	0,446 3
NOTE $F_\mathrm{a}/i\,Z\,D_\mathrm{w}^2 = (F_\mathrm{a}/C_\mathrm{0r})\,f_0$					

Moreover, the relationship between $F_\mathrm{r}/P_\mathrm{0r}$ and $F_\mathrm{a}\cot\alpha'/P_\mathrm{0r}$ is given in Figure 2.



NOTE A, B and C are used to define line segments.

Figure 2 — Theoretical relationship between radial and axial loads versus static equivalent load for radial contact bearings

5.1.2 Double-row radial bearings

Assuming both the bearing rings will yield a parallel displacement when a radial load and an axial load act simultaneously on a double-row radial bearing and designating each row as I and II,

$$F_r = F_{rI} + F_{rII};\quad F_a = F_{aI} + F_{aII}$$

and the maximum rolling element load for each row is given by Formulae (45), see Reference [16].

$$\left.\begin{aligned} Q_{max} &= \frac{F_r}{Z\cos\alpha\, J_r} = \frac{F_a}{Z\sin\alpha\, J_a} \\ Q_{maxII} &= Q_{maxI}\left(\frac{\varepsilon_{II}}{\varepsilon_I}\right)^t \end{aligned}\right\} \quad (45)$$

where

$$\left.\begin{aligned} J_r &= J_r(\varepsilon_I) + \left(\frac{\varepsilon_{II}}{\varepsilon_I}\right)^t J_r(\varepsilon_{II}) \\ J_a &= J_a(\varepsilon_I) - \left(\frac{\varepsilon_{II}}{\varepsilon_I}\right)^t J_a(\varepsilon_{II}) \end{aligned}\right\} \quad (46)$$

where t is the exponent in the deflection-load formula

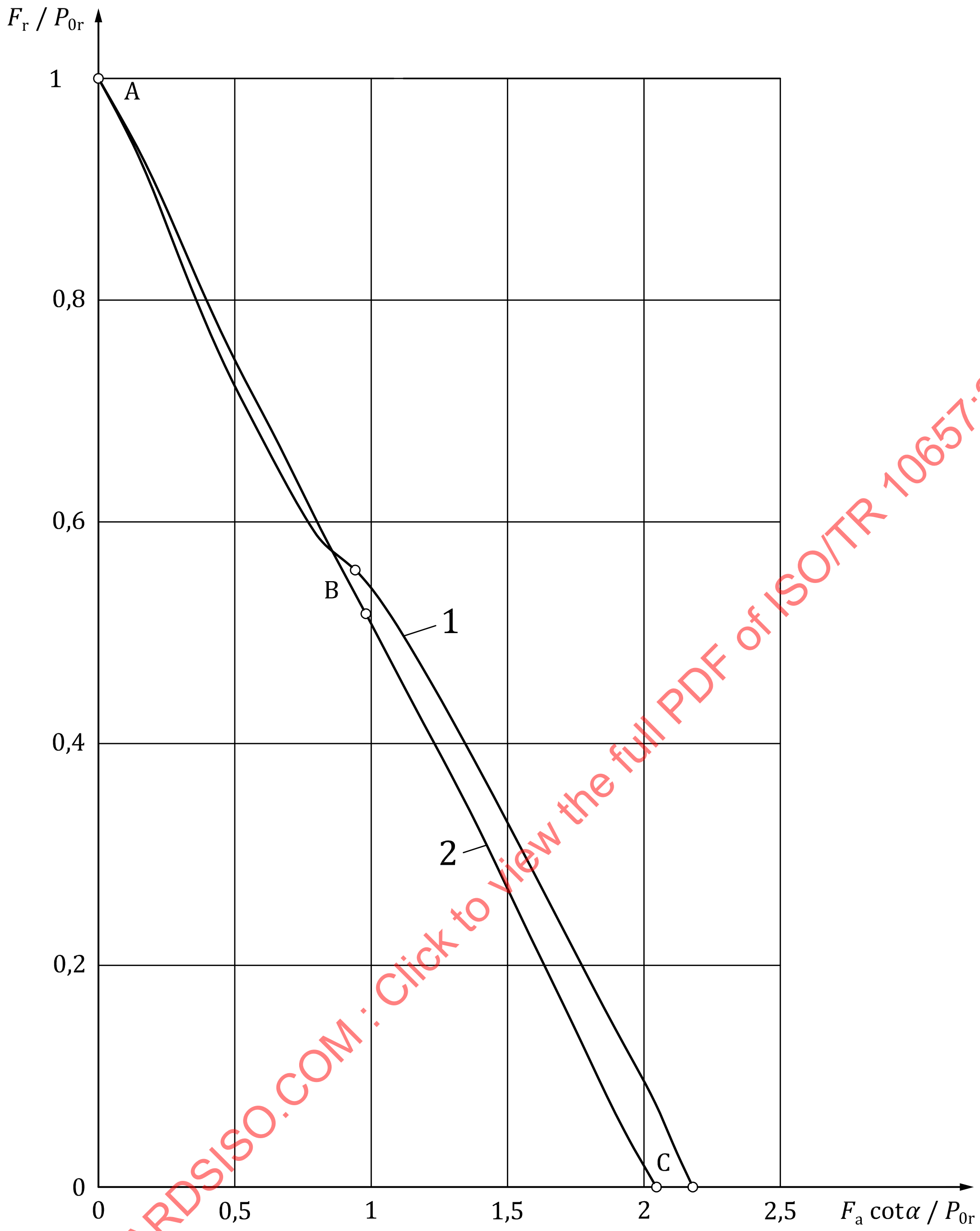
t = 3/2 for point contact;

t = 1,1 for line contact.

