BRITISH STANDARD

BS EN 13104:2009 +A2:2012

Incorporating Corrigendum May 2009

Railway applications — Wheelsets and bogies — Powered axles — Design method

ICS 45.040

British Standards

National foreword

This British Standard is the UK implementation of EN 13104:2009+A2:2012. It supersedes BS EN 13104:2009+A1:2010 which is withdrawn.

The UK committee draws users' attention to the distinction between normative and informative elements, as defined in Clause 3 of the CEN/CENELEC Internal Regulations, Part 3.

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The start and finish of text introduced or altered by amendment is indicated in the text by tags. Tags indicating changes to CEN text carry the number of the CEN amendment. For example, text altered by CEN amendment A1 is indicated by [A1⟩ ⟨A1].

The UK participation in its preparation was entrusted by Technical Committee RAE/3, Railway Applications - Rolling Stock Material, to Subcommittee RAE/3/-/1, Railway Applications - Wheels and Wheelsets. A list of organizations represented on this subcommittee can be obtained on request to its secretary.

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31 May 2009	Correction to national foreword
31 January 2011	Implementation of CEN amendment A1:2010
28 February 2013	Implementation of CEN amendment A2:2012

EUROPEAN STANDARD

NORME EUROPÉENNE

EUROPÄISCHE NORM

EN 13104:2009+A2

October 2012

ICS 45.040

Supersedes EN 13104:2009+A1:2010

English Version

Railway applications - Wheelsets and bogies - Powered axles - Design method

Applications ferroviaires - Essieux montés et bogies - Essieux-axes moteurs - Méthode de conception

Bahnanwendungen - Radsätze und Drehgestelle - Treibradsatzwellen - Konstruktionsverfahren

This European Standard was approved by CEN on 26 December 2008 and includes Amendment 1 approved by CEN on 14 September 2010 and Amendment 2 approved by CEN on 25 September 2012.

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EUROPEAN COMMITTEE FOR STANDARDIZATION
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Ref. No. EN 13104:2009+A2:2012: E

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Foreword

This document (EN 13104:2009+A2:2012) has been prepared by Technical Committee CEN/TC 256 "Railway applications", the secretariat of which is held by DIN.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by April 2013, and conflicting national standards shall be withdrawn at the latest by April 2013.

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. CEN [and/or CENELEC] shall not be held responsible for identifying any or all such patent rights.

This document comprises amendment 1 adopted by CEN on 2010-09-14 and amendment 2 adopted by CEN on 2012-09-25.

This document supersedes [A2⟩ EN 13104:2009+A1:2010 ⟨A2].

The start and end of the text added or modified by the amendment is indicated in the text by [A1⟩ ⟨A1] and [A2⟩ ⟨A2].

[A1⟩ This document has been prepared under a mandate given to CEN/CENELEC/ETSI by the European Commission and the European Free Trade Association, and supports essential requirements of EU Directive 2008/57/EC ⟨A1]

[A1⟩ For relationship with EU Directive 2008/57/EC, see informative Annex ZA, which is an integral part of this document. ⟨A1]

According to the CEN/CENELEC Internal Regulations, the national standards organizations of the following countries are bound to implement this European Standard: Austria, Belgium, Bulgaria, Croatia, Cyprus, Czech Republic, Denmark, Estonia, Finland, Former Yugoslav Republic of Macedonia, France, Germany, Greece, Hungary, Iceland, Ireland, Italy, Latvia, Lithuania, Luxembourg, Malta, Netherlands, Norway, Poland, Portugal, Romania, Slovakia, Slovenia, Spain, Sweden, Switzerland, Turkey and the United Kingdom.

Introduction

Railway axles were among the first train components to give rise to fatigue problems.

Many years ago, specific methods were developed in order to design these axles. They were based on a feedback process from the service behaviour of axles combined with the examination of failures and on fatigue tests conducted in the laboratory, so as to characterize and optimize the design and materials used for axles.

A European working group under the aegis of UIC[1] started to harmonize these methods at the beginning of the 1970s. This led to an ORE[2] document applicable to the design of trailer stock axles, subsequently incorporated into national standards (French, German, Italian).

This method was successfully extrapolated in France for the design of powered axles and the French standard also applies to such axles. Consequently this method was converted into a UIC leaflet.

The bibliography lists the relevant documents used for reference purposes. The method described therein is largely based on conventional loadings and applies the beam theory for the stress calculation. The shape and stress recommendations are derived from laboratory tests and the outcome is validated by many years of operations on the various railway systems.

This standard is based largely on this method which has been improved and its scope enlarged.

[1] UIC : Union Internationale des Chemins de fer.

[2] ORE: Office de Recherches et d'Essais de l'UIC.

1 Scope

This standard:

— defines the forces and moments to be taken into account with reference to masses, traction and braking conditions;

— gives the stress calculation method for axles with outside axle journals;

— specifies the maximum permissible stresses to be assumed in calculations for steel grade EA1N defined in EN 13261;

— describes the method for determination of the maximum permissible stresses for other steel grades;

— determines the diameters for the various sections of the axle and recommends the preferred shapes and transitions to ensure adequate service performance.

This standard is applicable to:

— solid and hollow powered axles for railway rolling stock;

— solid and hollow non-powered axles of motor bogies;

— solid and hollow non-powered axles of locomotives[3];

— axles defined in prEN 13261;

— all gauges[4].

This standard is applicable to axles fitted to rolling stock intended to run under normal European conditions. Before using this standard, if there is any doubt as to whether the railway operating conditions are normal, it is necessary to determine whether an additional design factor has to be applied to the maximum permissible stresses. The calculation of wheelsets for special applications (e.g. tamping/lining/levelling machines) may be made according to this standard only for the load cases of free-running and running in train formation. This standard does not apply to workload cases. They are calculated separately.

For light rail and tramway applications, other standards or documents agreed between the customer and supplier may be applied.

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.

EN 13260:2003, *Railway applications — Wheelsets and bogies — Wheelsets — Product requirements*

EN 13261:2003, *Railway applications — Wheelsets and bogies —– Axles — Product requirements*

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[3] In France, the interpretation of the term "locomotive" includes locomotives, locomoteurs or locotracteurs.

