INTERNATIONAL STANDARD

# ISO 16881-1

First edition
2005-05-15

# Cranes — Design calculation for rail wheels and associated trolley track supporting structure —

## Part 1: General

*Appareils de levage à charge suspendue — Calcul de conception des galets et de la structure de support du chariot de roulement —*

*Partie 1: Généralités*

Reference number
ISO 16881-1:2005(E)

© ISO 2005

Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS

Not for Resale

**PDF disclaimer**

This PDF file may contain embedded typefaces. In accordance with Adobe's licensing policy, this file may be printed or viewed but shall not be edited unless the typefaces which are embedded are licensed to and installed on the computer performing the editing. In downloading this file, parties accept therein the responsibility of not infringing Adobe's licensing policy. The ISO Central Secretariat accepts no liability in this area.

Adobe is a trademark of Adobe Systems Incorporated.

Details of the software products used to create this PDF file can be found in the General Info relative to the file; the PDF-creation parameters were optimized for printing. Every care has been taken to ensure that the file is suitable for use by ISO member bodies. In the unlikely event that a problem relating to it is found, please inform the Central Secretariat at the address given below.

© ISO 2005

All rights reserved. Unless otherwise specified, no part of this publication may be reproduced or utilized in any form or by any means, electronic or mechanical, including photocopying and microfilm, without permission in writing from either ISO at the address below or ISO's member body in the country of the requester.

ISO copyright office
Case postale 56 • CH-1211 Geneva 20
Tel. + 41 22 749 01 11
Fax + 41 22 749 09 47
E-mail copyright@iso.org
Web www.iso.org

Published in Switzerland

Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

# Contents

Page



© ISO 2005 – All rights reserved


Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

# 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.

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

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.

ISO 16881-1 was prepared by Technical Committee ISO/TC 96, *Cranes*, Subcommittee SC 9, *Bridge and gantry cranes*.

ISO 16881 consists of the following parts, under the general title *Cranes — Design calculation for rail wheels and associated trolley track supporting structure*:

— *Part 1: General*

The following parts are under preparation:

— *Part 2: Mobile cranes*

— *Part 3: Tower cranes*

— *Part 4: Jib cranes*

— *Part 5: Bridge and gantry cranes*


© ISO 2005 – All rights reserved

Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

## Introduction

This part of ISO 16881 establishes requirements and gives guidance and design rules that reflect the present state of the art in the field of crane machine design. The rules given represent good design practice that will ensure fulfilment of essential safety requirements and adequate service life of components. Deviation from these rules normally could lead to increased risks or reduction of service life, but it is acknowledged that new technical innovations, materials, etc. may enable new solutions that result in equal or improved safety and durability.

© ISO 2005 – All rights reserved

Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS

Not for Resale

# Cranes — Design calculation for rail wheels and associated trolley track supporting structure —

# Part 1: General

## 1 Scope

This part of ISO 16881 gives the requirements for the selection of the size for iron or steel wheels and presents the formulae for the local stresses in crane structures due to the effects of the wheel loads.

## 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.

ISO 4301-1, *Cranes and lifting appliances — Classification — Part 1: General*

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

ISO 8686-1, *Cranes — Design principles for loads and load combinations — Part 1: General*

ISO 8686-2, *Cranes — Design principles for loads and load combinations — Part 2: Mobile cranes*

ISO 8686-3, *Cranes — Design principles for loads and load combinations — Part 3: Tower cranes*

ISO 8686-4, *Cranes — Design principles for loads and load combinations — Part 4: Jib cranes*

ISO 8686-5, *Cranes — Design principles for loads and load combinations — Part 5: Overhead travelling and portal bridge cranes*

## 3 Terms and definitions

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

## 4 Requirements

### 4.1 Selection of rail wheels

#### 4.1.1 Rail wheel size

To determine the size of a rail wheel, the following checks shall be made:

a) verify that the wheel is capable of withstanding the maximum load to which it will be subjected;

b) verify that the wheel will allow the appliance to perform its normal duty without abnormal wear.