Assuming the bearing has no radial internal clearance, the static equivalent load $P_{0r} = F_r$, when the bearing rings displace in the radial direction ($\varepsilon_I = \varepsilon_{II} = 0{,}5$), and $Q_{max} = \dfrac{P_{0r}}{Z\cos\alpha\, J_r(0{,}5)}$ that is, in this case Formula (40) and Formula (41) are valid. The values calculated from Formula (40) and Formula (41) for a constant contact angle α are given in Table 6. In accordance with the functional relationship given in this table, the static equivalent radial load, P_{0r}, for the given values of F_r, F_a and α can be obtained. The relationship between F_r/P_{0r} and $F_a \cot\alpha/P_{0r}$ is shown in Figure 3. Furthermore, for double-row radial contact groove ball bearings, the contact angle of which varies with the load, P_{0r} can be obtained approximately by using α' by Formula (42) instead of α in Table 6.

Table 6 — Values of F_r/P_{0r} and $F_a \cot\alpha/P_{0r}$ versus $F_r \tan\alpha/F_a$ for double-row radial bearings

ε_I	ε_{II}	Ball bearings			Roller bearings		
		$F_r \tan\alpha/F_a$	F_r/P_{0r}	$F_a \cot\alpha/P_{0r}$	$F_r \tan\alpha/F_a$	F_r/P_{0r}	$F_a \cot\alpha/P_{0r}$
0,5	0,5	∞	1	0	∞	1	0
0,6	0,4	2,046 5	0,779 7	0,381 0	2,390 8	0,821 7	0,343 7
0,7	0,3	1,091 6	0,663 4	0,607 8	1,210 1	0,702 2	0,580 3
0,8	0,2	0,800 5	0,602 6	0,752 8	0,822 9	0,618 7	0,751 8
0,9	0,1	0,671 3	0,572 1	0,852 3	0,634 0	0,558 6	0,881 1
1	0	0,600 0	0,556 4	0,927 4	0,523 8	0,514 3	0,981 9
1,25	0	0,453 8	0,500 1	1,102 1	0,360 0	0,423 7	1,177 1
1,67	0	0,308 0	0,408 3	1,325 6	0,233 3	0,323 2	1,385 2
2,5	0	0,185 0	0,292 6	1,581 9	0,137 2	0,219 1	1,597 4
5	0	0,083 1	0,155 4	1,869 9	0,061 1	0,110 9	1,815 8
∞	0	0	0	2,185 0	0	0	2,038 3



Key

1 ball bearing

2 roller bearing

NOTE A, B and C are used to define line segments.

Figure 3 — Theoretical relationship between radial and axial loads versus static equivalent load for double-row radial bearings

5.2 Theoretical static equivalent axial load P_{0a} for thrust bearings

5.2.1 Single-direction thrust bearings

Single-direction thrust bearings which can support radial loads can be considered as single-row radial contact bearings with a large contact angle.

When bearing washers displace in the axial direction in Formula (41) which is valid for single-row radial contact bearings with a constant contact angle, $\varepsilon = \infty$ and $J_a = 1$, and since the static equivalent axial load $P_{0a} = F_a$, substituting this relationship, the following formula is obtained

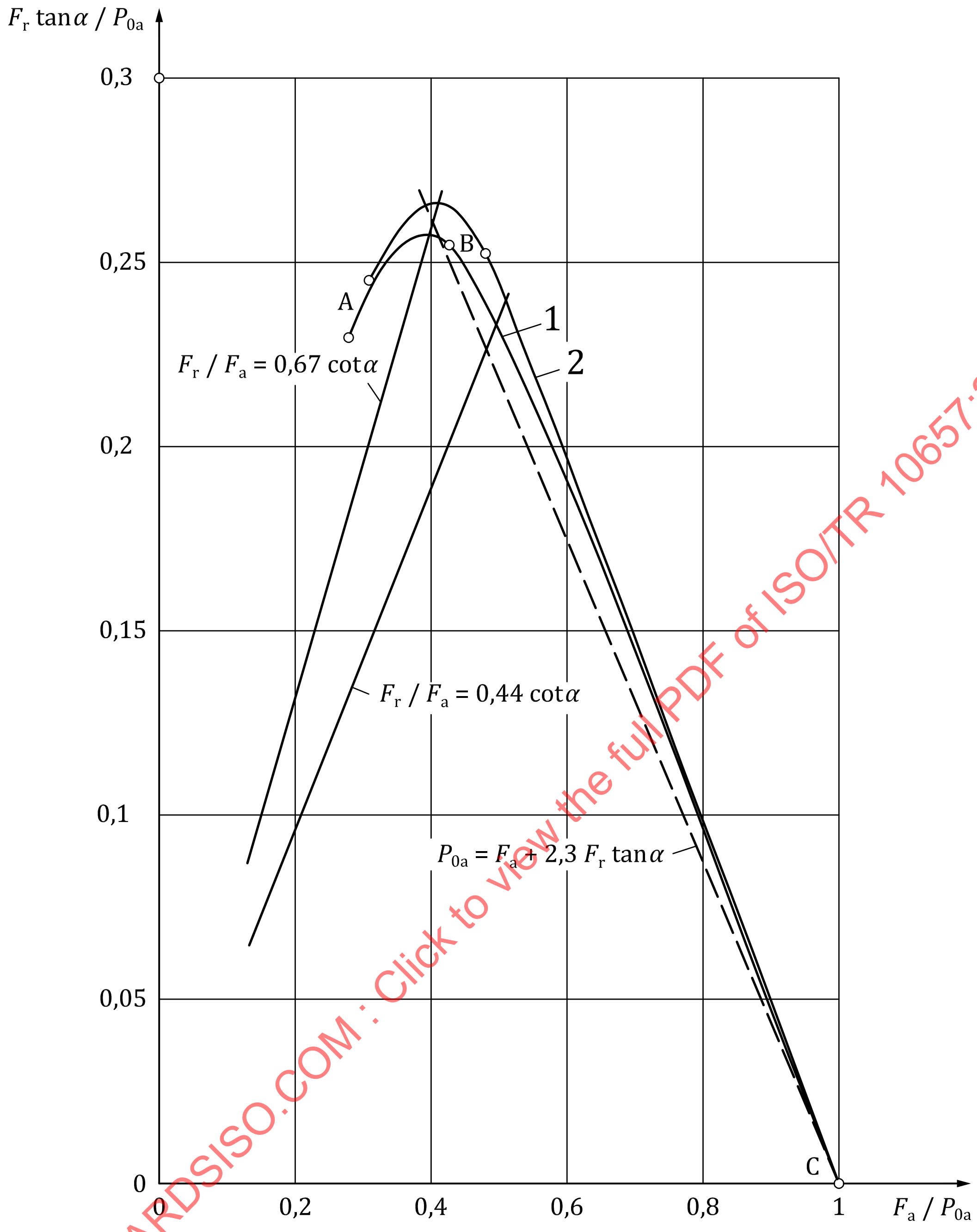
$$P_{0r} = P_{0a} \cot \alpha \, J_r \, (0{,}5)$$