[4] If the gauge is not standard, certain formulae need to be adapted

3 Symbols and abbreviations

For the purposes of this European Standard, the symbols and abbreviations in Table 1 apply:

Table 1

Symbol	Unit	Description
m_1	kg	Mass on journals (including bearings and axle boxes)
m_2	kg	Wheelset mass and masses on the wheelset between running surfaces (brake disc, etc.)
$m_1 + m_2$	kg	For the wheelset considered, proportion of the mass of the vehicle on the rails
g	m/s^2	Acceleration due to gravity
P	N	Half the vertical force per wheelset on the rail $\frac{(m_1 + m_2)g}{2}$
P_0	N	Vertical static force per journal when the wheelset is loaded symmetrically $\frac{m_1 g}{2}$
P_1	N	Vertical force on the more heavily-loaded journal
P_2	N	Vertical force on the less heavily-loaded journal
P'	N	Proportion of P braked by any mechanical braking system
Y_1	N	Wheel/rail horizontal force perpendicular to the rail on the side of the more heavily-loaded journal
Y_2	N	Wheel/rail horizontal force perpendicular to the rail on the side of the less heavily-loaded journal
H	N	Force balancing the forces Y_1 and Y_2
Q_1	N	Vertical reaction on the wheel situated on the side of the more heavily-loaded journal
Q_2	N	Vertical reaction on the wheel situated on the side of the less heavily-loaded journal
F_i	N	Forces exerted by the masses of the unsprung elements situated between the two wheels (brake disc(s), pinion, etc.)
F_f	N	Maximum force input of the brake shoes of the same shoeholder on one wheel or interface force of the pads on one disc
M_x	N·mm	Bending moment due to the masses in motion
M'_x , M'_z	N·mm	Bending moments due to braking
M'_y	N·mm	Torsional moment due to braking
M''_x , M''_z	N·mm	Bending moments due to traction
M''_y	N·mm	Torsional moment due to traction
MX , MZ	N·mm	Sum of bending moments
MY	N·mm	Sum of torsional moments
MR	N·mm	Resultant moment

Table 1 (*continued*)

Symbol	Unit	Description
$2b$	mm	Distance between vertical force input points on axle journals
$2s$	mm	Distance between wheel treads
h_1	mm	Height above the axle centreline of vehicle centre of gravity of masses carried by the wheelset
y_i	mm	Distance between the tread of one wheel and force F_i
y	mm	Abscissa for any section of the axle calculated from the section subject to force P_1
Γ		Average friction coefficient between the wheel and the brake shoe or between the brake pads and the disc
σ	N/mm^2	Stress calculated in one section
K		Fatigue stress concentration factor
R	mm	Nominal radius of the tread of a wheel
R_b	mm	Brake radius
d	mm	Diameter for one section of the axle
d'	mm	Bore diameter of a hollow axle
D	mm	Diameter used for determining K
r	mm	Radius of transition fillet or groove used to determine K
S		Security coefficient
G		Centre of gravity
R_{fL}	N/mm^2	Fatigue limit under rotating bending up to 10^7 cycles for unnotched test pieces
R_{fE}	N/mm^2	Fatigue limit under rotating bending up to 10^7 cycles for notched test pieces
a_q	m/s^2	Unbalanced transverse acceleration
f_q		Thrust factor

4 General

The major phases for the design of an axle are:

a) definition of the forces to be taken into account and calculation of the moments on the various sections of the axle;

b) selection of the diameters of the axle body and journals and - on the basis of these diameters - calculation of the diameters for the other parts;

c) the options taken are verified in the following manner:

- stress calculation for each section;
- comparison of these stresses with the maximum permissible stresses.

The maximum permissible stresses are mainly defined by:

- the steel grade;

— whether the axle is solid or hollow;

— the type of transmission of motor power.

An example of a data sheet with all these phases is given in Annex A.

5 Forces and moments to be taken into consideration

5.1 Types of forces

Two types of forces are to be taken into consideration as a function:

— of the masses in motion;

— of the braking system;

— of the traction.

5.2 Influence of masses in motion

The forces generated by masses in motion are concentrated along the vertical symmetry plane (y, z) (see Figure 1) intersecting the axle centreline.