© ISO 2005 – All rights reserved

Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

These verifications are made by means of the following two equations:

$$\frac{P_{\text{max}}}{b \cdot D} \leqslant 1{,}9 P_{\text{L}} \tag{1}$$

$$\frac{P_{\text{mean}}}{b \cdot D} \leqslant P_{\text{L}} \cdot c_1 \cdot c_2 \tag{2}$$

where

$D$ is the wheel diameter, in millimetres;

$b$ is the useful width of the rail, in millimetres;

$P_{\text{L}}$ is a limiting pressure dependent upon the metal used for the wheel, in newtons per square millimetre (N/mm$^2$), see Table 1;

$c_1$ is a coefficient depending on the speed of rotation of the wheel, see Table 2;

$c_2$ is a coefficient depending on the group of the mechanism, see Table 3;

$P_{\text{max}}$ is the maximum load on the wheel resulting from load combinations A, B or C, including consideration of both dynamic and static test loadings (load combinations are defined in ISO 8686-1 to ISO 8686-5);

$P_{\text{mean}}$ is the higher mean load value resulting from Equation (3) when considering both load combinations A and B.

The mean wheel load takes into account variations of the wheel loading, including, where applicable, positional changes of the handled load in relation to the supporting wheels during a working cycle. Equation (3) gives an approximate value of the resultant cubic mean loading.

When the work process is well known, the cubic mean load can be calculated more accurately using the wheel loads resulting from the actual positions of the handled load. In this calculation, the maximum lifted load shall be used, coefficient $c_2$ taking into account the variation of the load.

### 4.1.2 Determining the mean load

In order to determine the mean loads, the procedure is to consider the maximum and minimum loads withstood by the wheel in the loading cases considered, i.e. with the appliance in normal duty but omitting the dynamic coefficients $\phi$ when determining $P_{\text{mean}}$. The values of $P_{\text{mean}}$ are determined by the Equation (3) in the load combinations A and B.

$$P_{\text{mean}} = \frac{P_{\text{min,A,B}} + 2 P_{\text{max,A,B}}}{3} \tag{3}$$

### 4.1.3 Determining the useful rail width $b$

For rails having a flat or slightly convex bearing surface, of total width, $l$, with rounded corners of radius, $r$, at each side (see Figure 1), the useful width, $b$, shall be calculated using Equation (4):

$$b = l - 2r \tag{4}$$

For rails or wheels with a slightly convex bearing surface, the limiting pressure $P_{\text{L}}$ may be increased by 10 %. This allows for the improved contact of a rail to the rolling motion of the wheel.


© ISO 2005 – All rights reserved
Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

In the case of a flat, tapered or convex wheel running on the bottom flange of a beam, the useful width is calculated by Equation (5):

$$b = w - r \tag{5}$$

where the wheel tread of width, $w$, and the corner radius $r$ are to be taken according to Figure 2. The wheel diameter, $D$, shall be taken as the mean diameter on the projected width ($w - r$).

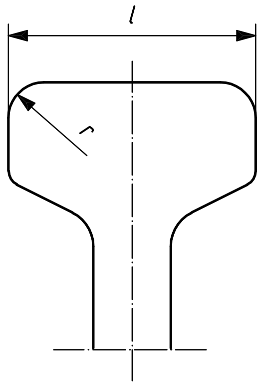



**Figure 1 — Rail dimensions**

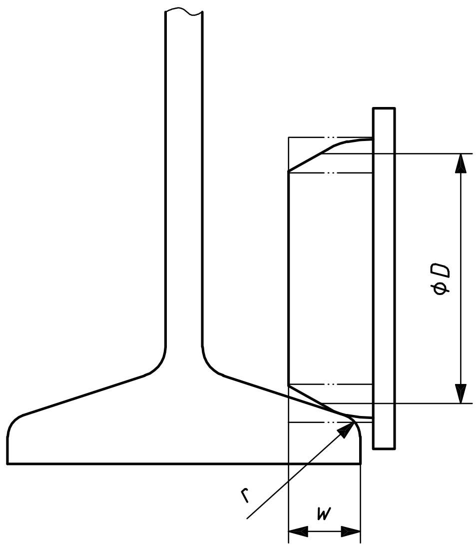



**Figure 2 — Dimensions of flange running wheel**

#### 4.1.4 Determining limiting pressure $P_L$

The value of $P_L$ is given in Table 1 as a function of the ultimate strength of the metal of which the rail wheel is made.

