TECHNICAL REPORT

ISO/TR 10657

First edition
1991-05-15

Explanatory notes on ISO 76

Notes explicatives sur l'ISO 76

Reference number
ISO/TR 10657 : 1991 (E)

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International Organization for Standardization
Case postale 56 • CH-1211 Genève 20 • Switzerland
Printed in Switzerland

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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 main task of ISO technical committees is to prepare International Standards. In exceptional circumstances a technical committee may propose the publication of a Technical Report of one of the following types:

— type 1, when the required support cannot be obtained for the publication of an International Standard, despite repeated efforts;

— type 2, when the subject is still under technical development or where for any other reason there is the future but not immediate possibility of an agreement on an International Standard;

— type 3, when a technical committee has collected data of a different kind from that which is normally published as an International Standard ("state of the art", for example).

Technical Reports of types 1 and 2 are subject to review within three years of publication, to decide whether they can be transformed into International Standards. Technical Reports of type 3 do not necessarily have to be reviewed until the data they provide are considered to be no longer valid or useful.

ISO/TR 10657, which is a Technical Report of type 3, was prepared by Technical Committee ISO/TC 4, *Rolling bearings,* Sub-Committee SC 8, *Load ratings and life.*

ISO/TR 10657 has been prepared for the guidance of users of ISO 76 : 1987, *Rolling bearings — Static load ratings*. It is a purely scientific document intended for use by specialists in this field, and it is not envisaged that it will become an International Standard.

Annexes A, B and C form an integral part of this Technical Report.

Explanatory notes on ISO 76

1 Scope

This Technical Report gives supplementary background information regarding the derivation of formulae and factors given in ISO 76, Rolling bearings - Static load ratings.

2 Brief History

2.1 ISO/R 76 - 1958

The ISO Recommendation 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, the Secretariat of which is held by the Sveriges Standardiseringskommission (SIS).

This Recommendation is based on the studies[1]*, [2] by A. Palmgren and so on. It is defined in the Recommendation that the basic static load ratings correspond to a total permanent deformation of rolling element and raceway at the most heavily stressed rolling element/ raceway contact of 0,0001 of the rolling element diameter. And then the Standard values confined to the basic static load ratings for special inner design rolling bearings are laid down.

* Figures in brackets indicate literature references in annex C.

Technical Committee ISO/TC 4 discussed the questions dealt with by the ISO Recommendation at the following meetings:

the third meeting, held in Göteborg, in September 1953,

the fourth meeting, held in Madrid, in May 1955,

the fifth meeting, held in Vienna, in September 1956.

At the third meeting of the Technical Committee, Working Group No. 3 was appointed to assist the ISO/TC 4 Secretariat in preparatory work and in drawing up proposals. The Working Group composed of Germany, Sweden and USA held the following meetings:

the first meeting, held in Madrid, in May 1955,

the second meeting, held in Vienna, in September 1956.

On 4th June 1957, the Draft ISO Recommendation was sent out to all the ISO Member Bodies and was approved by 28 (out of a total of 38) Member Bodies.

The Draft ISO Recommendation was then submitted by correspondence to the ISO Council, which decided, in December 1958, to accept it as an ISO Recommendation.

2.2 ISO 76 - 1978

The Working Group No. 3 was transformed into SC 8 at the 13th meeting held in Paris in May 1972. The SC 8 Secretariat proposed to include the revision of ISO/R 76 in the future work at the first meeting held in London in November 1973, and SC 8 requested its Secretariat to prepare a Draft Proposal for an International Standard replacing ISO/R 76 and it was decided that this proposal should be submitted to the SC 8 member Bodies for consideration prior to 1 October 1974 (SC 8 RESOLUTION 21, 21 London 1973).

The SC 8 Secretariat distributed a DRAFT PROPOSAL (Revision of ISO/R 76) in July 1974.

The TC 4 decided to include the revision of ISO/R 76 in its programme of work (TC 4 RESOLUTION 514, item No. 132) and SC 8 Secretariat was requested to prepare a Draft Proposal (SC 8 RESOLUTION 38, 13 Miami Beach 1974). As a result, the Secretariat submitted a DP [3] in January 1976.

The Draft Proposal DP 76 was accepted by correspondence by 6 of the 8 Members of SC 8. Of the remaining two, Japan would prefer further study and USA its counter proposal, document 4/8 N 64 [4]. The DIS 76 was then submitted to the ISO Central Secretariat. After the DIS had been approved by the ISO Member Bodies, the ISO Council decided in June 1978 to accept it as an International Standard.