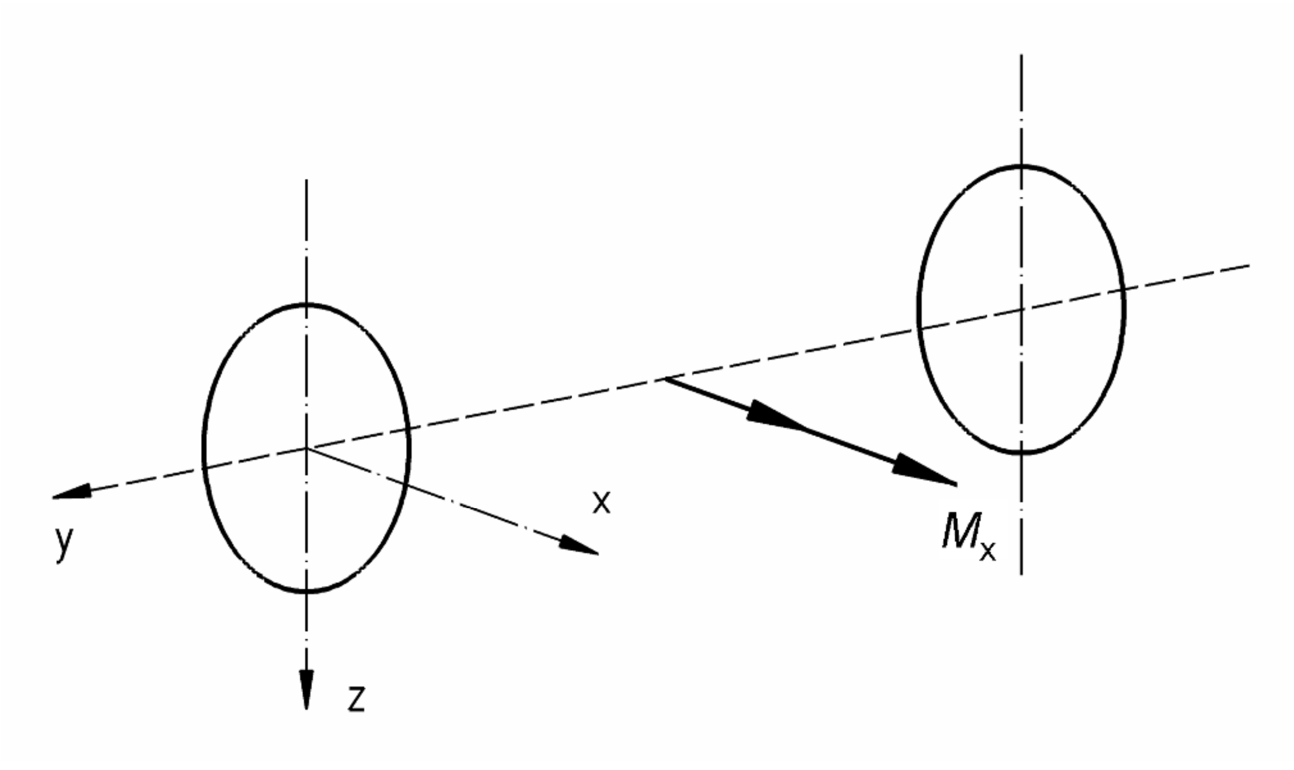


Figure 1

Unless otherwise defined by the customer, the masses $(m_1 + m_2)$ to be taken into account for the main types of rolling stock are defined in Table 2. For particular applications, e.g. suburban vehicles, other definitions for masses are necessary, in accordance with the specific operating requirements.

Table 2

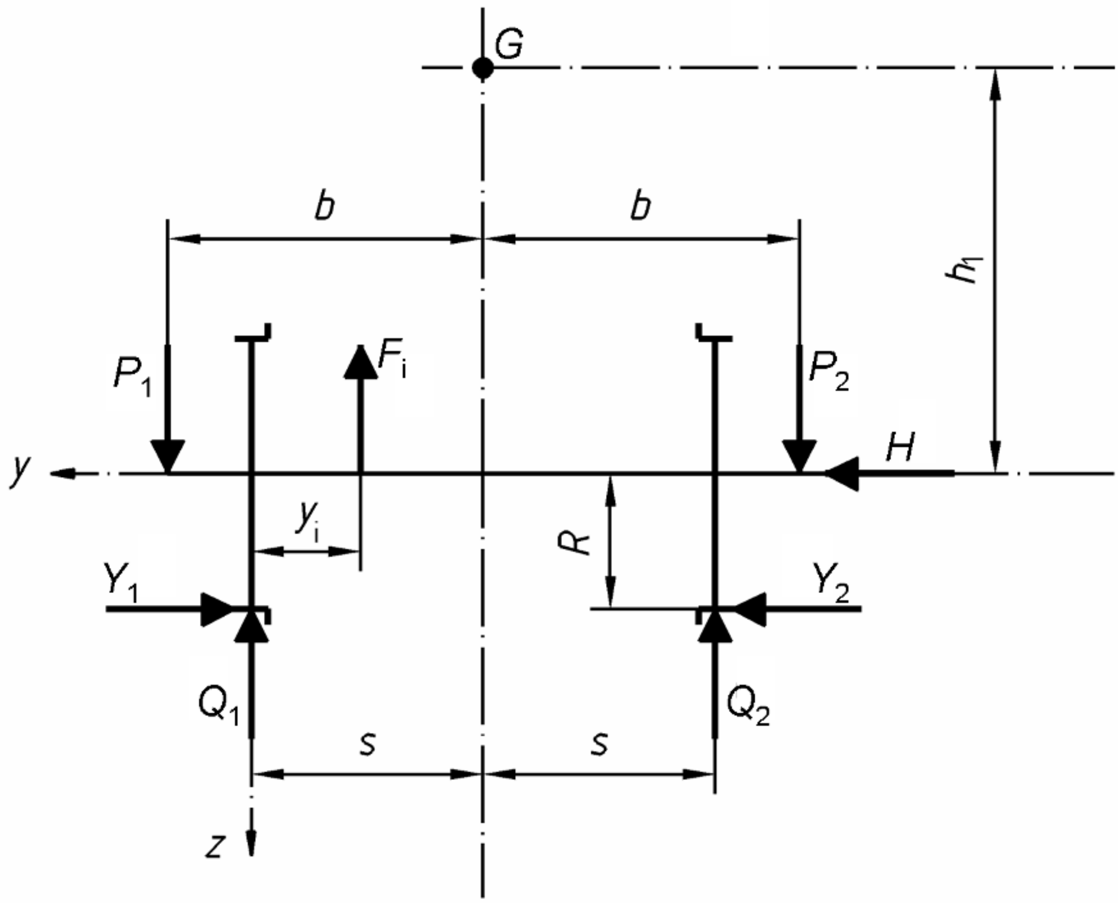
Type of rolling stock units	Mass $(m_1 + m_2)$
Traction units with no passenger accommodation, luggage areas and postal vans	For the axle considered, proportion of the wagon mass under maximum permissible loading in service
Traction units including passenger accommodation, luggage areas and postal vans 1 – Main line vehicles[a]	Mass in service + 1,2 × payload, "mass in service" is defined as: the vehicle mass without passengers, tanks full (of water, sand, fuel, etc.); "payload" is defined as the mass of a passenger estimated at 80 kg, including hand luggage; — 1 passenger per seat; — 2 passengers per m² in corridors and vestibules; — 2 passengers per attendant compartment; — 300 kg per m^2 in luggage compartments.
2 – Suburban vehicles[a, b]	Mass in service + 1,2 × payload, "mass in service" is defined as the vehicle mass without passengers, tanks full (of water, sand, fuel, etc.); "payload" is defined as the mass of a passenger, which is estimated at 70 kg (little or no luggage); — 1 passenger per seat; — 3 passengers per m² in corridor areas; — 4 or 5 passengers per m² in vestibule areas [b]; — 300 kg per m^2 in luggage compartments.

[a] The payloads to be taken into account to determine the mass of the mainline and suburban vehicles broadly reflect the normal operating conditions of the member railways of the International Union of Railways (UIC). If the operating conditions differ significantly, these masses may be modified, for example, by increasing or decreasing the number of passengers per m² in corridors and vestibules.

[b] These vehicles are sometimes associated with classes of passenger travel, i.e. 1st or 2nd class.

The bending moment M_x in any section is calculated from forces P_1, P_2, Q_1, Q_2, Y_1, Y_2 and F_i as shown in Figure 2. It represents the most adverse condition for the axle, i.e.:

- asymmetric distribution of forces;
- the direction of the forces F_i due to the masses of the unsprung components selected in such a manner that their effect on bending is added to that due to the vertical forces;
- the value of the forces F_i results from multiplying the mass of each unsprung component by 1 g.



