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Second edition
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Cranes — Design calculation for wheel/rail contacts and associated trolley track supporting structure —

Part 1: General

Appareils de levage à charge suspendue — Calcul de conception des contacts galets/rails et de la structure porteuse du chariot de roulement —

Partie 1: Généralités

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Contents

Page

Foreword

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The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular, the different approval criteria needed for the different types of ISO 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).

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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 96, *Cranes*, Subcommittee SC 10, *Design principles and requirements*.

This second edition cancels and replaces the first edition (ISO 16881-1:2005), which has been technically revised.

The main changes are as follows:

— improvements were made to Annex B (local stresses);

— tables were added to Annex C to cover American, Chinese, and Japanese steels.

A list of all parts in the ISO 16881 series can be found on the ISO website.

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

This document establishes requirements and gives guidance and design rules that reflect the state of the art in crane machine design. The rules represent good design practice that ensures that essential safety requirements are met and that the components have an adequate service life. Deviation from these rules can increase risk or reduce service life. However, new technical innovations and materials provide solutions that result in equal or improved safety and durability.

Cranes — Design calculation for wheel/rail contacts and associated trolley track supporting structure —

Part 1: General

1 Scope

This document specifies requirements for selecting the size of iron or steel wheels. It also presents formulae to determine local stresses in crane structures due to the effects of wheel loads.

This document covers requirements for steel and cast-iron wheels. It applies to metallic contacts only.

This document does not apply to roller bearings.

This document is used together with the classification of the ISO 4301 series and the loads and load combinations of the ISO 8686 series.

This document is based on the limit state method (see ISO 8686-1).

This document is for design purposes only. It is not a guarantee of actual performance.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 4301 (all parts), *Cranes — Classification*

ISO 4302, *Cranes — Wind load assessment*

ISO 4306-1, *Cranes — Vocabulary — Part 1: General*

ISO 6506-1, *Metallic materials — Brinell hardness test — Part 1: Test method*

ISO 8686 (all parts), *Cranes — Design principles for loads and load combinations*

ISO 11031, *Cranes — Principles for seismically resistant design*

ISO 12100, *Safety of machinery — General principles for design — Risk assessment and risk reduction*

ISO 12488-1, *Cranes — Tolerances for wheels and travel and traversing tracks — Part 1: General*

ISO 20332, *Cranes — Proof of competence of steel structures*

3 Terms and definitions

3.1 General

For the purposes of this document, the terms and definitions given in ISO 4306-1, ISO 12100 and the following apply.

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

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

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

3.2 Symbols and abbreviations

For the purposes of this document, the symbols and abbreviations given in Table 1 apply.

Table 1 — Symbols and abbreviations

Symbols, abbreviations	Description
b	load-bearing width
b_r, b_w	effective contact widths of rail and wheel
D_w	wheel diameter
E_m	equivalent modulus of elasticity
E_r	modulus of elasticity of the rail or track
E_w	modulus of elasticity of the wheel
F	wheel load
$F_{Rd,f}$	limit design contact force for fatigue
$F_{Rd,s}$	limit design contact force
$F_{Sd,f}$	design contact force for fatigue
$F_{Sd,f,i}$	design contact force in contact i
$F_{Sd,s}$	design contact force
F_u	reference contact force
f_f	factor of further influences in fatigue
f_{f1}	decreasing factor for edge pressure in fatigue
f_{f2}	decreasing factor for non-uniform pressure distribution in fatigue
f_{f3}	decreasing factor for skewing in fatigue
f_{f4}	materials factor in fatigue
f_{f5}	decreasing factor for driven wheels in fatigue
f_y	yield point
f_1	decreasing factor for edge pressure
f_2	decreasing factor for non-uniform pressure distribution
HBW	Brinell hardness
HB*	unit-consistent hardness
i	index of one rolling contact with $f_{sd,f,i}$
i_D	number of rolling contacts at reference point
i_{tot}	total number of rolling contacts during the useful life of a wheel, rail or track
m	exponent for wheel/rail contacts
k_c	contact force spectrum factor
r_k	radius of the rail surface or the second wheel radius
r_3	radius of the edge
s_c	contact force history parameter
S_C	classes of contact force history parameter s_c
w	width of projecting non-contact area
z_{mp}, z_{ml}	depth of point of maximum shear for point or line contact
α	skewing angle

Table 1 *(continued)*

Symbols, abbreviations	Description
α_g	part of the skewing angle α due to the slack of the guide
α_t	part of the skewing angle α due to tolerances
α_w	part of the skewing angle α due to wear
γ_{cf}	contact resistance factor for fatigue
γ_m	general resistance coefficient; γ_m=1,1
γ_n	risk coefficient
γ_p	partial safety factors
ν	radial strain coefficient (ν = 0,3 for steel)
ν_c	relative total number of rolling contacts
ϕ	dynamic factors (see ISO 8686 series)

4 General

4.1 General principles

The proof of competence for static strength and fatigue strength shall be fulfilled when selecting a wheel and rail combination. In the proof of competence for static strength, the material properties of the weaker party (wheel or rail) shall be applied. The proof of competence for fatigue strength, or rolling contact fatigue (RCF), shall be conducted separately to each party, applying its specific material property and number of rolling contacts.

The proof shall be applied to all arrangements in cranes, where a wheel/rail type of rolling contact occurs, e.g. crane travel wheels, trolley traverse wheels, guide rollers and wheels/rollers supporting slewing structures. The term wheel is used throughout this document for the rolling party in a contact.

The proof of competence criteria in Clauses 5 and 6 are based upon Hertz pressure on the contact surface and the shear stress below the surface due to wheel/rail contact.

NOTE Guidance on the nominal dimensions of wheels is given in Annex E.

4.1.1 Unit-consistent hardness

Some of the formulae in this document refer to unit-consistent hardness (HB*), which is based on the Brinell hardness (HBW) and given as a value without units in line with ISO 6506-1. The unit of HB* shall match the unit of the modulus of elasticity used in the calculation. Using SI units, the unit-consistent hardness is given by:

$$HB^* = HBW \cdot \frac{N}{mm^2}$$

where

HB* is the unit-consistent hardness;

HBW is the value of the Brinell hardness.

EXAMPLE If HBW = 300, then HB* = 300 N/mm^2.