Substituting this formula into Formula (40) and Formula (41), the following formulae yield

$$\frac{F_r \tan \alpha}{P_{0a}} = J_r \quad (47)$$

$$\frac{F_a}{P_{0a}} = J_a \quad (48)$$

The values in Table 7 can be obtained from Formula (47) and Formula (48). In accordance with the functional relationship given in this table, the static equivalent axial load P_{0a} for the given values of F_r, F_a and α can be obtained. The relationship between F_a / P_{0a} and $F_r \tan \alpha / P_{0a}$ is given in Figure 4.



Key

1 ball bearing

2 roller bearing

NOTE A, B and C are used to define line segments.

Figure 4 — Theoretical relationship between axial and radial loads versus static equivalent load for single-direction thrust bearings

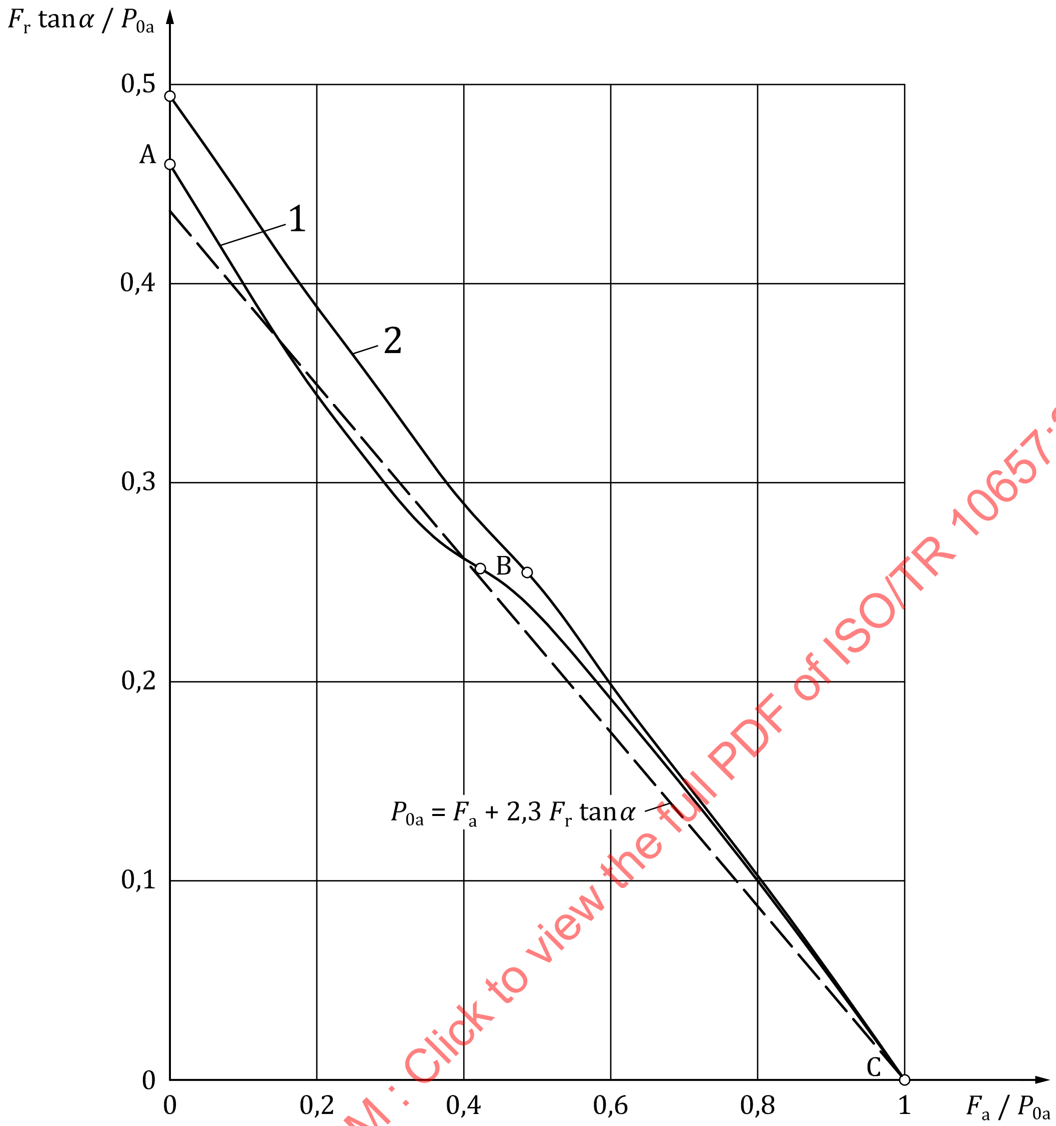
Table 7 — Values of F_a/P_{0a} and $F_r \tan \alpha/P_{0a}$ versus $F_r \tan \alpha/F_a$ for single-direction thrust bearings