© ISO 2005 – All rights reserved
Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

**Table 1 — Values of $P_L$**

| Ultimate strength of metal used for rail wheel<br>$f_u$<br>N/mm$^2$ | $P_L$<br>N/mm$^2$ | Minimum ultimate strength for rail<br>N/mm$^2$ |
|---|---|---|
| > 500 | 5,00 | 350 |
| > 600 | 5,60 | 350 |
| > 700 | 6,50 | 510 |
| > 800 | 7,20 | 510 |
| > 900 | 7,80 | 600 |
| > 1 000 | 8,50 | 700 |

The qualities of metal correspond to cast, forged or rolled steels, and spheroidal graphite cast iron.

The hardening of the wheel tread at the depth of 0,01$D$ may be taken into account when selecting the value of $P_L$.

In the case of rail wheels with tyres, consideration must obviously be given to the quality of the tyre, which shall be sufficiently thick not to roll itself out.

### 4.1.5 Determining coefficient $c_1$

The values of $c_1$ depend on the speed of rotation of the wheel and are given in Table 2.

**Table 2 — Values of $c_1$**

| Wheel rotation speed<br>r/min | $c_1$ | Wheel rotation speed<br>r/min | $c_1$ | Wheel rotation speed<br>r/min | $c_1$ |
|---|---|---|---|---|---|
| 200 | 0,66 | 50 | 0,94 | 16 | 1,09 |
| 160 | 0,72 | 45 | 0,96 | 14 | 1,10 |
| 125 | 0,77 | 40 | 0,97 | 12,5 | 1,11 |
| 112 | 0,79 | 35,5 | 0,99 | 11,2 | 1,12 |
| 100 | 0,82 | 31,5 | 1 | 10 | 1,13 |
| 90 | 0,84 | 28 | 1,02 | 8 | 1,14 |
| 80 | 0,87 | 25 | 1,03 | 6,3 | 1,15 |
| 71 | 0,89 | 22,4 | 1,04 | 5 | 1,16 |
| 63 | 0,91 | 20 | 1,06 | | |
| 56 | 0,92 | 18 | 1,07 | | |

### 4.1.6 Determining coefficient $c_2$

Coefficient $c_2$ depends on the group classification of the mechanism and is given in Table 3.


© ISO 2005 – All rights reserved
Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

**Table 3 — Values of $c_2$**

| Group classification of mechanism | $c_2$ |
|---|---|
| M1 et M2 | 1,25 |
| M3 et M4 | 1,12 |
| M5 | 1,00 |
| M6 | 0,90 |
| M7 et M8 | 0,80 |

The formulae apply only to wheels whose diameters do not exceed 1,25 m. For larger diameters, experience shows that the permissible pressures between the rail and the wheel must be lowered. The use of wheels of greater diameter is not recommended.

NOTE The wheel selection method presented here is based on FEM 1.001-1988, Booklet 4, as revised in Booklet 9 in 1998. This method is based on the group classification of mechanisms (classes M1 to M8) that is equal to the classification of ISO 4301-1.

## 4.2 Determination of the class of mechanism of the travel wheel

The determination of the mechanism class (according to ISO 4301-1) of the rail wheel is made in general terms as follows.

a) The number of working cycles, $C$, is taken as the specified value or the upper limit of the given U-class.

b) The average displacement of the travel motion, $x_m$, is specified according to intended use.

c) The wheel load spectrum class number, $L_i$, shall be taken as specified or to be calculated from a given spectrum or description of work cycles.

d) Total travel distance is $L = 2Cx_m$.

e) Running time $T = L/v_t$, where $v_t$ is the normal travel speed of the motion (usually, the maximum speed).

f) Operation time class number is calculated as $T_j = 1 + \text{Int}\,[\log (T/200{,}01\ \text{h})/\log (2)]$.

EXAMPLE 1 $T = 3\,200$, $T_j = 1 + \text{Int}\,[\log (3\,200/200{,}01)/0{,}693\,147] = 1 + \text{Int}\,(3{,}999\,9) = 4$; $\Rightarrow$ classe $T_4$.