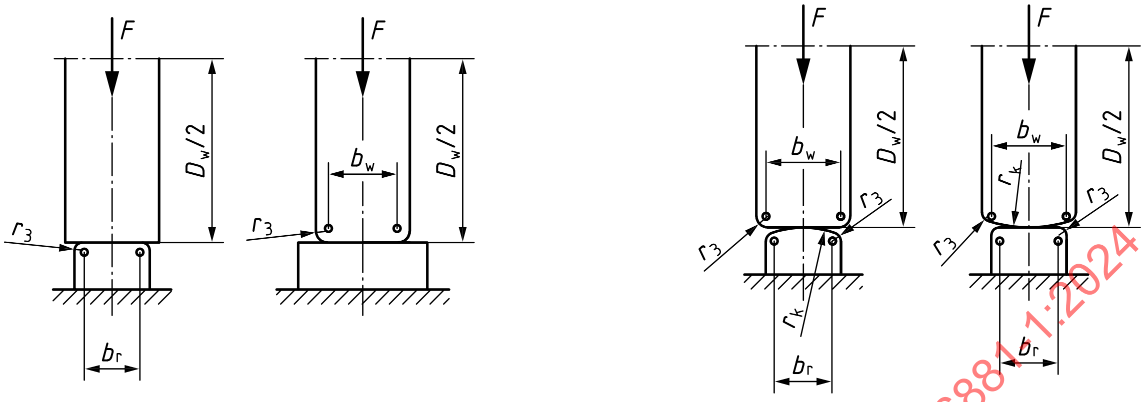
NOTE Annex D provides a table of hardness conversion.

4.2 Line and point contact cases

There are two main theoretical contact cases: a line contact and a point contact (see Figure 1).

Where the crown radius, r_k, in typical designs of crane wheels and rails is large in relation to the width of the wheel and rail, slight wear due to contact will occur for cranes. This document uses the line contact model to calculate Hertz pressure.

The conditions of point contact cases conforming to this assumption are stated in Figure 1 b).



a) Line contact **b) Point contact**

NOTE 1 The point contact in Figure 1 b) is assumed where the following condition applies:

$$5 \cdot min[b_r ; b_w] \le r_k \le 200 \cdot min[b_r ; b_w]$$

Where $r_k > 200 \cdot min[b_r ; b_w]$, the requirements for line contact shall be applied.

NOTE 2 The effective contact widths (b_w, b_r) are determined by deducting the effect of the corner radius equal to 2 × r_3 from the material width of the wheel/rail.

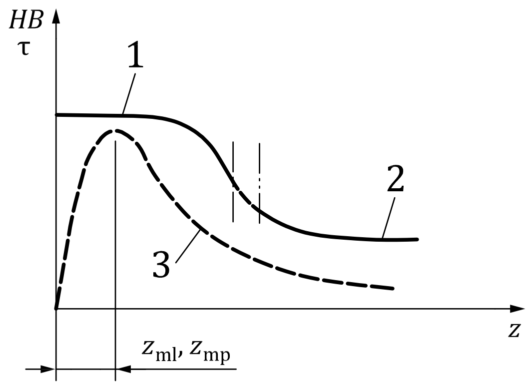
Figure 1 — Contact cases

4.3 Hardness profile below contact surface

The hardness shall extend deeper into the material than the depth of maximum shear, preferably twice this depth. The hardness value may be obtained using the ultimate strength of the material and the appropriate conversion tables for commonly used materials (see Annex C and Annex D).

Special care shall be taken with surface hardening and the depth zone to ensure that the hardness profile does not drop below the shear profile (see Figure 2).

The thickness of the surface-hardened layers should be determined according to ISO 18203.



Key

z	depth
z_{ml}, z_{mp}	depths of maximum shear stress
HB	unit-consistent hardness
1	hardness, the surface hardened zone
2	hardness, the natural hardness of the material
3	shear stress τ due to contact force

Figure 2 — Depth of hardness versus shear stress

The depth of the maximum shear for theoretical line contact cases shall be calculated using Formula (1).

$$z_{ml} = 0{,}50 \cdot \sqrt{F_{Sd0,s} \cdot \frac{\pi \cdot D_w \cdot \left(1 - \nu^2\right)}{b \cdot E_m}} \tag{1}$$

Theoretical point contact cases shall be calculated using Formula (2).

$$z_{mp} = 0{,}68 \cdot \sqrt[3]{\frac{F_{Sd0,s}}{E_m} \cdot \frac{1 - \nu^2}{\left(\frac{2}{D_w} + \frac{1}{r_k}\right)}} \tag{2}$$

where

$F_{Sd0,s}$ is the maximum design wheel/rail contact force within the load combinations A to C in accordance with the ISO 8686 series, taking into account the respective dynamic factors ϕ_i, but where all partial safety factors, γ_p, are set to 1. The most unfavourable load effects from possible positions of the mass of the hoist load and crane configurations shall be taken into account;

E_m is the equivalent modulus of elasticity (see 4.4);

b is the effective contact width of the rail (b_r) or the wheel (b_w) under consideration.

4.4 Equivalent modulus of elasticity

The equivalent modulus of elasticity shall be calculated using Formula (3), which also covers when the elastic modulus of the wheel and the rail are different:

$$E_m = \frac{2 \cdot E_w \cdot E_r}{E_w + E_r} \tag{3}$$

where

E_m is the equivalent modulus of elasticity;

E_w is the modulus of elasticity of the wheel;

E_r is the modulus of elasticity of the rail.

Values of the elastic moduli for selected materials are given in Table 2.

Table 2 — Values of elastic modulus

Wheel/rail material	Modulus of elasticity in N/mm^2
Steel	210 000
Cast iron	176 000
Steel/cast iron -combination	E_m = 191 500

5 Proof of static strength

5.1 General

The static strength of wheel/rail contacts shall be proven using Formula (4) for all relevant load combinations of the ISO 8686 series:

$$F_{Sd,s} \leq F_{Rd,s} \tag{4}$$

where

$F_{Sd,s}$ is the design contact force;

$F_{Rd,s}$ is the limit design contact force.

5.2 Design contact force

The design contact force $F_{Sd,s}$ of wheel/rail contacts shall be calculated for all relevant load combinations of the ISO 8686 series (eventually including the wind loads of ISO 4302 or the seismic loads of ISO 11031). This takes into account the respective dynamic factors ϕ_i, partial safety factors γ_p and where required the risk coefficient γ_n. The most unfavourable load effects from possible positions of the mass of the hoist load and crane configurations shall be taken into account.

5.3 Static limit design contact force

5.3.1 General

The static limit design contact force $F_{Rd,s}$ is specified as a force to cause a permanent radial deformation of 0,02 % of the wheel radius.