ε	Ball bearings			Roller bearings		
	$F_r \tan \alpha/F_a$	F_a/P_{0a}	$F_r \tan \alpha/P_{0a}$	$F_r \tan \alpha/F_a$	F_a/P_{0a}	$F_r \tan \alpha/P_{0a}$
0,5	0,822 5	0,278 2	0,228 8	0,794 0	0,309 0	0,245 3
0,6	0,783 5	0,308 4	0,241 6	0,748 2	0,343 3	0,256 8
0,7	0,742 7	0,337 4	0,250 5	0,700 0	0,376 6	0,263 6
0,8	0,699 5	0,365 8	0,255 9	0,648 4	0,409 9	0,265 8
0,9	0,652 9	0,394 5	0,257 6	0,591 7	0,444 1	0,262 8
1	0,600 0	0,424 4	0,254 6	0,523 8	0,481 7	0,252 3
1,25	0,453 8	0,504 4	0,228 9	0,360 0	0,577 5	0,207 9
1,67	0,308 0	0,606 7	0,186 8	0,233 3	0,679 6	0,158 6
2,5	0,185 0	0,724 0	0,133 9	0,137 2	0,783 7	0,107 5
5	0,083 1	0,855 8	0,071 1	0,061 1	0,890 9	0,054 4
∞	0	1	0	0	1	0

5.2.2 Double-direction thrust bearings

Double-direction thrust bearings which can support radial loads can be considered as double-row radial contact bearings with a large contact angle.

For this case, the same Formula (47) and Formula (48) as for single-direction thrust bearings are valid and Table 8 can be obtained. In accordance with the functional relationship given in this table, the static equivalent axial load P_{0a} for the given values of F_r, F_a and α can be obtained. The relationship between F_a/P_{0a} and $F_r \tan \alpha/P_{0a}$ is given in Figure 5.



Key

1 ball bearing

2 roller bearing

NOTE A, B and C are used to define line segments.

Figure 5 — Theoretical relationship between axial and radial loads versus static equivalent load for double-direction thrust bearings

Table 8 — Values of F_a/P_{0a} and $F_r \tan \alpha/P_{0a}$ versus $F_r \tan \alpha/F_a$ for double-direction thrust bearings

ε_I	ε_{II}	Ball bearings			Roller bearings		
		$F_r \tan \alpha/F_a$	F_a/P_{0a}	$F_r \tan \alpha/P_{0a}$	$F_r \tan \alpha/F_a$	F_a/P_{0a}	$F_r \tan \alpha/P_{0a}$
0,5	0,5	∞	0	0,457 7	∞	0	0,490 6
0,6	0,4	2,046 5	0,174 4	0,356 8	2,390 8	0,168 6	0,403 1
0,7	0,3	1,091 6	0,278 2	0,303 6	1,210 1	0,284 7	0,344 5
0,8	0,2	0,800 5	0,344 5	0,275 8	0,822 9	0,368 9	0,303 5
0,9	0,1	0,671 3	0,390 0	0,261 8	0,634 0	0,432 3	0,274 1
1	0	0,600 0	0,424 4	0,254 6	0,523 8	0,481 7	0,252 3
1,25	0	0,453 8	0,504 4	0,228 9	0,360 0	0,577 5	0,207 9
1,67	0	0,308 0	0,606 7	0,186 8	0,233 3	0,679 6	0,158 6
2,5	0	0,185 0	0,724 0	0,133 9	0,137 2	0,783 7	0,107 5
5	0	0,083 1	0,855 8	0,071 1	0,061 1	0,890 9	0,054 4
∞	0	0	1	0	0	1	0

5.3 Approximate formulae for theoretical static equivalent load

5.3.1 Radial bearings

From a practical standpoint, it is preferable to replace the theoretical curves in Figure 1 and Figure 3 for radial contact bearings with a constant contact angle by two straight line segments AB, BC and a straight line AC, respectively.

The static equivalent radial load P_{0r} given by the straight-line segments and straight lines is shown in Table 9 and Table 10, respectively.

For radial ball bearings the contact angle of which varies with the load, the formulae given in Table 9 and Table 10 are approximately applicable if $\cot \alpha'$ by Formula (42) is substituted for $\cot \alpha$ in the formulae.

