TECHNICAL REPORT

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Second edition
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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).

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

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

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

This second edition cancels and replaces the 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),

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

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

$$\sigma_{max} = \frac{3Q}{2\pi ab} \text{ or } Q = \frac{2\pi ab}{3}\sigma_{max} \quad (2)$$

According to the Hertz's theory,

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

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

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

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

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

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

$\rho_{12} \quad = 0$

$\rho_{22} \quad = 0$

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

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

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

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

$\nu \quad = 0{,}3$

Consequently,

$$Q = 2{,}762\,173\,2\times10^{-5}\frac{L_{we}}{\Sigma\rho}\sigma_{max}^2 \tag{11}$$

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,

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

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

where

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

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

$f_{i(e)}$ denotes

$f_i \quad = \dfrac{r_i}{D_w}$ for inner ring contact, and

$f_e \quad = \dfrac{r_e}{D_w}$ for outer ring contact

Substituting Formula (12) into Formula (7),

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

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.