EXAMPLE 2 $T = 3\,201$, $T_j = 1 + \text{Int}\,[\log (3\,201/200{,}01)/0{,}693\,147] = 1 + \text{Int}\,(4{,}000\,4) = 5$; $\Rightarrow$ classe $T_5$.

g) The mechanism class number is calculated as $m_m = L_i + T_j - 2$.

$T_j$, $L_i$, and $m_m$ shall be understood as integer numbers, e.g. 5, or as the corresponding symbol, $T_5$ .

## 4.3 Determination of local stresses due to wheel loads

The local stresses due to the wheel forces of a crab or a crane may be determined according to Annexes A and B.

When the permissible stress method is used, the local stresses according to Annexes A and B, combined with the global stresses due to the rated loads, shall not exceed the permissible stresses.

When the limit state method is used, the local stresses shall be calculated with the loads multiplied by the relevant partial load factors, $\gamma_p$. These stresses, combined with the global stresses, shall not exceed the yield stress divided by the material factor, $\gamma_m$.

Factors $\gamma_p$ and $\gamma_m$ shall be taken according to the relevant part(s) of ISO 8686.



Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

# Annex A
(informative)

# Distribution of wheel load under rail

The local stresses in the welds or web rivets and webs of rail bearing beams which arise from wheel loads acting normally and transversely to the rail shall be determined in accordance with 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 shall 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 due to a wheel load shall be calculated with the following Equation when the rail is supported immediately by the upper flange:

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

where

$F_{\text{max}}$ is the maximum wheel load including the amplification factors, $\phi_i$

$t_w$ is the web thickness (see Figure A.1);

$t_c$ is the web thickness [see Figure A.1 a)], or

the reduced weld throat thickness 1,4$a$ [see Figure A.1 b)], or

the reduced weld throat thickness 0,7$a$ [see Figure A.1 c)].

$J_c$ is the moment of inertia of the section made up of cross travel rail and a part of flange plate (hatched surface, see Figure A.2)

$J_c$ shall 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, and

— material of the rail and wheel.

The wear limit used in the calculations shall 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.


© ISO 2005 – All rights reserved
Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

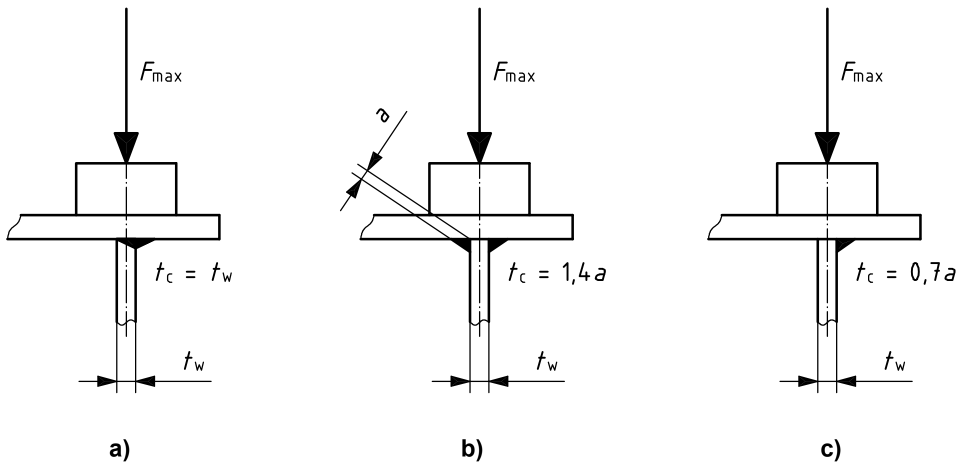



**Figure A.1 — Description of $t_w$ and $t_c$**

The effective flange width, $b_c$, is in this case calculated as

$$b_c = b_r + 0{,}4\ (50\ \text{mm} + 2h_c)$$

where

$b_r$ is the rail width;

$h_r$ is the rail height;

$h_c$ is the distance from top of the rail to the bottom of the flange.