Table 9 — Approximate formulae for theoretical static equivalent radial loads for single-row radial bearings (straight line segments AB and BC in Figure 1)

Bearing type	Abscissa			Approximate formulae of P_{0r}	
	Point A	Point B	Point C	Segment AB	Segment BC
Single-row ball bearings	(1,22, 1)	(2,21, 1)	(4,37, 0)	$P_{0r} = F_r$	$P_{0r} = 0{,}494\ F_r + 0{,}229 \cot \alpha\ F_a$
Single-row roller bearings	(1,26, 1)	(2,03, 1)	(4,08, 0)	$P_{0r} = F_r$	$P_{0r} = 0{,}502\ F_r + 0{,}245 \cot \alpha\ F_a$

Table 10 — Approximate formulae for theoretical static equivalent radial loads for double-row radial bearings (straight line AC in Figure 3)

Bearing type	Abscissa of point C	Approximate formulae of P_{0r}
Double-row ball bearings	(2,18, 0)	$P_{0r} = F_r + 0{,}459 \cot \alpha\ F_a$
Double-row roller bearings	(2,04, 0)	$P_{0r} = F_r + 0{,}490 \cot \alpha\ F_a$

For radial contact groove ball bearings replacing theoretical curve by straight line segments AB and BC in Figure 2, the static equivalent radial load P_{0r} is given in Table 11, and the value of $\cot \alpha'$ in the table is given by Formula (44).

Table 11 — Approximate formula for theoretical static equivalent radial load for radial contact groove ball bearings (straight line segments AB and BC in Figure 2)

Bearing type	Abscissa		Approximate formulae of P_{0r}	
	Point B	Point C	Segment AB	Segment BC
Radial contact groove ball bearings	(2,16, 1)	(4,37, 0)	$P_{0r} = F_r$	$P_{0r} = 0{,}506\ F_r + 0{,}229 \cot \alpha'\ F_a$

5.3.2 Thrust bearings

Replacing theoretical curves in Figure 4 and Figure 5 by the straight lines BC and AC, respectively, the static equivalent axial load P_{0a} is given in Table 12 and Table 13.

Table 12 — Approximate formulae for theoretical static equivalent axial loads for single-direction thrust bearings (straight line BC in Figure 4)

Bearing type	Coordinates of point B	Approximate formulae of P_{0a}
Single-direction ball bearings	(0,424, 0,255)	$P_{0a} = 2{,}26 \tan \alpha\ F_r + F_a$
Single-direction roller bearings	(0,482, 0,252)	$P_{0a} = 2{,}06 \tan \alpha\ F_r + F_a$

Table 13 — Approximate formulae for theoretical static equivalent axial loads for double-direction thrust bearings (straight line AC in Figure 5)

Bearing type	Coordinates of point A	Approximate formulae of P_{0a}
Double-direction ball bearings	(0, 0,458)	$P_{0a} = 2{,}18 \tan \alpha\ F_r + F_a$
Double-direction roller bearings	(0, 0,491)	$P_{0a} = 2{,}04 \tan \alpha\ F_r + F_a$

5.4 Practical formulae of static equivalent load

5.4.1 Radial bearings

Assuming the bearing has no radial internal clearance, maximum rolling element loads under radial load for single radial contact bearings with a contact angle α are given as follows since J_r (0,5) = 0,228 8 (ball bearings), J_r (0,5) = 0,245 3 (roller bearings) for ε = 0,5 in Formula (38),

$$Q_{max} = \frac{F_r}{0{,}228\,8\ Z \cos \alpha} = \frac{4{,}37\ F_r}{Z \cos \alpha} \quad \text{(ball bearings)}$$

$$Q_{max} = \frac{F_r}{0{,}245\,3\ Z \cos \alpha} = \frac{4{,}08\ F_r}{Z \cos \alpha} \quad \text{(roller bearings)}$$

However, taking into account internal clearances in practice, the following formula for either ball bearings or roller bearings has been adopted since R. Stribeck (1901)

$$Q_{max} = 5 \frac{F_r}{Z \cos \alpha} \tag{49}$$

Moreover, the factor f_0 involved in the formulae for basic radial load rating C_{0r} of radial bearings also is based on the assumption by which load distributions depend on Formula (49) (see ISO 76:2006, Table 1, NOTE).

For single-row radial bearings with a constant contact angle under combined radial load F_r and axial load F_a, the following formula can be obtained owing to H. Stellrecht[8] where all of rolling elements are subjected to the load (see Figure 6).

$$\frac{5P_{0r}}{Z\cos\alpha}=\frac{F_a}{Z\sin\alpha}+\frac{2{,}5F_r}{Z\cos\alpha} \qquad (50)$$

In the case of the first term is greater than or equal to the second term in the right hand of the above Formula (50), that is $F_r \leq 0{,}4 \cot\alpha\, F_a$ (corresponds to $\varepsilon > 1{,}25$ in Table 1 and segment BC in Figure 1), this formula is valid.

The following formula can be obtained from Formula (50)

$$P_{0r} = 0{,}5\,F_r + 0{,}2\cot\alpha\,F_a \qquad (51)$$

When $F_r = 0{,}4\cot\alpha\,F_a$, the right-hand terms in Formula (51) will equal F_r, thus for $F_r < 0{,}4\cot\alpha\,F_a$, P_{0r} is set to

$P_{0r} = F_r$ (Segment AB in Figure 7)

