TECHNICAL REPORT

ISO/TR 1281-1

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Rolling bearings — Explanatory notes on ISO 281 —

Part 1: Basic dynamic load rating and basic rating life

Roulements — Notes explicatives sur l'ISO 281 —

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

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Foreword

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

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

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In exceptional circumstances, 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), it may decide by a simple majority vote of its participating members to publish a Technical Report. A Technical Report is entirely informative in nature and does not have to be reviewed until the data it provides are considered to be no longer valid or useful.

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ISO/TR 1281-1 was prepared by Technical Committee ISO/TC 4, *Rolling bearings*, Subcommittee SC 8, *Load ratings and life*.

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This first edition of ISO/TR 1281-1, together with the first edition of ISO/TR 1281-2, cancels and replaces the first edition of ISO/TR 8646:1985, which has been technically revised.

ISO/TR 1281 consists of the following parts, under the general title *Rolling bearings — Explanatory notes on ISO 281*:

— *Part 1: Basic dynamic load rating and basic rating life*

— *Part 2: Modified rating life calculation, based on a systems approach of fatigue stresses*

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Introduction

ISO/R281:1962

A first discussion on an international level of the question of standardizing calculation methods for load ratings of rolling bearings took place at the 1934 conference of the International Federation of the National Standardizing Associations (ISA). When ISA held its last conference in 1939, no progress had been made. However, in its 1945 report on the state of rolling bearing standardization, the ISA 4 Secretariat included proposals for definition of concepts fundamental to load rating and life calculation standards. This report was distributed in 1949 as document ISO/TC 4 (Secretariat-1)1, and the definitions it contained are in essence those given in ISO 281:2007 for the concepts "life" and "basic dynamic load rating" (now divided into "basic dynamic radial load rating" and "basic dynamic axial load rating").

In 1946, on the initiative of the Anti-Friction Bearing Manufacturers Association (AFBMA), New York, discussions of load rating and life calculation standards started between industries in the USA and Sweden. Chiefly on the basis of the results appearing in Reference [1], an AFBMA standard, *Method of evaluating load ratings of annular ball bearings*, was worked out and published in 1949. On the same basis, the member body for Sweden presented, in February 1950, a first proposal to ISO, "Load rating of ball bearings" [doc. ISO/TC 4/SC 1 (Sweden-1)1].

In view of the results of both further research and a modification to the AFBMA standard in 1950, as well as interest in roller bearing rating standards, in 1951, the member body for Sweden submitted a modified proposal for rating of ball bearings [doc. ISO/TC 4/SC 1 (Sweden-6)20] as well as a proposal for rating of roller bearings [doc. ISO/TC 4/SC 1 (Sweden-7)21].

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Load rating and life calculation methods were then studied by ISO/TC 4, ISO/TC 4/SC 1 and ISO/TC 4/WG 3 at 11 different meetings from 1951 to 1959. Reference [2] was then of considerable use, serving as a major basis for the sections regarding roller bearing rating.

The framework for the Recommendation was settled at a TC 4/WG 3 meeting in 1956. At the time, deliberations on the draft for revision of AFBMA standards were concluded in the USA and ASA B3 approved the revised standard. It was proposed to the meeting by the USA and discussed in detail, together with the Secretariat's proposal. At the meeting, a WG 3 proposal was prepared which adopted many parts of the USA proposal.

In 1957, a Draft Proposal (document TC 4 N145) based on the WG proposal was issued. At the WG 3 meeting the next year, this Draft Proposal was investigated in detail, and at the subsequent TC 4 meeting, the adoption of TC 4 N145, with some minor amendments, was concluded. Then, Draft ISO Recommendation No. 278 (as TC 4 N188) was issued in 1959, and ISO/R281 accepted by ISO Council in 1962.

ISO 281/1:1977

In 1964, the member body for Sweden suggested that, in view of the development of imposed bearing steels, the time had come to review ISO/R281 and submitted a proposal [ISO/TC 4/WG 3 (Sweden-1)9]. However, at this time, WG 3 was not in favour of a revision.

In 1969, on the other hand, TC 4 followed a suggestion by the member body for Japan (doc. TC 4 N627) and reconstituted its WG 3, giving it the task of revising ISO/R281. The AFBMA load rating working group had at this time started revision work. The member body for the USA submitted the Draft AFBMA standard, *Load ratings and fatigue life for ball bearings* [ISO/TC 4/WG 3 (USA-1)11], for consideration in 1970 and *Load ratings and fatigue life for roller bearings* [ISO/TC 4/WG 3 (USA-3)19] in 1971.

In 1972, TC 4/WG 3 was reorganized and became TC 4/SC 8. This proposal was investigated in detail at five meetings from 1971 to 1974. The third and final Draft Proposal (doc. TC 4/SC 8 N23), with some amendments, was circulated as a Draft International Standard in 1976 and became ISO 281-1:1977.

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

Furthermore, supplementary background information regarding the derivation of mathematical expressions and factors given in ISO 281-1:1977 was published, first as ISO 281-2, *Explanatory notes*, in 1979; however, TC 4/SC 8 and TC 4 later decided to publish it as ISO/TR 8646:1985.