The static limit design contact force depends on:

— the material properties of the wheel and the rail (modulus of elasticity, yield stress and hardness);

— geometry (radii of wheel and rail);

— further influences (stiffness, edge effects).

5.3.2 Calculation of the limit design force

The static limit design contact force shall be calculated separately both for the wheel and the rail, either by Formula (5) or Formula (6). For the proof of competence in accordance with Formula (4), the value taken for $F_{Rd,s}$ shall be the smaller of the values obtained either for the wheel or the rail. The effective load-bearing width is the same in both calculations.

Formula (5) applies to non-surface hardened materials only, e.g. materials as cast, forged, rolled or quenched and tempered.

$$F_{Rd,s} = \frac{(7 \cdot HB)^2}{\gamma_m} \cdot \frac{\pi \cdot D_w \cdot b \cdot \left(1 - v^2\right)}{E_m} \cdot f_1 \cdot f_2 \tag{5}$$

Formula (6) applies to surface hardened materials, e.g. flame or induction hardened, provided that surface hardness is equal to or greater than $HB = 0{,}6 \times f_y$, and the depth of the hardened layer meets the requirements of 4.3.

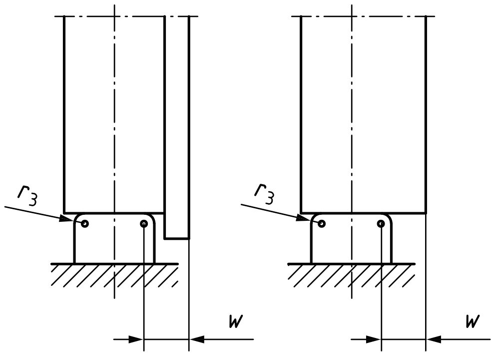
$$F_{Rd,s} = \frac{\left(4{,}2 \cdot f_y\right)^2}{\gamma_m} \cdot \frac{\pi \cdot D_w \cdot b \cdot \left(1 - v^2\right)}{E_m} \cdot f_1 \cdot f_2 \tag{6}$$

where

$F_{Rd,s}$	is the static limit design contact force;
E_m	is the equivalent modulus of elasticity;
v	is the radial strain coefficient (v = 0,3 for steel);
D_w	is the wheel diameter;
b	is the effective load-bearing width taken as $b = min[b_r\ ; b_w]$ (see Figure 1);
HB	is the unit-consistent hardness (see 4.1.1) based on the natural hardness of the material at the depth of maximum shear (see Annex C);
γ_m	is the general resistance coefficient, γ_m = 1,1;
f_y	is the yield stress of the material below the hardened surface, i.e. the natural yield stress of the material prior to the surface hardening process (see Annex C).
f_1	is the decreasing factor for edge pressure: for the line contact, see 5.3.3; for point contact cases, f_1 may be set to 1.
f_2	is the decreasing factor for non-uniform pressure distribution: for line contact, see 5.3.4; for point contact cases, f_2 may be set to 1.

5.3.3 Edge pressure

Formulae for the limit design contact force in the line contact case are derived from two bodies in contact, infinitely wide or of the same width. Factor f_1 (Table 3) introduces a correction to the limit design contact force for when the two bodies are of unequal width (see Figure 3). Where the rail is wider than the wheel, the radius of the edge (r_3) shall be taken as that of the wheel.



Key

r_3 radius of the edge of the narrower party (wheel or rail)

w width of the projecting non-contact area

Figure 3 — Edge pressure

Table 3 — Factor f_1 for edge pressure in line contact

Ratio r_3 / w	Factor f_1
$r_3 / w \leq 0{,}1$	0,85
$0{,}1 < r_3 / w < 0{,}8$	$[0{,}58 + 0{,}15 \cdot (r_3 / w)] / 0{,}7$
$r_3 / w \geq 0{,}8$	1,0

where

w is the width of the projecting non-contact area;

r_3 is the radius of the edge of the narrower party (wheel or rail).

5.3.4 Non-uniform pressure distribution

In the case of line contact, an ideal uniform distribution across the tread of the wheel is dependent on the rail fixing being elastic enough or its support and/or wheels having self-aligning suspension. Otherwise, the limit design contact force will be diminished due to pressure being distributed in non-uniform ways, due to deformation in the crane structure (e.g. the main girders bending) or misalignment in the rail. This effect is taken into account by factor f_2 (Table 4).

Table 4 — Factor f_2 for non-uniform pressure distribution in line contact

	Tolerance class of ISO 12488-1			
	1	2	3	4
Wheels with self-aligning suspension	1,0	1,0	0,95	0,9
Rail mounted on elastic support	0,95	0,9	0,85	0,8
Rail mounted on rigid support	0,9	0,85	0,8	0,7

6 Proof of fatigue strength

6.1 General

The proof of competence of the fatigue strength of wheels and rails shall be carried out in accordance with ISO 20332 and ISO 8686-1. The wheels and the rails shall have a specified design life that is proportionate to that of the related crane or hoist. The proof covers hazards related to rolling contact fatigue, i.e. surface cracking and pitting of wheels and rails.

The fatigue strength of wheel/rail contacts shall be proven by Formula (7) for each wheel and for all points on the rails under consideration.

$$F_{\mathrm{Sd,f}} \leq F_{\mathrm{Rd,f}} \tag{7}$$

where

$F_{\mathrm{Sd,f}}$ is the maximum design contact force for fatigue;

$F_{\mathrm{Rd,f}}$ is the limit design contact force for fatigue.

6.2 Design contact force

The design contact force $F_{\mathrm{Sd,f}}$ shall be calculated for the load combination A of the ISO 8686 series (regular loads). This includes the risk coefficient and all dynamic factors $\phi_i = 1$ and all partial safety factors $\gamma_{\mathrm{p}} = 1$. For the purpose of this document, the skewing forces acting on guide rollers shall be considered as regular loads.

6.3 Limit design contact force

6.3.1 Basic equation

The limit design contact force $F_{\mathrm{Rd,f}}$ shall be calculated separately both for the wheel and for the rail using Formula (8):

$$F_{Rd,f} = \frac{F_u}{\gamma_{cf} \cdot \sqrt[m]{s_c}} \cdot f_f \tag{8}$$

where

F_{u} is the reference contact force;

s_{c} is the contact force history parameter, calculated separately for the wheel and the rail;

γ_{cf} is the contact resistance factor for fatigue $\gamma_{\mathrm{cf}} = 1{,}1$;

f_{f} is the factor of further influences;

m is the exponent for wheel/rail contacts, m = 3,33.