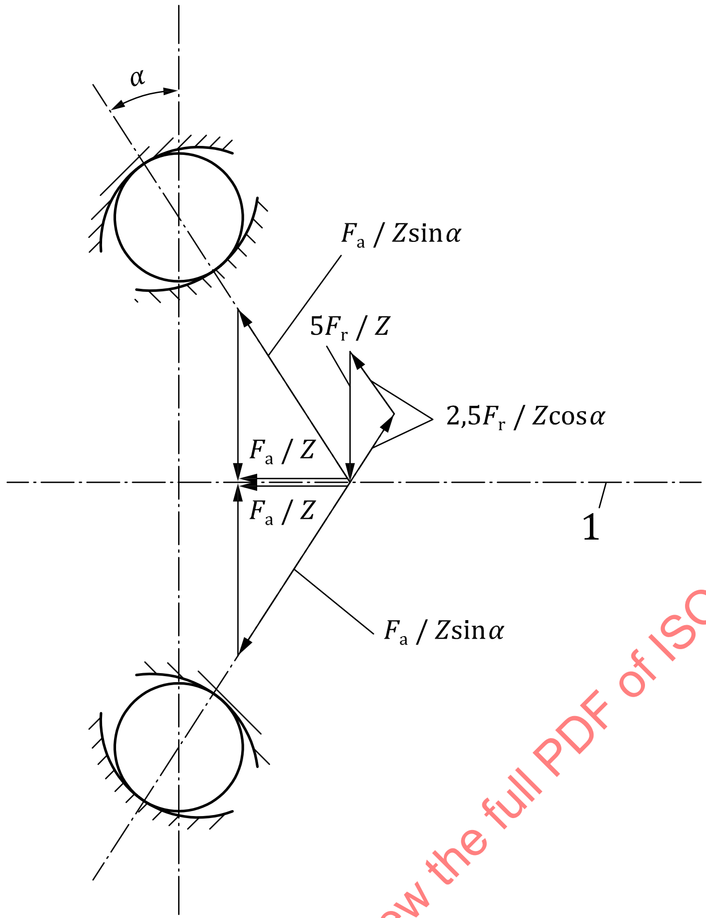
When a double-row radial bearing with a number of rolling elements per row Z is subjected to a load by its one row only and all of the rolling elements Z are subjected to the load (corresponds to $\varepsilon \geq 1$ in Table 6 and segment BC in Figure 3), the following formula can be obtained from the same consideration as for the above single-row bearings

$$\frac{5P_{0r}}{2Z\cos\alpha}=\frac{F_a}{Z\sin\alpha}+\frac{2{,}5F_r}{Z\cos\alpha} \qquad (52)$$

Consequently,

$$P_{0r} = F_r + 0{,}4\cot\alpha\,F_a \qquad (53)$$

This formula is valid where $F_r \leq 0{,}4\cot\alpha\,F_a$ (segment B_1C_1 in Figure 7), and $P_{0r} = 2\,F_r$ (point B_1) where $F_r = 0{,}4\cot\alpha\,F_a$. $P_{0r} = F_r$ at point A_1 in Figure 7, and it can be considered that Formula (53) also is valid for segment A_1B_1.



Key

1 bearing axis

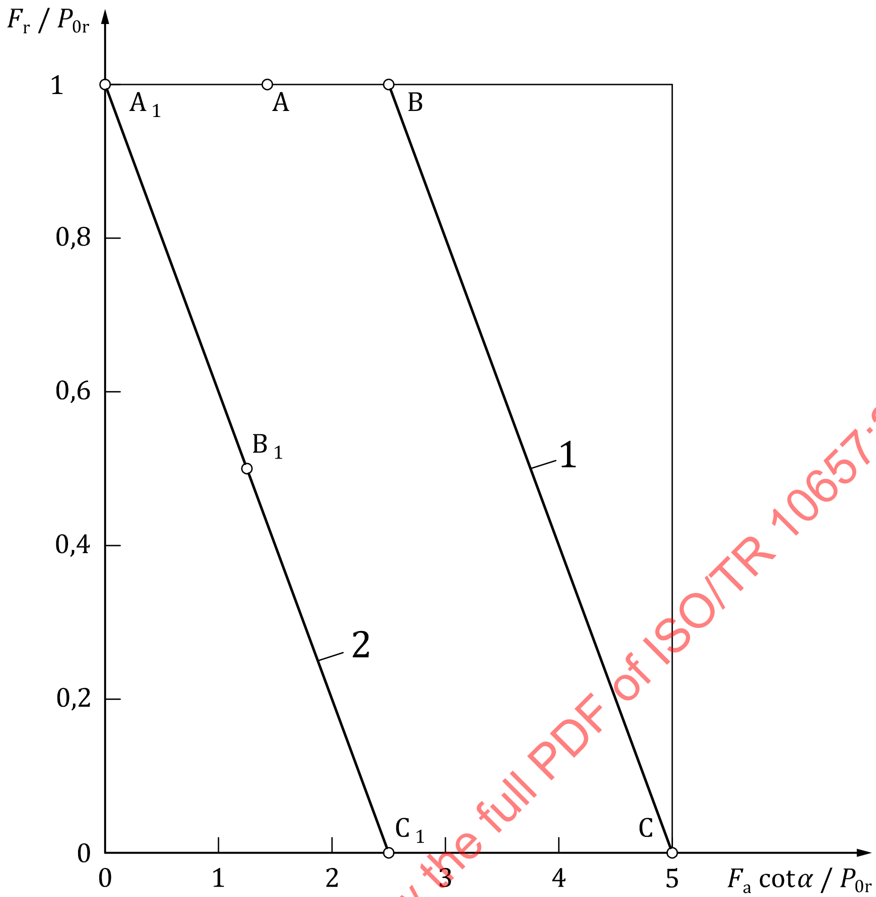
Figure 6 — Rolling element load in a single-row radial bearing

For radial bearings, for which the contact angles vary with the axial loads, α' can be adopted instead of α in Formula (51) and Formula (53).

That is,

Single-row bearings: $P_{0r} = 0{,}5\ F_r + 0{,}2\ \cot\alpha'\ F_a$ (54)

Double-row bearings: $P_{0r} = F_r + 0{,}4\ \cot\alpha'\ F_a$ (55)



Key

1 single-row radial bearing

2 double-row radial bearing

NOTE A, B, C and C_1 are used to define line segments.