This Standard 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 15^0 and 45^0 for angular contact groove ball bearings were added to the table which shows the values of X_0 and Y_0 in the formulae to calculate the static equivalent radial loads of radial ball bearings.

2.3 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 became No. 157 of the programme of work of TC 4.

This Draft Proposal DP 76 was dealt with at the following TC 4 meetings:

15th meeting, held in Moscow, in April 1977,

16th meeting, held in London, in November 1979,

17th meeting, held in Budapest, in May 1983,

and then, dealt with at the following SC 8 meetings:

the third meeting, held in London, in November 1979,

the fourth meeting, held in Budapest, in May 1983,

the fifth meeting, held in Arlington, in November 1984.

The following resolutions for the contents of the Standard were adopted during the third meeting to the fifth meeting:

SC 8, considering the proposals made in the documents 4/8 N 75 [5] and 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 (SC 8 RESOLUTION 45, 5 London 1979), and SC 8 requested its Secretariat to prepare a new draft. The new draft should be prepared with the principles and formulae of the document 4/8 N 75, and to include levels of contact stress for various rolling element contact stated to be generally corresponding to a permanent deformation of $10^{-4}D_w$ at the centre of the most heavily stressed rolling element/raceway contact. For roller bearings a stress level of 4000 MPa was agreed (SC 8 RESOLUTION 51, 4 Budapest 1983) and then SC 8 agreed, by a majority vote, that static load ratings should correspond to calculated contact stresses of

4000 MPa for roller bearings,

4600 MPa for self-aligning ball bearings and

4200 MPa for all other ball bearings to which the

Standard applies (SC 8 RESOLUTION 56, 3 Arlington 1984). Moreover, SC 8 recommended the document 4/8 N 121 [7] , amended in accordance with SC 8 Resolution 56, as a revised ISO 76 (SC 8 RESOLUTION 57, 4 Arlington 1984).

For these calculated contact stresses, a total permanent deformation occures at the centre of the most heavily stressed rolling element /raceway contact, and its deformation is approximately 0,0001 of the rolling element diameter.

The DIS 76 was submitted to the ISO Central Secretariat 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, SC 8 decided at its fifth meeting in April 1986 that supplementary background information, regarding the derivation of formulae and factors given in ISO 76, should be published as a Technical Report (SC 8 RESOLUTION 71, 11 Hangzhou 1986).

3 Basic Static Load Ratings

(1) Basic equation for point contact The relationship between a calculated contact stress and a rolling element load within an elliptical contact area is given as follows [8],

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

where

σ = calculated contact stress, MPa,

a = major semi axis of the contact ellipse, mm,

b = minor semi axis of the contact ellipse, mm,

Q = normal force between rolling element and raceway, N,

x = distance in a-direction, mm,

y = distance in b-direction, mm.

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

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

According to the Hertz's theory,

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

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

where

$\varkappa = a/b$

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

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

E_1, E_2 = modulus of elasticity (Young's modulus), MPa,

ν_1, ν_2 = Poisson's ratio,

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

$\rho_{11}, \rho_{21} = \frac{2}{D_w}$ = principal curvature of body 1 (ball),

ρ_{12}, ρ_{22} = principal curvature of body 2 (ring) at the point contact.

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

$$Q = \sigma_{max}^3 \frac{32\pi}{3E_o^2} \varkappa \left(\frac{E(\varkappa)}{\sum\rho}\right)^2 , \quad (3\text{-}5)$$

and

$$1 - \frac{2}{\varkappa^2 - 1}\left(\frac{K(\varkappa)}{E(\varkappa)} - 1\right) - F(\rho) = 0 , \quad (3\text{-}6)$$

where

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

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

$\nu = 0{,}3$

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

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

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

Consequently,

$$Q = 6{,}4762065 \times 10^{-10} \kappa \left(\frac{E(\kappa)}{\Sigma \rho}\right)^2 \sigma_{max}^3 \; . \qquad (3\text{-}7)$$

(2) Basic equation for line contact   The relationship between a calculated contact stress and a rolling element load for a line contact is given as follows [9] ,

$$\sigma = \frac{2Q}{\pi L_{we} b} \left[1 - \left(\frac{y}{b}\right)^2 \right]^{1/2} , \qquad (3\text{-}8)$$

where

σ = calculated contact stress, MPa ,

b = semiwidth of the contact surface, mm ,

L_{we} = length of roller applicable to calculate load ratings, mm ,

Q = normal force between rolling element and raceway, N ,

y = distance in b-direction, mm .