6.3.2 Reference contact force

The limit design contact force of a wheel or rail subjected to rolling contact fatigue is characterized by the reference contact force F_{u} which represents the fatigue strength under $6{,}4 \times 10^6$ rolling contacts under constant contact force and a probability of survival of 90 % (i. e. avoiding cracks, pitting).

The reference contact force shall be calculated separately both for the and for the rail. The material property used in the calculation shall be specific for the party calculated, either wheel or rail. The effective load-bearing width is the same in both calculations.

Formula (9) applies to non-surface hardened materials only, e.g. materials as cast, forged, rolled or quenched and tempered.

$$F_u = (3{,}0 \cdot \mathrm{HB})^2 \cdot \frac{\pi \cdot D_w \cdot b \cdot \left(1 - v^2\right)}{E_m} \tag{9}$$

Formula (10) applies to surface hardened materials, e.g. flame or induction hardened, provided surface hardness is equal to or greater than $\mathrm{HB} = 0{,}6 \cdot f_y$, and the depth of the hardened layer meets the requirements of 4.3.

$$F_u = \left(1{,}8 \cdot f_y\right)^2 \cdot \frac{\pi \cdot D_w \cdot b \cdot \left(1 - v^2\right)}{E_m} \tag{10}$$

where

E_m is the equivalent modulus of elasticity;

v is the radial strain coefficient (v = 0,3 for steel);

D_w is the wheel diameter;

b is the effective load-bearing width taken as $b = min[b_r\ ; b_w]$ (see Figure 1);

HB is the unit-consistent hardness, based on the natural hardness of the material, at the depth of maximum shear (see Annex C);

f_y is the yield stress of the material below the hardened surface, i.e. the natural yield stress of the material prior to the surface hardening process (see Annex C).

6.3.3 Contact force history parameter

Like the stress history parameter (see ISO 20332), the contact force history parameter shall be calculated using Formula (11):

$$s_c = k_c \cdot v_c \tag{11}$$

where

k_c is the contact force spectrum factor;

v_c is the relative total number of rolling contacts.

The contact force history parameter describes the fatigue effect of the specified use in terms of rolling contacts in a particular wheel/rail pair.

6.3.4 Contact force spectrum factor

The contact force spectrum factor k_c shall be calculated by Formula (12):

$$k_c = 1 / i_{tot} \cdot \sum_{i=1}^{i_{tot}} \left(\frac{F_{Sd,f,i}}{F_{Sd,f}}\right)^m \tag{12}$$

where

i is the index of a rolling contact with $F_{Sd,f,i}$;

i_{tot} is the total number of rolling contacts during the design life of the wheel or rail;

$F_{Sd,f,i}$ is the design contact force for fatigue in a contact i;

$F_{Sd,f}$ is the maximum of all forces $F_{Sd,f,i}$;

m is the exponent for wheel/rail contacts, m = 3,33.

NOTE Formula (12) implies that the rolling contacts are counted individually.

6.3.5 Counting of rolling contacts

The total number of rolling contacts shall be calculated separately for wheel and for rail. For a wheel, one revolution is equivalent to one rolling contact. For a selected point on the rail, any wheel passing over represents one rolling contact. Where the wheel is not rolling but the load is fluctuating in cycles, one load cycle shall be considered one rolling contact.

Formulae (13) and (14) show the calculation for the total number of rolling contacts during the design life of the wheel or rail, i_{tot}, for a work cycle comprising a two-way motion over the point of the rail under consideration, i.e. a work cycle with the laden crane or trolley passing over the point in one direction and unladen on the return part of the work cycle.

For a running wheel:

$$i_{tot} = \frac{1}{l_w} \cdot \frac{2 \cdot \bar{x} \cdot C}{\pi \cdot D_w} \tag{13}$$

where

$\bar{x}$ is the average displacement of the related crane motion (see ISO 4301-1);

C is the total number of working cycles during the design life of the crane (see ISO 4301-1);

l_w is the design number of wheels used during the design life of the crane (i.e. the number of wheel replacements + 1);

D_w is the wheel diameter.

For a point on the rail with wheels passing over:

$$i_{tot} = 2 \cdot n_w \cdot C \tag{14}$$

where n_w is the total number of wheels of the crane passing over the point under consideration on the particular rail.

6.3.6 Relative total number of rolling contacts

The relative total number of rolling contacts, ν_c, is calculated using Formula (15):

$$\nu_c = \frac{i_{tot}}{i_D} \tag{15}$$

where

i_{tot} is the total number of rolling contacts during the design life of the wheel or rail;

i_D is the number of rolling contacts at reference point, $i_D = 6{,}4 \cdot 10^6$.

6.3.7 Classification of contact force history parameter

In the proof of competence calculations for a particular use specified in accordance with ISO 4301-1, the contact force history parameter shall be determined by Formula (11).

Wheels and rails may be assigned to classified sets of values of contact force history parameters. Table 5 shows a series of parameters and the symbols of the related classes. Where classification is referred to, compatibility between the selected S_c class and the specified use shall be shown in the proof of competence calculations.

A change in the wheel diameter changes the s_c parameter and may change the S_c class.

Table 5 — Classes S_c of contact force history parameter s_c

Class	S_c 0	S_c 1	S_c 2	S_c 3	S_c 4	S_c 5	S_c 6	S_c 7	S_c 8	S_c 9
s_c	0,008	0,016	0,032	0,063	0,125	0,25	0,5	1,0	2,0	4,0

6.4 Factors of further influences

6.4.1 Basic equation

The factor f_f calculated using Formula (16) takes into account further influences on the limit design contact force:

$$f_f = f_{f1} \cdot f_{f2} \cdot f_{f3} \cdot f_{f4} \qquad (16)$$

where f_{f1} to f_{f4} are the factors of influences as given in 6.4.2 to 6.4.5.

6.4.2 Edge pressure for fatigue

Due to the lateral movement of wheels, the edge pressure effect on the wider party (wheel or rail) may be neglected and the factor f_{f1} is set to 1. For the narrower party with the edge radius r_3 (see Figure 3) Formula (17) applies:

$$f_{f1} = f_1 \qquad (17)$$

where f_1 is the factor for edge pressure as given in 5.3.3.

6.4.3 Non-uniform pressure distribution for fatigue

For the proof of fatigue strength, the non-uniform pressure distribution may be neglected and f_{f2} is set to 1.