Figure 7 — Relationship between radial and axial loads versus static equivalent load for radial bearings

However, strictly speaking, α only on the right hand in Formula (50) and Formula (52) can be replaced by α', and it is necessary to take into account that the static load capacity decreases because of manufacturing inaccuracy as an axial run out where a radial bearing is subjected to an axial load. Therefore, taking a reduction factor $(1 - k_0 \sin \alpha)$, Formulae (54) and (55) for P_{0r} become as follows:

— single-row bearings: $$P_{0r} = \frac{0{,}5 \cos \alpha}{\cos \alpha'} F_r + \frac{0{,}2 \cos \alpha}{\sin \alpha' (1 - k_0 \sin \alpha)} F_a \quad (56)$$

— double-row bearings: $$P_{0r} = \frac{\cos \alpha}{\cos \alpha'} F_r + \frac{0{,}4 \cos \alpha}{\sin \alpha' (1 - k_0 \sin \alpha)} F_a \quad (57)$$

There is an experiment by A. Palmgren[9] for the magnitude of rolling element load under combined load for the single-row radial contact groove ball bearing. Figure 8 shows his results for F_r/P_{0r} dependent on $F_a \cot \alpha'/P_{0r}$.

NOTE This experiment had been done for bearings which have D_w = 16,5 mm, groove radius r = 0,53 D_w, Z = 12, and the value of α' for the experiment values in Figure 8 is based on Formula (58) (the unit for F_a is kgf):

$$\tan\alpha' \approx \left(\frac{2c}{2\frac{r}{D_w}-1}\right)^{3/8}\left(\frac{F_a}{Z D_w^2}\right)^{1/4} = 0{,}050\,24\,F_a^{1/4} \quad (58)$$

Taking the straight line AC_1 on the safety side of all of experiment values shown in Figure 8, the following formula is obtained

$$P_{0r} = F_r + 0{,}2 \cot\alpha' F_a \quad (59)$$

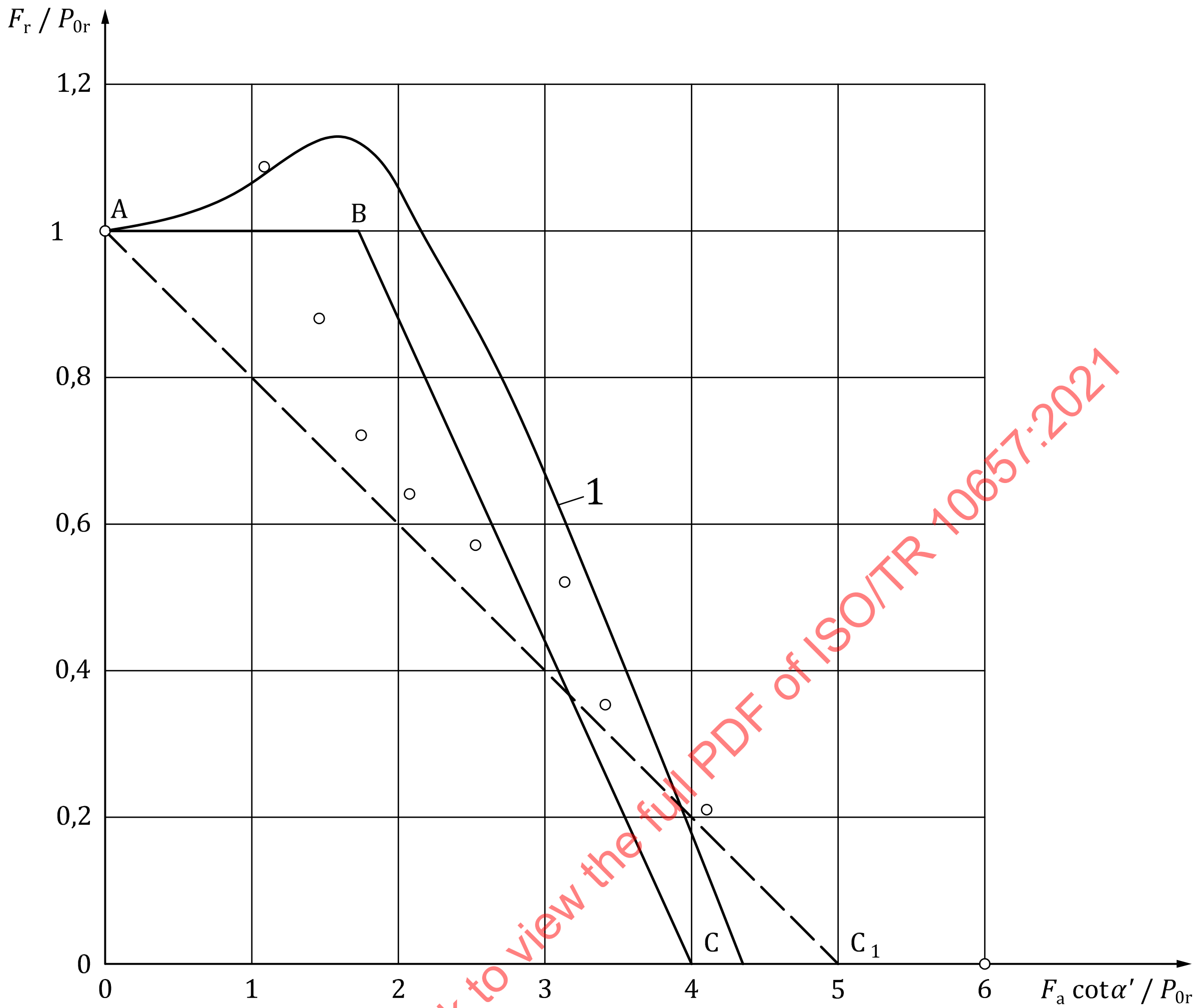
And also, in Figure 8, the straight-line segments AB and BC [coordinates of point B (1,7, 1,0)] can take into account theoretical curve (see Figure 2) and experiment results. In this case, the following formula is obtained