Key

G centre of gravity of vehicle

Figure 2

Table 3 shows the values of the forces calculated from m_1 .

The formulae coefficient values are applicable to standard gauge axles and classical suspension. For very different gauges, metric gauge for example, or a new system of suspension, tilting system for example, other values shall be considered (see Annexes B and C).

Table 3

For all wheelset defined in the scope of this standard	$P_1 = (0{,}625 + 0{,}0875 h_1 / b) m_1 g$ $P_2 = (0{,}625 - 0{,}0875 h_1 / b) m_1 g$ $Y_1 = 0{,}35 m_1 g$ $Y_2 = 0{,}175 m_1 g$ $H = Y_1 - Y_2 = 0{,}175 m_1 g$

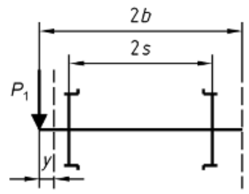
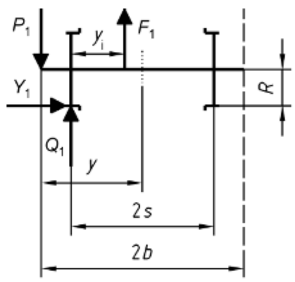
For all wheelsets

$$Q_1 = \frac{1}{2s}[P_1(b+s) - P_2(b-s) + (Y_1 - Y_2)R - \sum_i F_\mathrm{i}(2s - y_\mathrm{i})]$$

$$Q_2 = \frac{1}{2s}[P_2(b+s) - P_1(b-s) - (Y_1 - Y_2)R - \sum_i F_\mathrm{i} y_\mathrm{i}]$$

Table 4 shows the formulae to calculate M_x for each zone of the axle and the general outline of M_x variations along the axle.

Table 4

<table>
<tr><th>Zone of the axle</th><th>M_x [a]</th></tr>
<tr><td>Between loading plane and running surface</td><td>$M_x = P_1 y$

</td></tr>
<tr><td>Between running surfaces</td><td>$M_x = P_1 y - Q_1 (y - b + s) + Y_1 R - \sum_i F_i (y - b + s - y_i)$


F_i : force(s) on the left of the section considered</td></tr>
<tr><td>General outline of M_x variations</td><td>
</td></tr>
<tr><td colspan="2">[a] For a non-symmetric wheelset, the calculations shall be carried out after applying the load alternately to the two journals to determine the worst case.</td></tr>
</table>

5.3 Effects due to braking

Braking generates moments that can be represented by three components: $M_x^{'}$, $M_y^{'}$, $M_z^{'}$ (see Figure 3).

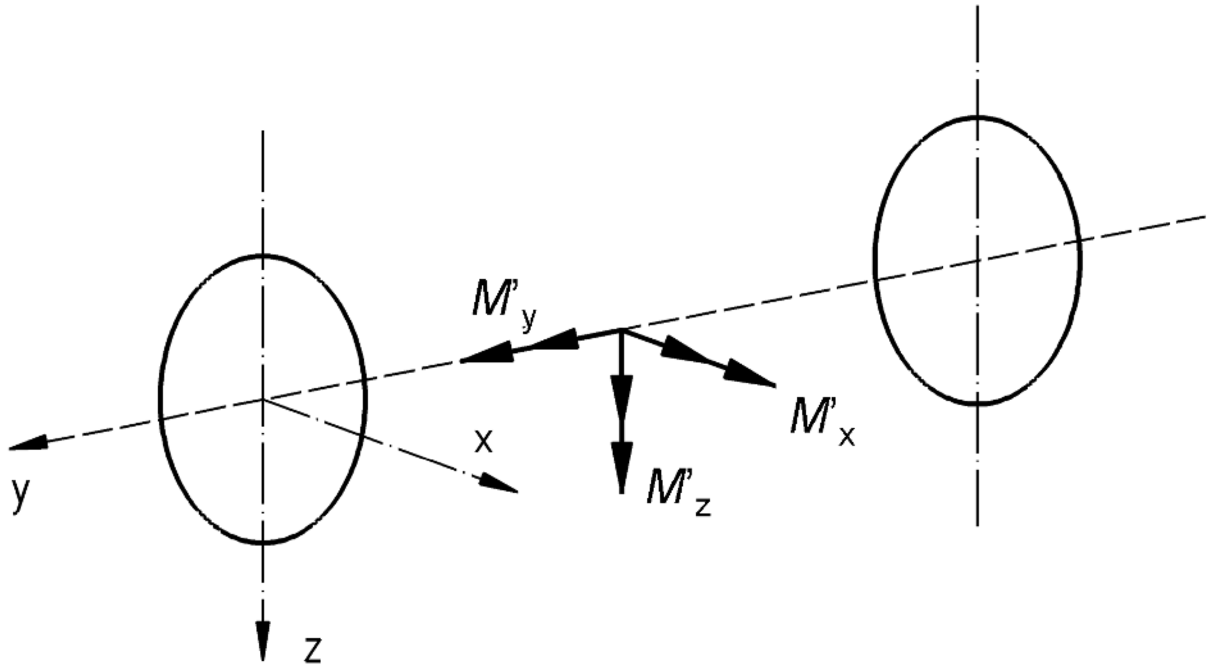


Figure 3

— the bending component $M_x^{'}$ is due to the vertical forces parallel to the z axis;

— the bending component $M_z^{'}$ is due to the horizontal forces parallel to the x axis;

— the torsional component $M_y^{'}$ is directed along the axle centreline (y axis); it is due to the forces applied tangentially to the wheels.

The components $M_x^{'}$, $M_y^{'}$ and $M_z^{'}$ are shown in Table 6 for each method of braking.

If several methods of braking are superimposed, the values corresponding to each method shall be added.

For example, forces and moments due to electric braking or regenerative braking shall be added.

NOTE If other methods of braking are used, the forces and moments to be taken into account can be obtained on the basis of the same principles as those shown in Table 6. Special attention should be paid to the calculation of the $M_x^{'}$ component, which is to be added directly to the M_x component representing masses in motion.

5.4 Effects due to curving and wheel geometry

For an unbraked wheelset, the torsional moment $M_y^{'}$ is equal to 0,2 *PR* to account for possible differences in wheel diameters and the effect of passing through curves.