6.4.4 Skewing

A skewing wheel wears out the wheel and rail and shortens useful life. Wear increases proportionally in relation to the skew angle α. This effect is taken into account by factor f_{f3} calculated using Formulae (18).

$$f_{f3} = 1 \qquad \text{for } \alpha \leq 0{,}005 \text{ rad}$$

$$f_{f3} = \sqrt[3]{\frac{0{,}005}{\alpha}} \qquad \text{for } \alpha > 0{,}005 \text{ rad} \qquad (18)$$

where $\alpha = \alpha_g + \alpha_w + \alpha_t$ is the skew angle of the crane in radians, calculated in accordance with ISO 8686-1.

The part of the skew angle due to tolerances α_t shall be chosen according to the tolerance class in Table 6.

Table 6 — Alignment angle of a single wheel

Alignment	Tolerance class of ISO 12488-1			
	1	2	3	4
α_t [rad]	0,001 5	0,002 5	0,003 5	0,004 5

6.4.5 Mechanical drive factor

In an unclean environment, the mechanical abrasion effects on the driven wheels shall be taken into account by factor f_{f4} calculated using Formulae (19):

$$f_{f4} = 0{,}95 \text{ for driven wheels in an unclean environment,} \tag{19}$$

$$f_{f4} = 1{,}0 \text{ for non-driven wheels or wheels in a clean environment.}$$

7 Determination of local stresses due to wheel loads

Local stresses due to the wheel forces of a trolley or a crane may be determined according to Annexes A and B.

Local stresses shall be calculated using the wheel load F that can be either the static design contact force $F_{Sd,s}$ (see 5.2) or the fatigue design contact force $F_{Sd,f}$ (see 6.2, with all dynamic factors $\phi_i = 1$). These stresses, combined with overall stresses, shall not exceed the relevant limit design stress in accordance with ISO 20332.

Annex A
(informative)

Distribution of wheel load under rail

Local stresses in the welds or web rivets and webs of rail bearing beams that arise from wheel loads acting normally and transversely to the rail should be determined according to the rail and flange system. The method presented in this annex is valid when the web and rail alignment meets the tolerances of ISO 12488-1. When the tolerances are exceeded, the resulting bending moments should be taken into account.

Unless a more accurate calculation is made, the local vertical stress in a web or in the upper welds of girders caused by a wheel load should be calculated with Formula (A.1) when the rail is supported immediately by the upper flange:

$$\sigma_Z = 0{,}32 \cdot \frac{F}{t_c} \cdot \sqrt[3]{\frac{t_w}{J_c}} \qquad \text{(A.1)}$$

where

F	is the wheel load (see 7);
t_w	is the web thickness (see Figure A.1);
t_c	is either the web thickness [see Figure A.1 a)], the reduced effective weld throat thickness 1,4 · a [see Figure A.1 b)], or the reduced effective weld throat thickness 0,7· a [see Figure A.1 c)];
J_c	is the moment of inertia of the section made up of a cross travel rail and a part of the flange plate (hatched surface, see Figure A.2).

J_c should normally be calculated using 90 % of the available section of the crane rail. This value may be adjusted as the function of the expected:

— number of cycles,

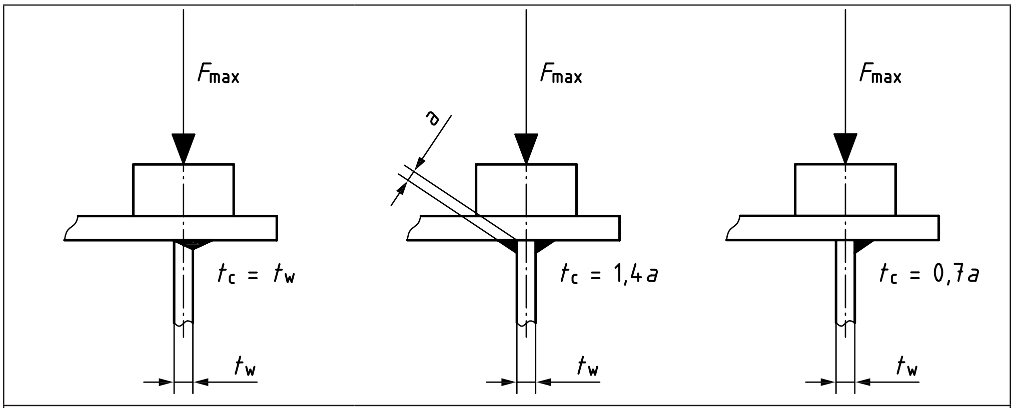
— load spectrum,

— alignment tolerances,

— material of the rail and wheel.

The wear limit used in the calculations should be stated in the operating instructions.

In the case of a clamped cross travel rail, J_c is calculated as the sum of the individual moments of inertia from the rail and the relevant part of the flange.



a) Full penetration weld | **b) Fillet welded from both sides** | **c) Fillet welded from one side**

Figure A.1 — Description of t_w and t_c

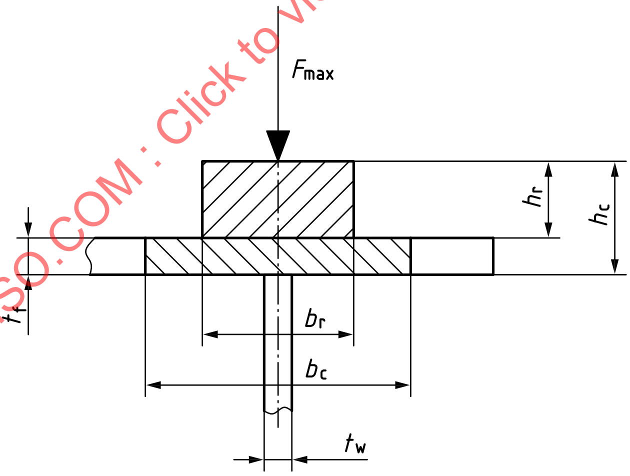
The effective flange width, b_c, is in this case calculated by Formula (A.2):

$$b_c = b_r + 0{,}4 \cdot (50 + 2 \cdot h_c) \qquad \text{(A.2)}$$

where

b_r is the rail width (mm);

h_c is the distance (mm) from top of the rail to the bottom of the flange.



Key

h_r rail height, expressed in mm

Figure A.2 — Marked area for J_c

Annex B
(informative)

Local stresses in wheel supporting flanges

B.1 General

When trolleys travel on the girder flanges of a girder, irrespective of the girder support arrangement, flange bending stresses occur as local stresses in the area of the point of application of the wheel load F.