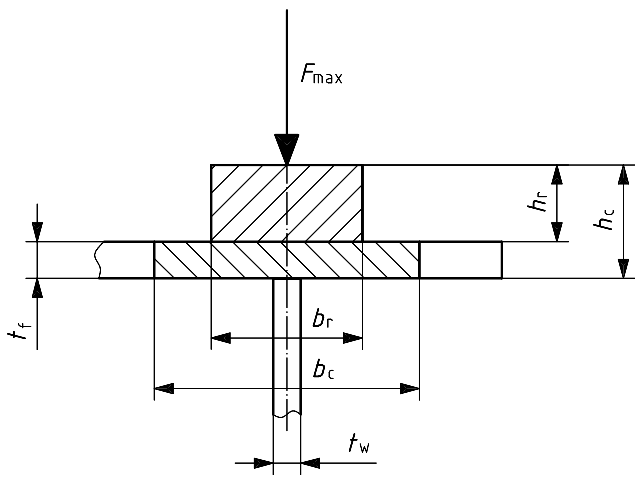



**Figure A.2 — Marked area for $J_c$**



© ISO 2005 – All rights reserved
Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

# Annex B
(informative)

# Local stresses in wheel-supporting flanges

## B.1 General

When crabs travel on the girder flanges of a girder, irrespective of the girder support arrangement, local flange bending stresses occur 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:

— main girder as I-beam (B.2);

— main girder as box girder (B.3).

When determining the reference stresses according to ISO 8686-1 to ISO 8686-5 and in the fatigue stress verification, the local stresses shall be superimposed on the global stresses. Attention must be paid to the plus and minus signs. In the load combinations A, B and C (see Table 1) and in the fatigue strength verification (load combinations A), the local stresses in the plates and full penetration welds shall be multiplied by 0,75 before superimposing with the global stresses. In fatigue analysis, when coefficient 0,75 is used, the combined stresses shall be compared to the tension fatigue strength of the weld joint or the detail.

The local stresses can be reduced by factor 0,75 because of the extra plastic bending capacity of the flange plate or extra plastic tension capacity of the web.

In fatigue analysis the effect of the local stress can be reduced, because the fatigue strength in bending of a plate is typically 30 % to 60 % higher than in tension, for the same joint or detail.

If the wheel loads, $F$, are not symmetrical, the local stresses are calculated with the maximum wheel load and the relevant distance, $i$. In addition to these flange bending stresses and the main stresses, torsion stresses from the resulting non-symmetrical load application point must be calculated in the girder cross section.

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

These stresses act in the two directions X and Y as $\sigma_{F\mathrm{X}}$ and $\sigma_{F\mathrm{Y}}$ (see Figure B.1).

The stresses are calculated with the help of the following equations

$$\sigma_{F\mathrm{X}} = c_{\mathrm{X}}(\lambda)\ \frac{F}{t_{\mathrm{f}}^2} \tag{B.1}$$

$$\sigma_{F\mathrm{Y}} = c_{\mathrm{Y}}(\lambda)\ \frac{F}{t_{\mathrm{f}}^2} \tag{B.2}$$

The numerical indexes have the following meanings:

0 stress at the transition web/flange;

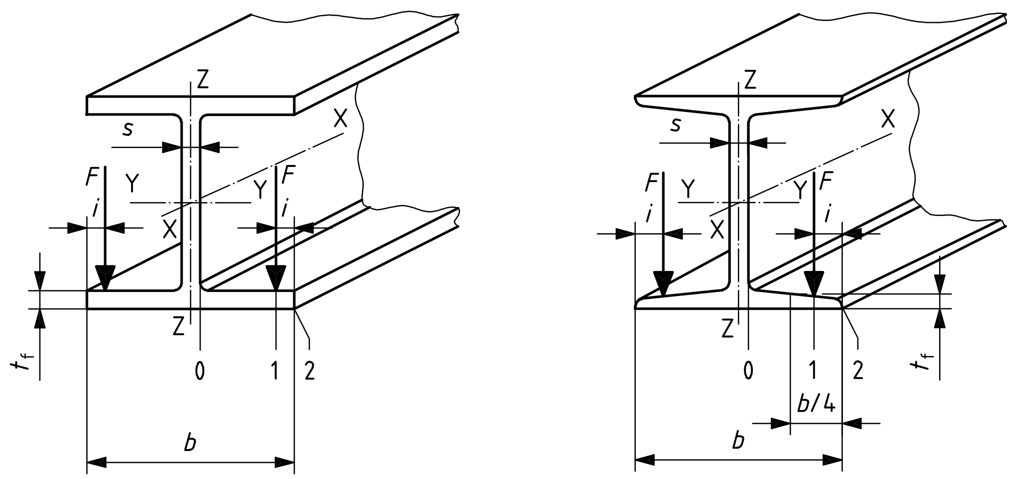
1 stress at the load application point;

2 stress at the edge of the girder.