For a braked wheelset, these effects are included in the effects due to braking.

5.5 Effects due to traction

The forces generated in the axle from the transmission of the driving torque under constant adhesion conditions can normally be neglected. Calculation and experience have shown that the bending moments $M_x^{"}$ and $M_z^{"}$, and torsional moment $M_y^{"}$, are smaller than those generated by braking. Traction and braking moments do not occur simultaneously.

The axle design should also take into account the instantaneous loss of traction, e.g. short-circuit overload.

Where traction control systems adopt a technique to maintain the tractive effort at the limit of adhesion, any resultant controlled oscillations about the mean driving torque shall be considered in determining the magnitude of the torsional moment $M_y^{"}$.

For some applications, when driving torque is very high in starting conditions, and when they occur very often, the calculation shall be done as follows:

a) with the usual conditions described as above in 5.2, 5.3 and 5.4;

b) with the following starting conditions:

   1) effects due to masses in motion given by Table 5;

   2) effects due to starting driving torque.

The effect of the conditions defined in b 1) and b 2) shall be combined.

The most severe conditions between a) and b) have to be used to define the axle.

Table 5

Starting forces	$P_1 = 0{,}55 m_1 g$ $P_2 = 0{,}55 m_1 g$ $Y_1 = 0{,}10 m_1 g$ $Y_2 = 0{,}05 m_1 g$ $H = 0{,}05 m_1 g$

5.6 Calculation of the resultant moment

In every section, the maximum stresses are calculated from the resultant moment MR (see the following note), which is equal to:

$$MR = \sqrt{MX^2 + MY^2 + MZ^2}$$

where MX, MY and MZ are the sums of the various components due to masses in motion and braking:

$MX = M_x + \sum M'_x$ 5

$MY = \sum M'_y$ 5

$MZ = \sum M'_z$ 5

NOTE At a point on the outer surface of a solid cylinder (also in the case of a hollow one) with d as diameter, the components MX , MY and MZ generate:

— a direct stress for MX and MZ ;

— a shear stress for MY .

The direct stress has the following value (bending of beams with a circular section):

$$\sigma_n = \frac{32\sqrt{MX^2 + MZ^2}}{\pi d^3}$$

The value of the shear stress is the following (torsion of beams with a circular section):

$$\sigma_t = \frac{16MY}{\pi d^3}$$

As a result, the two principal stresses σ_1 and σ_2 are obtained as:

$$\sigma_1 = \frac{\sigma_n + \sqrt{\sigma_n^2 + 4\sigma_t^2}}{2} \quad \sigma_2 = \frac{\sigma_n - \sqrt{\sigma_n^2 + 4\sigma_t^2}}{2}$$

Since the direct stress has a much higher absolute value (10 to 20 times) than the shear stress, the diameter of the largest Mohr's circle is selected ($\sigma_1 - \sigma_2$ in this case) as a check of the value assumed for d .

$$\sigma = \sigma_1 - \sigma_2 = \sqrt{\sigma_n^2 + 4\sigma_t^2} = \frac{32}{\pi d^3}\sqrt{MX^2 + MZ^2 + MY^2}$$

As a result, the definition of a resultant moment is:

$$MR = \sqrt{MX^2 + MY^2 + MZ^2}$$

5 The values M'_x , M'_y , M'_z may be replaced respectively by M''_x , M''_y and M''_z if the moments due to traction are greater than the moments due to braking.

Table 6

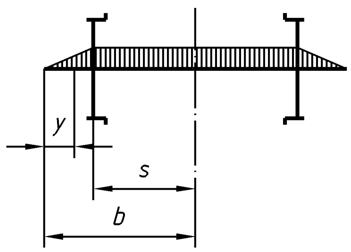
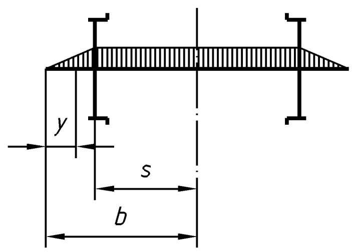
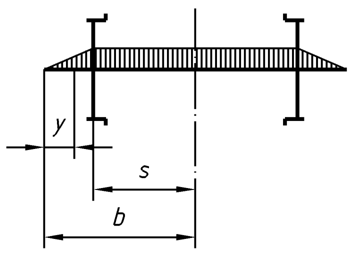
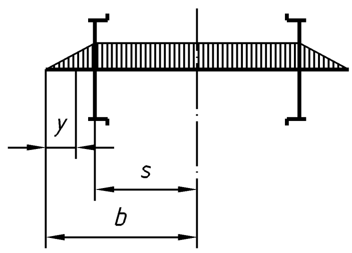
Components M'_x, M'_z, M'_y	**Method of braking used**			
	Friction brake blocks on both sides of each wheel		Friction brake block on one side only of each wheel	
	Between loading plane and running surface	Between running surfaces	Between loading plane and running surface	Between running surfaces
M'_x	$M'_x = 0{,}3 F_f \Gamma y$ [a] [b] 	$M'_x = 0{,}3 F_f \Gamma (b - s)$ [a] [b]	$M'_x = F_f \Gamma y$ [b] 	$M'_x = F_f \Gamma (b - s)$ [b]
M'_z	$M'_z = F_f (0{,}3 + \Gamma) y$ [a] 	$M'_z = F_f (0{,}3 + \Gamma)(b - s)$ [A]	$M'_z = F_f (1 + \Gamma) y$ 	$M'_z = F_f (1 + \Gamma)(b - s)$
M'_y	$M'_y = 0$	$M'_y = 0{,}3 P'R$ [c] [d]	$M'_y = 0$	$M'_y = 0{,}3 P'R$ [c] [d]

Table 6 (*continued*)