In this annex, formulae and coefficients are given for two types of main girders:

— I-beam as main girder (see B.2);

— box girder as main girder (see B.3), with a partial penetration weld between the web and the bottom flange.

If the wheel loads, F (see Figure B.1), are not symmetrical, the local stresses should be calculated with the maximum wheel load and the relevant wheel location (distance i, see Figures B.1 and B.2). In addition to these flange bending stresses and the main stresses, torsion stresses due to the resulting, non-symmetrical load action should be calculated in the girder cross section.

In the proof of competence of static and fatigue strength according to ISO 20332, the local stresses calculated according to B.2 and B.3 are additionally multiplied by a factor of 0,75. In the static proof, the reduction of local stresses is based on the real bending capacity of the flange plate.

In the fatigue proof, the effect of local stress may be reduced by a factor set equal to 0,90 because the fatigue strength when a plate is bending is higher than when it is tense.

The proof of competence should be done separately for the local stresses as such and the local stresses combined with the maximum global bending stresses of the girder.

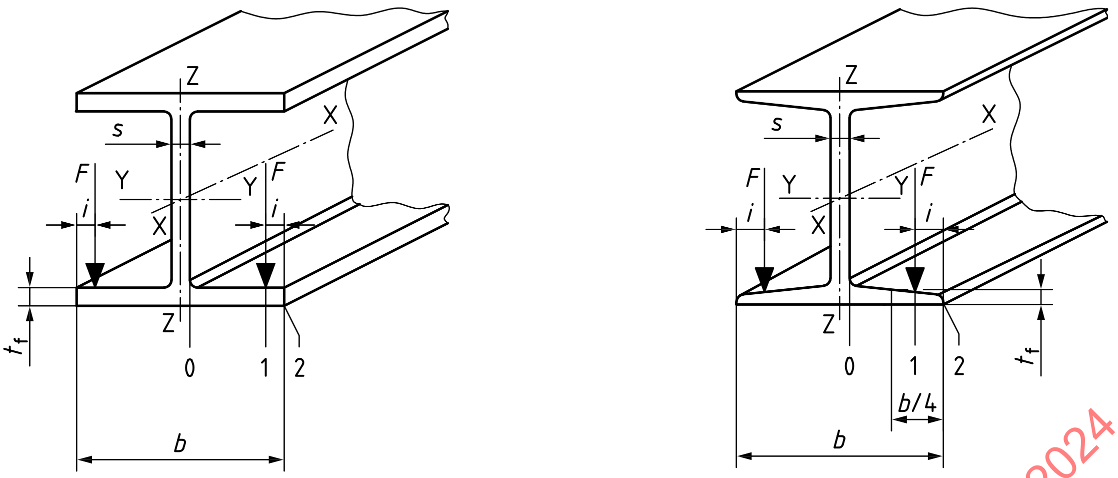
B.2 Local stresses in wheel supporting flanges — I-beam as main girder

These stresses act in the two directions X and Y as σ_{FX} and σ_{FY} [see Figure B.1 a) and b)].

The stresses are calculated using Formulae (B.1) and (B.2):

$$\sigma_{FX} = c_X(\lambda) \cdot \frac{F}{t_f^2} \tag{B.1}$$

$$\sigma_{FY} = c_Y(\lambda) \cdot \frac{F}{t_f^2} \tag{B.2}$$



a) I-beam with flanges of uniform thickness **b) I-beam with tapered flanges**

Key

0 stress at the transition web/flange
1 stress at the load application point
2 stress at the edge of the girder
s thickness of the web
b width of the flange
i distance from the girder edge to the point of wheel load application
t_f theoretical thickness of the flange (without tolerances and wear); for the girder with tapered flanges, t_f is taken at the point of wheel load application (see point 1, Figure B.1 b)
F maximum wheel load (see Clause 7) including the dynamic factors ϕ_i (set equal to 1,0 for the fatigue proof)

Figure B.1 — Calculation points for local stresses in I-beams

The coefficients $c_X(\lambda)$ and $c_Y(\lambda)$ are given in Table B.1 for stresses at the lower surface of the bottom flange at calculation points 0, 1, and 2. The stresses at the upper surfaces of the flange have the opposite sign.

The variable λ is calculated using Formula (B.3):

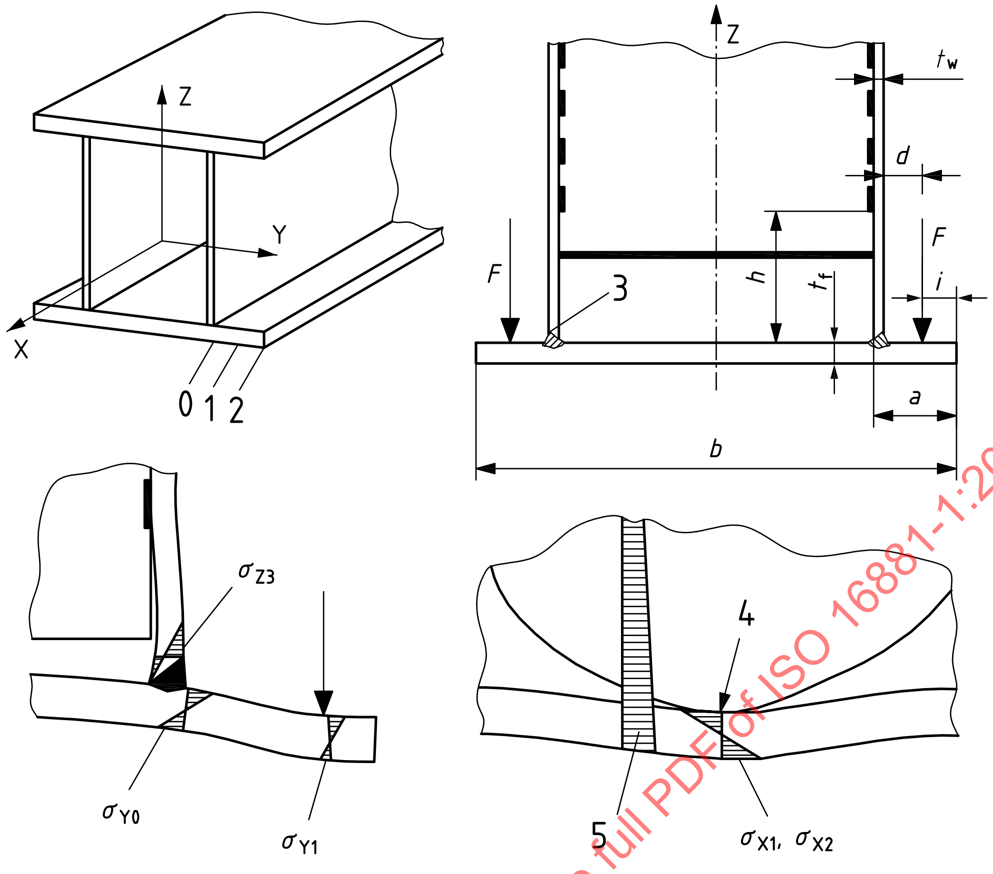
$$\lambda = \frac{i}{0{,}5 \cdot (b - s)} \tag{B.3}$$