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Rolling bearings — Explanatory notes on ISO 281 —

Part 1: **Basic dynamic load rating and basic rating life**

1 Scope

This part of ISO/TR 1281 gives supplementary background information regarding the derivation of mathematical expressions and factors given in ISO 281:2007.

2 Normative references

The following referenced documents are indispensable for the application 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.

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

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3 Symbols

		Clause
A	constant of proportionality	7
A_1	constant of proportionality determined experimentally	4
B_1	constant of proportionality determined experimentally	4
C_1	basic dynamic radial load rating of a rotating ring	4, 5
C_2	basic dynamic radial load rating of a stationary ring	4, 5
C_a	basic dynamic axial load rating for thrust ball or roller bearing	4, 6
C_{a1}	basic dynamic axial load rating of the rotating ring of an entire thrust ball or roller bearing	4
C_{a2}	basic dynamic axial load rating of the stationary ring of an entire thrust ball or roller bearing	4
C_{ak}	basic dynamic axial load rating as a row k of an entire thrust ball or roller bearing	4
C_{a1k}	basic dynamic axial load rating as a row k of the rotating ring of thrust ball or roller bearing	4
C_{a2k}	basic dynamic axial load rating as a row k of the stationary ring of thrust ball or roller bearing	4
C_e	basic dynamic load rating for outer ring	5
C_i	basic dynamic load rating for inner ring	5
C_r	basic dynamic radial load rating for radial ball or roller bearing	4, 5, 6

D_{pw}	pitch diameter of ball or roller set	4
D_{w}	ball diameter	4, 5
D_{we}	mean roller diameter	4
E_{o}	modulus of elasticity	4
F_{a}	axial load	5
F_{r}	radial load	4, 5
J_1	factor relating mean equivalent load on a rotating ring to Q_{max}	4, 5
J_2	factor relating mean equivalent load on a stationary ring to Q_{max}	4, 5
J_{a}	axial load integral	5
J_{r}	radial load integral	4, 5
L	bearing life	7
L_{10}	basic rating life	6, 7
L_{we}	effective contact length of roller	4
$L_{\mathrm{we}k}$	L_{we} per row k	4
N	number of stress applications to a point on the raceway	4
P_{a}	dynamic equivalent axial load for thrust bearing	5, 6
P_{r}	dynamic equivalent radial load for radial bearing	5, 6
P_{r1}	dynamic equivalent radial load for the rotating ring	5
P_{r2}	dynamic equivalent radial load for the stationary ring	5
Q	normal force between a rolling element and the raceways	4, 6
Q_C	rolling element load for the basic dynamic load rating of the bearing	4, 6
Q_{C_1}	rolling element load for the basic dynamic load rating of a ring rotating relative to the applied load	4, 5
Q_{C_2}	rolling element load for the basic dynamic load rating of a ring stationary relative to the applied load	4, 5
Q_{max}	maximum rolling element load	4, 5
S	probability of survival, reliability	4, 7
V	volume representative of the stress concentration	4
V_{f}	rotation factor	5
X	radial load factor for radial bearing	5
X_{a}	radial load factor for thrust bearing	5
Y	axial load factor for radial bearing	5
Y_{a}	axial load factor for thrust bearing	5
Z	number of balls or rollers per row	4, 5
Z_k	number of balls or rollers per row k	4
a	semimajor axis of the projected contact ellipse	4
a_1	life adjustment factor for reliability	7
b	semiminor axis of the projected contact ellipse	4
c	exponent determined experimentally	4, 6
c_{c}	compression constant	5

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e	measure of life scatter, i.e. Weibull slope determined experimentally	4, 5, 6, 7
f_c	factor which depends on the geometry of the bearing components, the accuracy to which the various components are made, and the material	4
h	exponent determined experimentally	4, 6
i	number of rows of balls or rollers	4
l	circumference of the raceway	4
r	cross-sectional raceway groove radius	5
r_e	cross-sectional raceway groove radius of outer ring or housing washer	4
r_i	cross-sectional raceway groove radius of inner ring or shaft washer	4
t	auxiliary parameter	4
v	$J_2(0,5)/J_1(0,5)$	5
z_o	depth of the maximum orthogonal subsurface shear stress	4
α	nominal contact angle	4, 5
α'	actual contact angle	5
γ	$D_w \cos \alpha / D_{pw}$ for ball bearings with $\alpha \neq 90°$	4
	D_w / D_{pw} for ball bearings with $\alpha = 90°$	
	$D_{we} \cos \alpha / D_{pw}$ for roller bearings with $\alpha \neq 90°$	
	D_{we} / D_{pw} for roller bearings with $\alpha = 90°$	
ε	parameter indicating the width of the loaded zone in the bearing	5
η	reduction factor	4, 5
λ	reduction factor	4
μ	factor introduced by Hertz	4
ν	factor introduced by Hertz, or adjustment factor for exponent variation	4
σ_{max}	maximum contact stress	4
$\Sigma\rho$	curvature sum	4
τ_o	maximum orthogonal subsurface shear stress	4
φ_o	one half of the loaded arc	5

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4 Basic dynamic load rating

The background to basic dynamic load ratings of rolling bearings according to ISO 281 appears in References [1] and [2].