© ISO 2005 – All rights reserved

Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale




**a) I-beam with parallel flanges** **b) I-beam with inclined flanges**

**Figure B.1**

The variables $F$, $t_f$, $i$ and $\lambda$ have the following meanings:

$F$ is the maximum wheel load including the amplification factors $\phi_i$

$t_f$ is the theoretical thickness of the flange (without tolerances and wear); for the girder with inclined flanges, $t_f$ is taken at the point of load application, see point 1, Figure B.1 b);

$i$ is the distance from the girder edge to the point of load application;

$b$ is the width of the flange;

$s$ is the thickness of the web;

$\lambda$ is calculated from the formula.

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

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

**Table B.1 — Coefficients of local stresses**

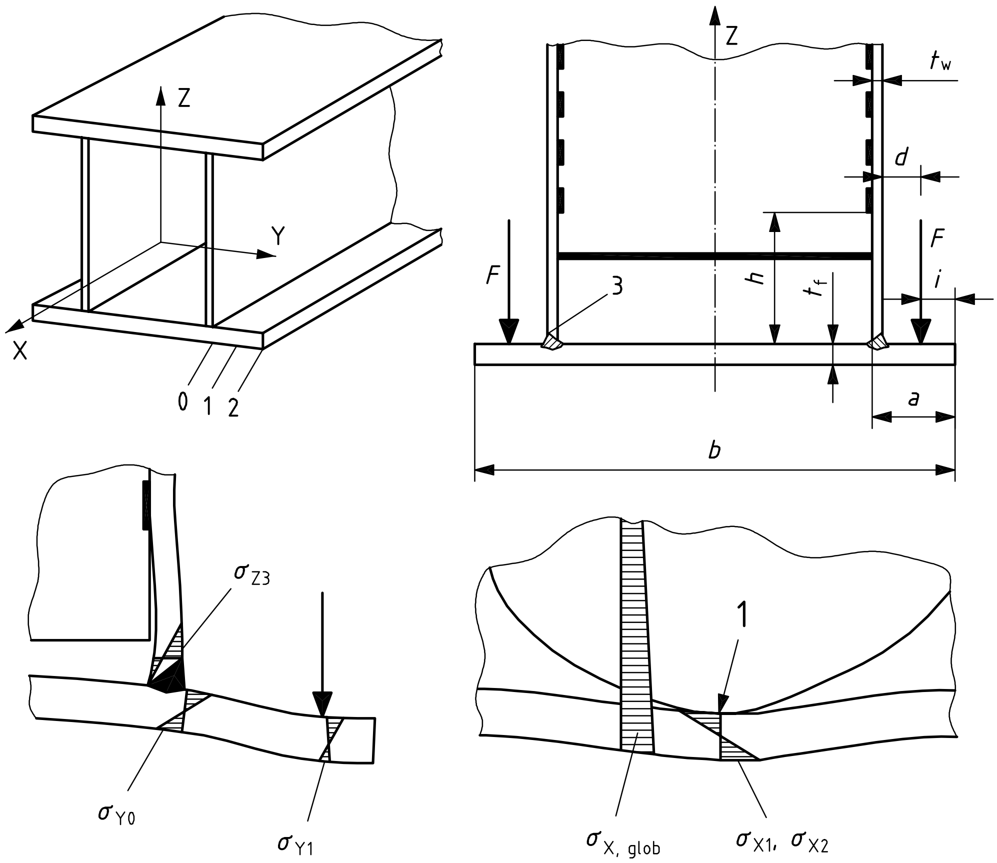
| | **I-beam with parallel flanges** | **I-beam with inclined flanges** |
|---|---|---|
| **Longitudinal bending stresses** | $c_{X0} = 0{,}050 - 0{,}580\lambda + 0{,}148\ e^{3{,}015\lambda}$ | $c_{X0} = -0{,}981 - 1{,}479\lambda + 1{,}120\ e^{1{,}322\lambda}$ |
| | $c_{X1} = 2{,}230 - 1{,}490\lambda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

© ISO 2005 – All rights reserved

Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

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

See Figure B.2.