Table B.1 — Coefficients of local stresses

Direction of stresses	I-beam with flanges of uniform thickness	I-beam with tapered flanges
Longitudinal bending stresses	$c_{X0} = 0{,}050 - 0{,}580 \cdot \lambda + 0{,}148 \cdot e^{3{,}015 \cdot \lambda}$ $c_{X1} = 2{,}230 - 1{,}490 \cdot \lambda + 1{,}390 \cdot e^{-18{,}33 \cdot \lambda}$ $c_{X2} = 0{,}730 - 1{,}580 \cdot \lambda + 2{,}910 \cdot e^{-6{,}00 \cdot \lambda}$	$c_{X0} = -0{,}981 - 1{,}479 \cdot \lambda + 1{,}120 \cdot e^{1{,}322 \cdot \lambda}$ $c_{X1} = 1{,}810 - 1{,}150 \cdot \lambda + 1{,}060 \cdot e^{-7{,}700 \cdot \lambda}$ $c_{X2} = 1{,}990 - 2{,}810 \cdot \lambda + 0{,}840 \cdot e^{-4{,}690 \cdot \lambda}$
Transverse bending stresses	$c_{Y0} = -2{,}110 + 1{,}977 \cdot \lambda + 0{,}007\,6 \cdot e^{6{,}53 \cdot \lambda}$ $c_{Y1} = 10{,}108 - 7{,}408 \cdot \lambda - 10{,}108 \cdot e^{-1{,}364 \cdot \lambda}$ $c_{Y2} = 0$	$c_{Y0} = -1{,}096 + 1{,}095 \cdot \lambda + 0{,}192 \cdot e^{-6{,}000 \cdot \lambda}$ $c_{Y1} = 3{,}965 - 4{,}835 \cdot \lambda - 3{,}965 \cdot e^{-2{,}675 \cdot \lambda}$ $c_{Y2} = 0$

B.3 Local stresses of a box girder with the wheel loads on the bottom flange

See Figure B.2.



Key

0, 1, 2 calculation points as in B.2

3 calculation point at the weld toe of the web

4 trolley wheel

5 global bending stress σ_{xg}

Figure B.2 — Symbols used in calculation of local stresses in box girder

Formulae and coefficients for the calculation of the local stresses at the bottom flange of a box girder are given in Table B.2. The symbols are presented in Figures B.2 and B.3. The formulae and coefficients are based on the results of finite element analyses and are approximations.

The signs of the stresses at points 0, 1 and 2 are valid at the bottom surface. The upper surface stress has the opposite sign.

Table B.2 — Formulae for stresses and coefficients

Point	Stress formula	Coefficients	Symbols and limits
0	$\sigma_{X0} = c_{X0} \cdot \frac{F}{t_f^2}$ $\sigma_{Y0} = c_{Y0} \cdot \frac{F}{t_f^2}$	$c_{X0} = 0{,}123 + 0{,}48 \cdot \lambda + 0{,}194 \cdot \lambda^2$ $- 0{,}5 \cdot \arctan(5 \cdot r_t - 1{,}375)$ $c_{Y0} = -1{,}3067 - 1{,}45 \cdot r_t$ $+ 0{,}5833 \cdot r_t^2 + 1{,}933 \cdot \lambda$	Valid for all formulae $r_t = t_w / t_f$ $2 \cdot a < b < 16 \cdot a$ $0{,}1 < i/a < 0{,}5$ $0{,}15 < r_t < 0{,}8$
1	$\sigma_{X1} = c_{X1} \cdot \frac{F}{t_f^2}$ $\sigma_{Y1} = c_{Y1} \cdot \frac{F}{t_f^2}$	$c_{X1} = 2{,}23 - 1{,}49 \cdot \lambda + 2 \cdot e^{-18{,}33 \cdot \lambda} (1 + 1{,}5 \cdot r_t) + 0{,}4 \cdot r_t^{2{,}5}$ $c_{Y1} = 0{,}33 \cdot (r_t - 1) + (1 + 2 \cdot r_t) \cdot \left[0{,}3 \cdot \lambda + 0{,}4 \cdot \sin\left(3{,}4 \cdot \lambda + 0{,}4 \cdot r_t^2\right)\right]$	
2	$\sigma_{X2} = c_{X2} \cdot \frac{F}{t_f^2}$ $\sigma_{Y2} = 0$	$c_{X2} = -0{,}95 + \frac{2{,}70}{(2 \cdot \lambda + 0{,}5)^{r_t^{0{,}333}}} + \left[1{,}2 \cdot (\lambda - 0{,}1)^{0{,}25} - 0{,}76\right] \cdot \left(\frac{0{,}2}{r_t}\right)^4$ $c_{Y2} = 0$	
3 At the web plate and at the weld toe	Stress at web is the sum of membrane (m) and bending (b) stress $\sigma_{Z3} = \sigma_{Z3m} + \sigma_{Z3b} =$ $\sigma_{Z3} = c_{Zm} \cdot \frac{F}{(d + t_f) \cdot t_w}$ $+ k_{Zh} \cdot c_{Zb} \cdot \frac{6 \cdot F \cdot d}{t_w^3 \cdot \left[1 + (2 \cdot r_t)^{-3}\right]}$	$c_{Zm} = 0{,}4 + 1{,}8 \cdot r_t^2$ $c_{Zb} = \left(0{,}01 + 0{,}0212 \cdot r_t^3\right) \cdot \left(0{,}125 \cdot \frac{b}{a} - 0{,}25\right)^{0{,}125}$ $k_{Zh} = 1 + \frac{k_{Z0}}{1 + 0{,}0004536 \cdot r_h^3}$ $k_{Z0} = 2 + 1{,}5 \cdot \sin[1{,}5 \cdot \pi \cdot (0{,}35 - r_t)]$ $+ 0{,}45 \cdot \sin[4 \cdot \pi \cdot (r_t - 0{,}5)]$	$r_h = \frac{h}{t_w}$ $4\ \text{mm} \le t_w \le 12\ \text{mm}$ $50 \cdot t_w < h$ $0 \le h < 50 \cdot t_w$