$$P_{0r} = 0{,}575\,F_r + 0{,}25 \cot\alpha' F_a \quad (60)$$

5.4.2 Thrust bearings

Since single and double-direction thrust bearings with a contact angle $\alpha \neq 90°$ can be regarded as single and double-row radial bearings with a large constant contact angle, the formula for the static equivalent axial load of thrust bearings is based on Formula (51) and Formula (53). Since the ratio of static radial load capacity to static axial load capacity is 0,2 cot α for single-row bearings and 0,4 cot α for double-row bearings, the following formula is obtained by means of dividing both sides in Formula (51) and Formula (53) by 0,2 cot α and 0,4 cot α, respectively,

$$P_{0a} = 2{,}5\;F_r\;\tan\alpha + F_a \quad (61)$$



Key

1 theoretical curve

○ experiment value by A. Palmgren (1920)

NOTE A, B, C and C_1 are used to define line segments.

Figure 8 — Relationship between radial and axial loads versus static equivalent load for radial contact groove ball bearings

5.5 Static radial load factor X_0 and static axial load factor Y_0

5.5.1 Radial bearings

5.5.1.1 Radial contact groove ball bearings

For radial contact groove ball bearings, from Formula (59) for the static equivalent radial load,

$$X_0 = 1; \quad Y_0 = 0{,}2 \cot \alpha' \tag{62}$$

However, using in Formula (62) the contact angle $\alpha' = 15°$ corresponding to

a) conformity $2\,r/D_w = 1{,}035$ and axial load $F_a \approx 0{,}1$ C0r (relatively small in safety side), or

b) $2\ r/D_w = 1{,}035$ and $F_a/i\ Z\ D_w^2\ \sin\alpha' = 4{,}903\ 3\ \text{N/mm}^2$,

Y_0 yields

$$Y_0 = 0{,}2 \times 3{,}732 = 0{,}746\ 4 \approx 0{,}75$$

Factors $X_0 = 1$ and $Y_0 = 0{,}75$ had been adopted at the initial period[10],[11].

NOTE 1 For a), from Formula (44), $\tan\alpha' = 0{,}263\ 6$ (see Table 14). Consequently, $\alpha' = 14{,}8°$.

NOTE 2 For b), from Formula (42), $\cos\alpha' = 0{,}965\ 08 \cos\alpha$ is obtained. Consequently, $\alpha' = 15{,}2°$.

Secondly, from Formula (60),

$$X_0 = 0{,}575 \approx 0{,}6; \quad Y_0 = 0{,}25 \cot\alpha \tag{63}$$

However, if $2\ r/D_w = 1{,}035$, from Formula (44) (units: kgf and mm),

$$\tan\alpha' = 0{,}444\,18\left(\frac{F_a}{i\ Z\ D_w^2}\right)^{1/4} = 0{,}468\,72\left(\frac{F_a}{C_{0r}}\right)^{1/4}$$

yields [$C_{0r} = 1{,}24\ i\ Z\ D_w^2$ (kgf)], values of Y_0 versus each value for F_a/C_{0r} become as shown in Table 14.

Table 14 — Values of factor Y_0 for radial contact groove ball bearings

F_a/C_{0r}	0,05	0,1	0,2	0,5	1
$\tan\alpha'$	0,221 6	0,263 6	0,313 5	0,394 1	0,468 7
$Y_0 = 0{,}25 \cot\alpha'$	1,128	0,948	0,797	0,634	0,533
Practical values for Y_0 [Formula (63)]	1,1	0,9	0,8	0,6	0,5

$X_0 = 0{,}6$ and practical values for Y_0 in the lowest line of the above Table 14 had been adopted after $X_0 = 1$ and $Y_0 = 0{,}75$[12],[13].

However, ISO/R 76 and ISO 76 have the given $X_0 = 0{,}6$ and $Y_0 = 0{,}5$ (corresponds to $F_a/C_{0r} = 1$). As for this, A. Palmgren had mentioned the following sentence in his book[14]: "The Y_0 factors for radial contact groove ball bearings are calculated for thrust loads of the same magnitude as basic static load rating. With lighter thrust loads, smaller contact angles are developed, and then somewhat greater Y_0 factors can be justified."

The fact that ISO/R 76 specified the values of Y_0 only corresponding to $F_a \approx C_{0r}$ is based on its influence being relatively small except F_a/C_{0r} ($\alpha \neq 0°$) and F_r/C_{0r} being particularly similar (consequently, P_{0r}/C_{0r} being small), owing to there being the condition of $P_{0r} \geq F_r$, and the calculation of the static equivalent load necessary to design is done to ascertain $P_{0r} \leq C_{0r}$ usually.

Moreover, if $\alpha = 5°$ is adopted taking into account radial internal clearances, the value of α' for $2\ r/D_w = 1{,}035$ and $F_a = C_{0r} = 12{,}258\ i\ Z D_w^2$ is 26,6° from Formula (42) and $Y_0 = 0{,}25 \cot\alpha' = 0{,}499\ 2 \approx 0{,}5$ yields from Formula (63) and Formula (56),

$$X_0 = 0{,}5\frac{\cos\alpha}{\cos\alpha'} = 0{,}557\,1 \approx 0{,}6$$

$$Y_0 = \frac{0{,}2\cos\alpha}{\sin\alpha'(1 - 0{,}2\sin\alpha)} = 0{,}452\,9 \approx 0{,}5$$