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

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

where

S is the probability of survival;

τ_o is the maximum orthogonal subsurface shear stress;

N is the number of stress applications to a point on the raceway;

V is the volume representative of the stress concentration;

z_o is the depth of the maximum orthogonal subsurface shear stress;

c, h are experimentally determined exponents;

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

For "point" contact conditions (ball bearings) it is assumed that the volume, V, representative of the stress concentration in Correlation (1) is proportional to the major axis of the projected contact ellipse, $2a$, the circumference of the raceway, l, and the depth, z_o, of the maximum orthogonal subsurface shear stress, τ_o:

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

Substituting Correlation (2) into Correlation (1):

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

"Line" contact was considered in References [1] and [2] to be approached under conditions where the major axis of the calculated Hertz contact ellipse is 1,5 times the effective roller contact length:

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

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

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

(for variable definitions, see 4.1).

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

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

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

$$z_o = \zeta b$$

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

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

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

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$$b = v \left(\frac{3\,Q}{E_0 \sum \rho} \right)^{1/3}$$

where

σ_{max} is the maximum contact stress;

t is the auxiliary parameter;

a is the semimajor axis of the projected contact ellipse;

b is the semiminor axis of the projected contact ellipse;

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

E_0 is the modulus of elasticity;

$\Sigma\rho$ is the curvature sum;

μ, v are factors introduced by Hertz.

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

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

where

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

D_w is the ball diameter;

γ is $D_w \cos \alpha / D_{pw}$;

in which

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

α is the nominal contact angle;

Z is the number of balls per row.

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

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

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

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

where

Q_{C_1} is the rolling element load for the basic dynamic load rating of a ring rotating relative to the applied load;

Q_{C_2} is the rolling element load for the basic dynamic load rating of a ring stationary relative to the applied load;

$J_r = J_r\,(0{,}5)$ is the radial load integral (see Table 3);

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

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

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

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

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

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$$C_r = 0{,}41 \frac{1{,}3}{4^{(2c+h-2)/(c-h+2)} \, 0{,}5^{3e/(c-h+2)}} A_1 \left[\frac{2r_i}{2r_i - D_w} \right]^{0{,}41} \frac{(1-\gamma)^{(1{,}59c+1{,}41h-5{,}82)/(c-h+2)}}{(1+\gamma)^{3e/(c-h+2)}} \gamma^{3/(c-h+2)} \times$$
$$\left[1 + \left\{ 1{,}04 \left[\frac{r_i}{r_e} \left(\frac{2r_e - D_w}{2r_i - D_w} \right) \right]^{0{,}41} \left(\frac{1-\gamma}{1+\gamma} \right)^{(1{,}59c+1{,}41h+3e-5{,}82)/(c-h+2)} \right\}^{(c-h+2)/3} \right]^{-3/(c-h+2)} \times \tag{10}$$
$$(i\cos\alpha)^{(c-h-1)/(c-h+2)} \, Z^{(c-h-3e+2)/(c-h+2)} \, D_w^{(2c+h-5)/(c-h+2)}$$

where

A_1 is the experimentally determined proportionality constant;

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

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

i is the number of rows of balls.

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

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

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

Consequently, the factor f_c is given by:

$$f_c = 0,41\lambda \frac{1,3}{4^{(2c+h-2)/(c-h+2)}\, 0,5^{3e/(c-h+2)}} A_1 \left(\frac{2r_i}{2r_i - D_w}\right)^{0,41} \frac{(1-\gamma)^{(1,59c+1,41h-5,82)/(c-h+2)}}{(1+\gamma)^{3e/(c-h+2)}} \gamma^{3/(c-h+2)} \times$$
$$\left[\!\!\left[1+\left\{1,04\left[\frac{r_i}{r_e}\left(\frac{2r_e - D_w}{2r_i - D_w}\right)\right]^{0,41}\left(\frac{1-\gamma}{1+\gamma}\right)^{(1,59c+1,41h+3e-5,82)/(c-h+2)}\right\}^{(c-h+2)/3}\right]\!\!\right]^{-3/(c-h+2)} \qquad (12)$$

Based on References [1] and [2], the following values were assigned to the experimental constants in the load rating equations:

$e = 10/9$

$c = 31/3$

$h = 7/3$

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

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

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

$$f_c = 0,089\, A_1\, 0,41\lambda \left(\frac{2r_i}{2r_i - D_w}\right)^{0,41} \frac{\gamma^{0,3}(1-\gamma)^{1,39}}{(1+\gamma)^{1/3}} \times$$
$$\left[\!\!\left[1+\left\{1,04\left(\frac{1-\gamma}{1+\gamma}\right)^{1,72}\left[\frac{r_i}{r_e}\left(\frac{2r_e - D_w}{2r_i - D_w}\right)\right]^{0,41}\right\}^{10/3}\right]\!\!\right]^{-3/10} \qquad (15)$$

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

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

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

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

As in 4.1, for thrust ball bearings with contact angle $\alpha \neq 90°$:

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

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