Dimensions in millimetres

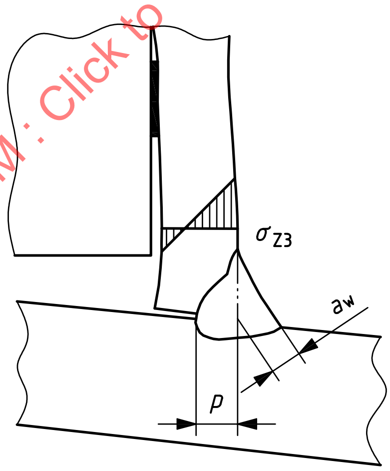


Figure B.3 — Symbols used in calculation of fillet weld

At point 3, the stress σ_{z3} (see Table B.2) also serves as the reference stress:

— for the proof of static strength in accordance with ISO 20332, the weld design stress $\sigma_{w,Sd}$ should be equal to σ_{z3};

— for the proof of fatigue strength, the maximum range of stress $\Delta\sigma_{w,Sd}$ should be equal to σ_{z3}.

The partial safety factors and dynamic load factors should be taken into account when calculating wheel forces as per ISO 8686 series.

At point 3, the stress σ_{z3} (see Table B.2) also serves as the reference stress for the weld in both static and fatigue proofs of competences, if the weld is at quality level C of ISO 5817 or better and meets the requirement $a_w \geq 0{,}5 \cdot t_w$.

For the fatigue proof of competence, the notch classes and the related conditions regarding the weld quality level and the size of penetration (p) and weld throat thickness (a_w) should be taken in accordance with the relevant constructional detail in ISO 20332.

Annex C
(informative)

Strength properties for a selection of wheel and rail materials

The values for ultimate strength, f_u, and design hardness, HBW, given in Table C.1 for European wheel and rail materials are conservative because they depend on the size of the rail/wheel under consideration. More accurate values are given in the relevant material standard.

Tables C.2, C.3 and C.4 provide values for wheel and rail materials commonly used in the USA, China and Japan respectively.

Table C.1 — European wheel and rail materials and their strength properties

Wheel materials					**Design hardness HBW**
Designation	**Standard**	**Material no.**	**Delivery state**	**Ultimate strength f_u (N/mm²)**	
GE300	EN 10293	1.0558	+N	520	155
EN-GJS 700-2	EN 1563	5.3300	as cast	700	225
25CrMo4	EN ISO 683-2	1.7218	+QT	650	190
34CrMo4	EN ISO 683-2	1.7220	+QT	700	210
42CrMo4	EN ISO 683-2	1.7225	+QT	750	225
33NiCrMoV14-5	EN 10250-3	1.6956	+QT	950	295
Wheel materials, surface hardened					**Minimum surface hardness HBW**
Designation	**Standard**	**Material no.**	**Delivery state**	**Design yield stress f_y (N/mm²)**	
42CrMo4	EN ISO 683-2	1.7225	+N, surface hardened	420	515 [a]
Rail materials					**Design hardness HBW**
Designation	**Standard**	**Material no.**	**Delivery state**	**Ultimate strength f_u (N/mm²)**	
S235 [b]	EN 10025-2	---	+N	350	110
S355 [b]	EN 10025-2	---	+N	450	155
S690QL	EN 10025-6	1.8928	+QT	710	225
C35E	EN ISO 683-1	1.1181	+N	550	155
C55	EN ISO 683-1	1.0535	+N	700	190
R260Mn	EN 13674-1	1.0624	+N	880	260

+N normalized
+QT quenched and tempered
[a] Hardness is specified based on the hardening process and required depth.
[b] Table values are valid for any of the quality grades JR, J0, J2 and K2.

Table C.2 — USA wheel and rail materials and their strength properties

Wheel materials					Design hardness HBW
Designation	**Standard**	**UNS Material no.**	**Delivery state**	**Ultimate strength** f_u (N/mm^2)	
4140	ASTM A29 ASTM A668	G41400	+NT	607	220
Wheel materials, surface hardened					**Minimum surface hardness** HBW
Designation	**Standard**	**UNS Material no.**	**Delivery state**	**Design yield stress** f_y (N/mm^2)	
4140	ASTM A29 ASTM A668	G41400	+QT	869	280
1070	ASTM A29	G10700	Rim Quenched	996	321
1055	ASTM A29	G10550	Case Hardened	690	615
1070	ASTM A29	G10700	Case Hardened	690	615
A504 Class A	ASTM A504		Rim Quench	747	255
A504 Class B	ASTM A504		Rim Quench	937	302
A504 Class C	ASTM A504		Rim Quench	996	321
A504 Class L	ASTM A504		Rim Quench	543	197
Rails					**Design hardness** HBW
Designation [a]	**Standard**	**Rail type**	**Delivery state**	**Ultimate strength** f_u (N/mm^2)	
ASCE 60, ASCE 80	ASTM A1	Standard carbon	As rolled	680	201
1055	ASTM A29				
1070	ASTM A29				
ASCE 85	ASTM A1	Standard carbon	As rolled	680	201
ASCE 60, ASCE 80	ASTM A1	High strength	Head hardened	920	277
ASCE 85	ASTM A1	High strength	Head hardened	1 060	321
ARA-A 90	ASTM-A1 ASTM A759	Standard carbon	As rolled	680	201
ARA-A 90	ASTM-A1 ASTM A759	High strength	Head hardened	1 060	321
ARA-B 100	ASTM-A1 ASTM A759	Standard carbon	As rolled	680	201
ARA-B 100	ASTM-A1 ASTM A759	High strength	Head hardened	1 002	300
104, 135, 171, 175	ASTM A1 ASTM A759	Standard carbon	Head hardened	1 002	300
104, 135, 171, 175	ASTM A1, ASTM A759	High strength	Fully heat treated	1 130	341

[a] Designation by weight in lb/yd. Light rails are 100 lb/yd or less. Heavy rails exceed 100 lb/yd. Common heavy rails designations are 104, 135, 171, 175 and MRS 87A.