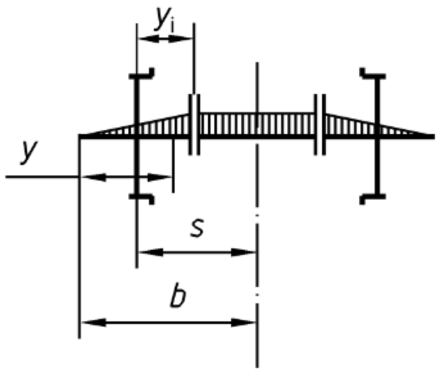
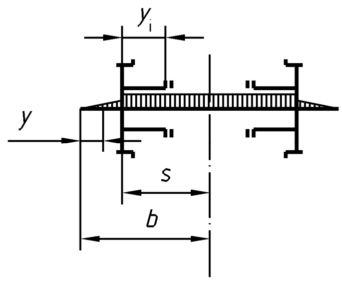
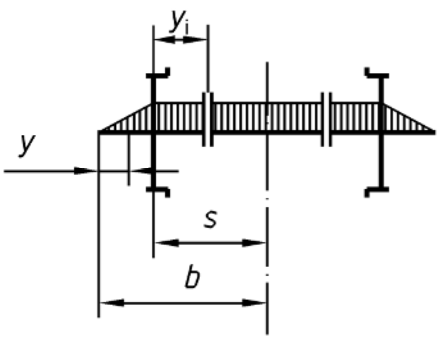
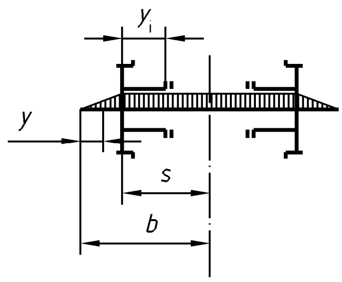
Components	**Method of braking used**				
M'_x, M'_z, M'_y	Two disc brakes mounted on the axle			Two disc brakes attached to the wheel hub[f]	
	Between loading plane and running surface	Between running surfaces and disc	Between discs	Between loading plane and running surface	Between running surfaces
M'_x	$M'_x = F_f \Gamma y$ [b] 	$M'_x = F_f \Gamma (b - s + y_i)$ [B]		$M'_x = F_f \Gamma y$ [b] 	$M'_x = F_f \Gamma (b - s + y_i)$ [b]
M'_z	$M'_z = F_f \Gamma \dfrac{R_b}{R} y$ [b] 	$M'_z = F_f \Gamma \dfrac{R_b}{R} (b - s)$ [B]		$M'_z = F_f \Gamma \dfrac{R_b}{R} y$ [b] 	$M'_z = F_f \Gamma (b - s) \dfrac{R_b}{R}$ [b]
M'_y	$M'_y = 0$	$M'_y = 0{,}3 P' R$ [d e]	$M'_y = 0{,}3\, P'R$ [d, e]	$M'_y = 0$	$M'_y = 0{,}3\, P'R$ [d, e]

Table 6 (*continued*)

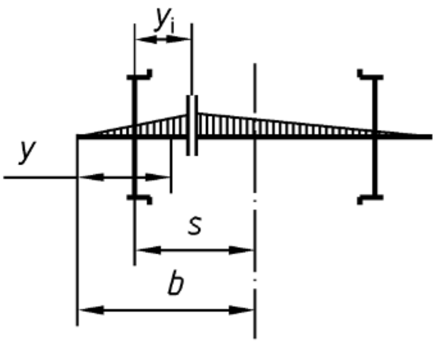
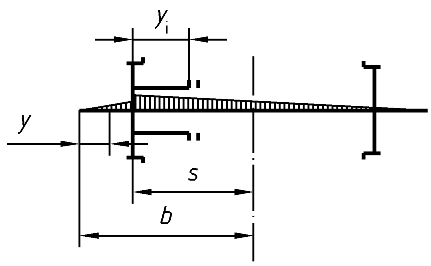
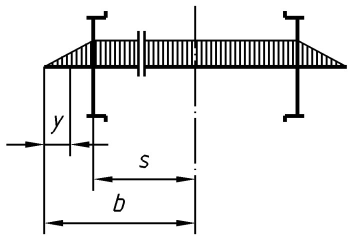
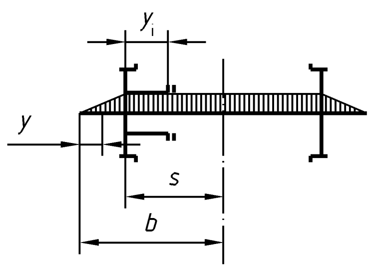
Components M'_x, M'_z, M'_y	**Method of braking used**			
	One disc brake mounted on the axle		One disc brake attached to the wheel hub[f]	
	Between first loading plane and disc	Between disc and second loading plane	Between first loading plane and disc	Between disc and second loading plane
M'_x	$M'_x = F_f \Gamma \frac{(b+s-y_i)}{2b} y$ b 	$M'_x = F_f \Gamma \frac{(b-s+y_i)}{2b} (2b-y)$ B	$M'_x = \frac{1}{2b} F_f \Gamma y (b+s-y_i)$ b 	$M'_x = F_f \Gamma \frac{(b-s+y_i)}{2b} (2b-y)$ b
	Between loading planes and running surface	Between running surfaces		
M'_z	$M'_z = \frac{1}{2} F_f \Gamma \frac{R_b}{R} y$ 	$M'_z = \frac{1}{2} F_f \Gamma \frac{R_b}{R} (b-s)$	$M'_z = \frac{1}{2} F_f \Gamma \frac{R_b}{R} y$ 	$M'_z = \frac{1}{2} F_f \Gamma (b-s) \frac{R_b}{R}$
	Between loading planes and running surface	Between running surfaces		
M'_y	$M'_y = 0$	$M'_y = 0,3\ P'R$ d, e	$M'_y = 0$	$M'_y = 0,3\ P'R$ d, e

Table 6 (*continued*)

[a] The coefficient 0,3 results from experiments which established the possible differences between the applied forces of two blocks on each wheel.
[b] Unless other values are justified: for brake blocks: $\Gamma = 0{,}1$ for cast iron blocks; $\Gamma = 0{,}17$ for all blocks with low-friction coefficient excluding cast iron; $\Gamma = 0{,}25$ for all blocks with high-friction coefficient excluding cast iron. for brake pads: $\Gamma = 0{,}35$.
[c] This value was obtained from experimental tests and corresponds to a braking force difference between the two wheels producing a force difference tangential to the wheels and equates to $0{,}3P'$. It includes the torsional moment as specified in 5.3.
[d] P' is the proportion of P braked with the method of braking considered.
[e] By convention, the torsional moment between running surfaces is selected at the value of $0{,}3P'R$. It includes the torsional moment due to braking and the torsional moment as specified in 5.4.
[f] When the disc is mounted on the wheel web, then $y_i = 0$

6 Determination of geometric characteristics of the various parts of the axle

6.1 Stresses in the various sections of the axle

On any section of the axle with *d* as diameter, the stress[6] to be taken into account is the following:

— for a solid axle (see Figure 4a): $\sigma = \dfrac{K \times 32 \times MR}{\pi d^3}$ [7]

— for a hollow axle (see Figure 4b):

— on the outer surface: $\sigma = \dfrac{K \times 32 \times MR \times d}{\pi (d^4 - d'^4)}$

— in the bore: $\sigma = \dfrac{K \times 32 \times MR \times d'}{\pi (d^4 - d'^4)}$

[6] In the case of a conical wheel seat, the stress is calculated for the section where the resultant moment is the highest and the diameter of this section is taken to be equal to the lower diameter of the wheel seat.