**Key**

1 crab wheel

**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 formulae and coefficients are based on curve fitting on the results of finite element method calculation. So, they are approximations and do not have any direct physical basis.

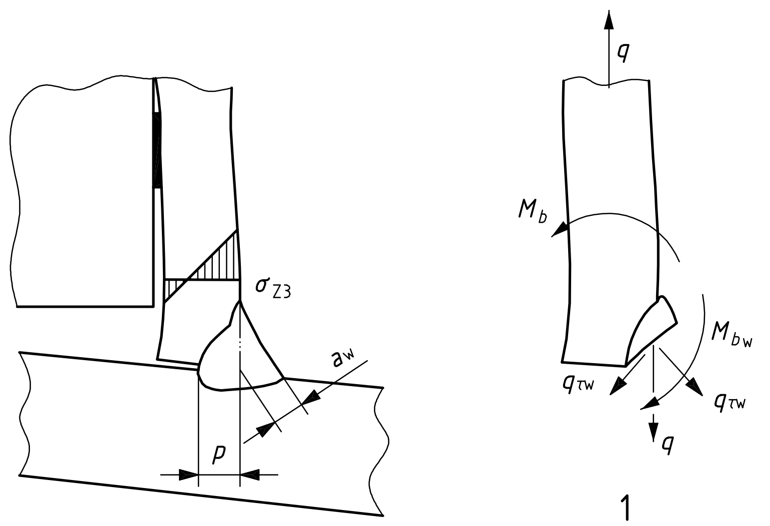
The signs of the stresses at the points 0, 1, 2 are valid at the bottom surface. The upper surface stress has the opposite sign.

Formulae for the local stresses at the fillet weld with partial penetration follow the explaining figure B.3.


© ISO 2005 – All rights reserved
Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

**Table B.2 — Formulae for stresses and coefficients**

| Point | Stress formula | Coefficients | Symbols and limits |
|---|---|---|---|
| 0 | $\sigma_{X0} = c_{X0}\frac{F}{t_f^2}$<br>$\sigma_{Y0} = c_{Y0}\frac{F}{t_f^2}$ | $c_{X0} = 0{,}123 + 0{,}48\lambda + 0{,}194\lambda^2$<br>$-0{,}5\arctan(5r_t - 1{,}375)$<br>$c_{Y0} = -1{,}306\,7 - 1{,}45r_t$<br>$+0{,}583\,3r_t^2 + 1{,}933\lambda$ | Valid for all formulae<br>$r_t = t_w / t_f$<br>$2a < b < 16a$<br>$0{,}1 < i/a < 0{,}5$<br>$0{,}15 < r_t < 0{,}8$ |
| 1 | $\sigma_{X1} = c_{X1}\frac{F}{t_f^2}$<br>$\sigma_{Y1} = c_{Y1}\frac{F}{t_f^2}$ | $c_{X1} = 2{,}23 - 1{,}49\lambda + 2e^{-18{,}33\lambda}(1 + 1{,}5r_t) + 0{,}4r_t^{2{,}5}$<br>$c_{Y1} = 0{,}33(r_t - 1) + (1 + 2r_t)\left[0{,}3\lambda + 0{,}4\sin\left(3{,}4\lambda + 0{,}4r_t^2\right)\right]$ | |
| 2 | $\sigma_{X2} = c_{X2}\frac{F}{t_f^2}$<br>$\sigma_{Y2} = 0$ | $c_{X2} = -0{,}95 + \frac{2{,}70}{(2\lambda + 0{,}5)^{r_t^{0{,}333}}} + \left[1{,}2\cdot(\lambda - 0{,}1)^{0{,}25} - 0{,}76\right]\left(\frac{0{,}2}{r_t}\right)^4$<br>$c_{Y2} = 0$ | |
| 3<br>At the web plate and at the weld toe | Stress at web is the sum of membrane ($m$) and bending ($b$) stress<br>$\sigma_{Z3} = \sigma_{Z3m} + \sigma_{Z3b} =$<br>$\sigma_{Z3} = c_{Zm}\frac{F}{(d + t_f)\,t_w}$<br>$+k_{Zh}c_{Zb}\frac{6Fd}{t_w^3\left[1 + (2r_t)^{-3}\right]}$ | $c_{Zm} = 0{,}4 + 1{,}8r_t^2$<br>$c_{Zb} = \left(0{,}01 + 0{,}021\,2r_t^3\right)\cdot\left(0{,}125\frac{b}{a} - 0{,}25\right)^{0{,}125}$<br>$k_{Zh} = 1$<br>$k_{Zh} = 1 + \frac{k_{Z0}}{1 + 0{,}000\,453\,6r_h^3}$,<br>$k_{Z0} = 2 + 1{,}5\sin\left[1{,}5\pi(0{,}35 - r_t)\right]$<br>$+0{,}45\sin\left[4\pi(r_t - 0{,}5)\right]$ | $r_h = \frac{h}{t_w}$<br>$4\text{ mm} \leqslant t_w \leqslant 12\text{ mm}$<br>$50t_w < h$<br>$0 \leqslant h < 50t_w$ |