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

$$\sigma_{max} = \frac{2Q}{\pi L_{we} b} , \quad \text{or} \quad Q = \frac{\pi L_{we} b}{2} \sigma_{max} \; . \qquad (3\text{-}9)$$

And also b is given by the following equation,

$$b = \left[\frac{4Q}{\pi L_{we} \Sigma \rho} \left(\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right) \right]^{1/2} , \qquad (3\text{-}10)$$

where

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

$$\rho_{11} = \frac{2}{D_{we}} , \quad \rho_{12} = \pm \frac{2}{D_{we}} \frac{\gamma}{1 \mp \gamma} , \rho_{21} = 0 , \quad \rho_{22} = 0 ,$$

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

D_{we} = roller diameter applicable to calculate load ratings, mm ,

$$\gamma = \frac{D_{we} \cos\alpha}{D_{pw}} ,$$

D_{pw} = pitch diameter of roller set, mm .

Substituting equation (3-10) into equation (3-9) for the case of
$E_1 = E_2 = E$ and $\nu_1 = \nu_2 = \nu$,

$$Q = 2\pi\sigma^2_{max} \frac{L_{we}}{E_o \sum\rho} ,$$

where

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

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

$\nu = 0{,}3$.

Consequently,

$$Q = 2{,}7621732 \times 10^{-5} \frac{L_{we}}{\sum\rho} \sigma^2_{max} . \qquad (3\text{-}11)$$

3.1 Basic static radial load rating C_{or} for radial ball bearings

3.1.1 Radial and angular contact groove ball bearings

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

$$\sum\rho = \frac{2}{D_w}\left(2 \pm \frac{\gamma}{1 \mp \gamma} - \frac{1}{2f_{i(e)}}\right), \qquad (3\text{-}12)$$

$$F(\rho) = \frac{\pm\dfrac{\gamma}{1 \mp \gamma} + \dfrac{1}{2f_{i(e)}}}{2 \pm \dfrac{\gamma}{1 \mp \gamma} - \dfrac{1}{2f_{i(e)}}}. \qquad (3\text{-}13)$$

where

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

D_w = ball diameter, mm,

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

D_{pw} = pitch diameter of ball set, mm,

$f_i = \dfrac{r_i}{D_w}$,

$f_e = \dfrac{r_e}{D_w}$,

r_i = inner ring groove radius, mm,

r_e = outer ring groove radius, mm.

Substituting equation (3-12) into equation (3-7),

$$Q = 6{,}4762065 \times 10^{-10}\kappa\left(\frac{D_w}{2}\,\frac{E(\kappa)}{2 \pm \dfrac{\gamma}{1 \mp \gamma} - \dfrac{1}{2f_{i(e)}}}\right)^2 \sigma_{max}^3 . \qquad (3\text{-}14)$$

Substituting equations (3-12) and (3-14) into the following equation [10], furthermore exchanging Q for Q_{max},

$$C_{or} = \frac{1}{S} Z Q_{max} \cos\alpha , \qquad (3\text{-}15)$$

where

C_{or} = basic static radial load rating, N ,

Z = number of balls per row ,

Q_{max} = maximum normal force between rolling element and raceway, N ,

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,

α = nominal contact angle, ° ,

$$C_{or} = 0{,}2ZQ_{max}\cos\alpha \quad . \qquad (3\text{-}16)$$

Consequently,

$$C_{or} = 0{,}2 \times 6{,}4762065 \times 10^{-10} \times (4000)^3 \left(\frac{\sigma_{max}}{4000}\right)^3 \kappa \times \frac{1}{4}\left(\frac{E(\kappa)}{2 \pm \frac{\gamma}{1 \mp \gamma} - \frac{1}{2f_{i(e)}}}\right)^2 ZD_w^2\cos\alpha \ ,$$

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,

$$C_{or} = f_o iZD_w^2\cos\alpha \ , \qquad (3\text{-}17)$$

where

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

$$= 2{,}072\left(\frac{\sigma_{max}}{4000}\right)^3 \kappa \left(\frac{E(\kappa)}{2 \pm \frac{\gamma}{1 \mp \gamma} - \frac{1}{2f_{i(e)}}}\right)^2 . \qquad (3\text{-}18)$$