[7] K is a fatigue stress concentration factor (i.e. it takes into account the geometry and the material properties).

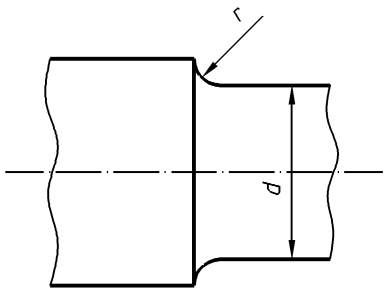


Figure 4a

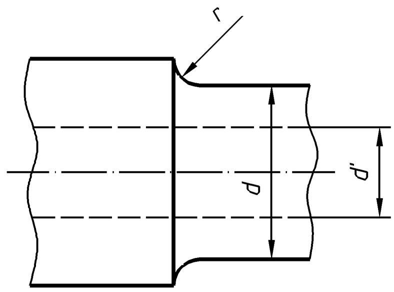


Figure 4b

In a cylindrical part situated on the surface of a solid or hollow axle and in the bore of a hollow axle, the stress concentration factor K is equal to 1. However, each change in section produces a stress increment, the maximum value of which can be found:

— at the bottom of a transition between two adjacent cylindrical parts with different diameters;

— at the groove bottom.

NOTE When the transition comprises several radii, it is recommended that the critical section should not be at the intersection of two radii. If this situation occurs, it is necessary to calculate the stress level at each intersection of the transition radii.

The stress concentration factor K to calculate this increment is shown in the nomograms in Figure 5 (transition between two cylindrical parts) and in Figure 6 (groove bottom). It is obtained from two ratios:

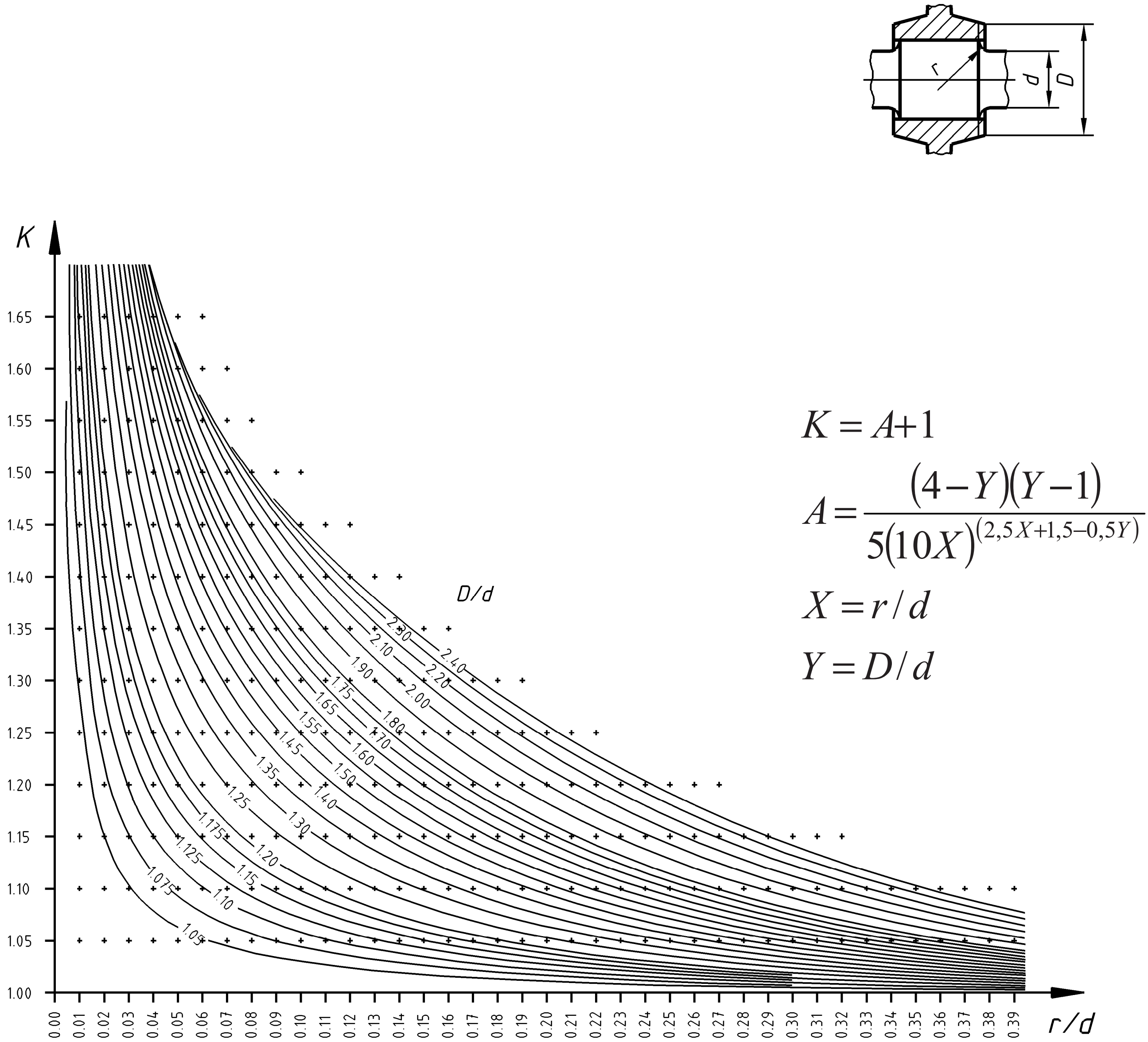
$$\frac{r}{d} \text{ and } \frac{D}{d}$$

where:

r is the transition fillet or groove radius;

d is the diameter of the cylindrical part in which the stress concentration is calculated;

D is the diameter of the other cylindrical part.