**Key**

1 freebody diagram

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

--`,,`,`,-`-`,,`,,`,`,,`---

© ISO 2005 – All rights reserved
Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

$q = \sigma_{Z3m} t_w$; $M_b = \sigma_{Z3b} t_w^2 / 6$; $a_{wp} = a_w + p / \sqrt{2}$; $e = t_w / 2 + a_w / (2\sqrt{2}) - 0{,}75 p$;

$q_{\sigma_w} = q_{\tau_w} = q / \sqrt{2}$; $M_{bw} = M_b - qe$;

$\sigma_n = \tau_w = \dfrac{q_{\sigma_w}}{a_{wp}}$; $\sigma_{bw} = \dfrac{6 M_{bw}}{a_{wp}^2}$;

Stress in weld surface: $\sigma_{ws} = \sigma_n + \sigma_{bw}$;

Stress in weld root: $\sigma_{wr} = \sigma_n - \sigma_{bw}$.

Application of the formula for the stress combination at the weld root:

$$\left( \frac{\sigma_{X,glob} - 0{,}75 \sigma_{X0}}{f_{Rd,X}} \right)^2 + \left( \frac{\sigma_{wr}}{f_{Rd,Y}} \right)^2 - \frac{\left( \sigma_{X,glob} - 0{,}75 \sigma_{X0} \right) \sigma_{wr}}{f_{Rd,X} \cdot f_{Rd,Y}} + \left( \frac{\tau_{glob}}{f_{Rd,\tau X}} \right)^2 + \left( \frac{\tau_w}{f_{Rd,\tau Y}} \right)^2 \leqslant 1$$

Subscript “glob” refers to the global stresses at the calculation point (longitudinal normal stress and shear stress that goes through the weld).

The reduction factor 0,75 shall not be applied to the transverse stresses of the fillet weld.

NOTE 1 The calculation of local stresses according to B.1 and B.2 complies with FEM 9.341 and CMAA 74.

NOTE 2 B.3 is a further development based on finite element calculation.


© ISO 2005 – All rights reserved
Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

# Bibliography


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

[2] FEM 1.001:1998, *Rules for the design of hoisting appliances*[1)]

[3] FEM 9.341, *Rules for the design of series lifting equipment — Local girder stresses*

[4] CMAA n°74, *Specifications for Top Running Single Girder Electric Overhead Traveling Cranes*[2)]


1) FEM = Fédération européenne de la manutention.

2) CMAA = Crane Manufacturers Association of America.

© ISO 2005 – All rights reserved

Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS
Not for Resale

ICS 53.020.20

Price based on 13 pages

--`,,`,`,-`-`,,`,,`,`,,`---

© ISO 2005 – All rights reserved

Copyright International Organization for Standardization
Reproduced by IHS under license with ISO
No reproduction or networking permitted without license from IHS Not for Resale