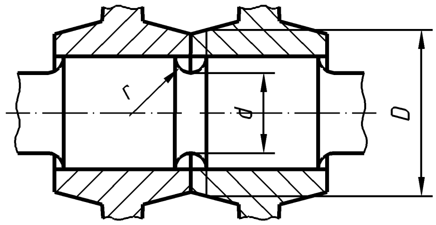
$$K = A + 1$$

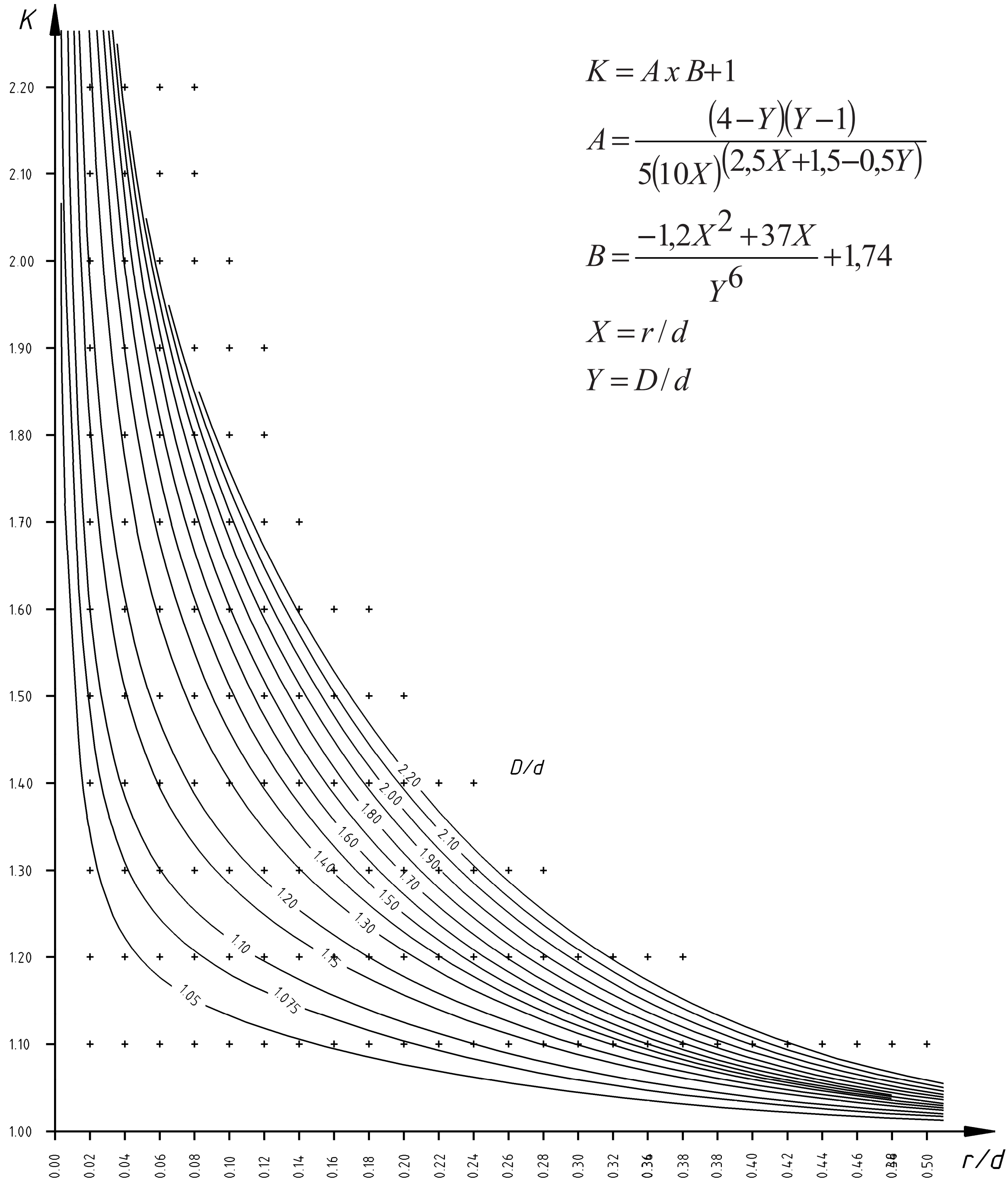
$$A = \frac{(4 - Y)(Y - 1)}{5(10X)^{(2,5X + 1,5 - 0,5Y)}}$$

$$X = r/d$$

$$Y = D/d$$

Figure 5 — Stress concentration factor *K* as a function of *D/d* and *r/d*





$$K = A\,x\,B + 1$$

$$A = \frac{(4-Y)(Y-1)}{5(10X)^{(2{,}5X+1{,}5-0{,}5Y)}}$$

$$B = \frac{-1{,}2X^2 + 37X}{Y^6} + 1{,}74$$

$$X = r/d$$

$$Y = D/d$$

Figure 6 — Stress concentration factor *K* as a function of *D/d* and *r/d* (groove bottom)

When a wheel, a disc, a pinion or a bearing is press-fitted (cold or hot) on a seat, D is to be assumed to be equal to the diameter of the hub or the bearing ring (see Figures 7a, 7b and 7c). For a collar or deflector or cross-bar, D is assumed to be equal to the diameter of the bearing seat, since the interference fit of these parts is very small.

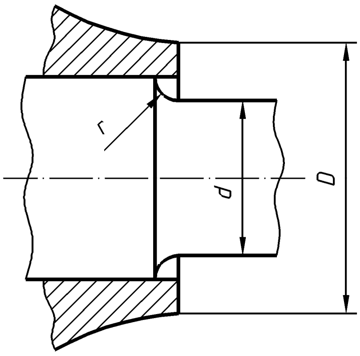


Figure 7a [8]

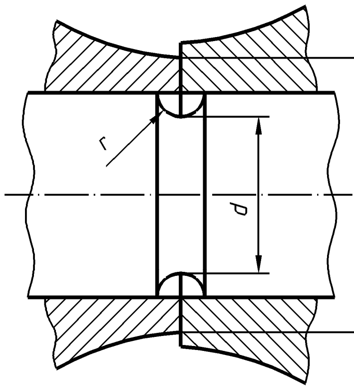


Figure 7b [8]

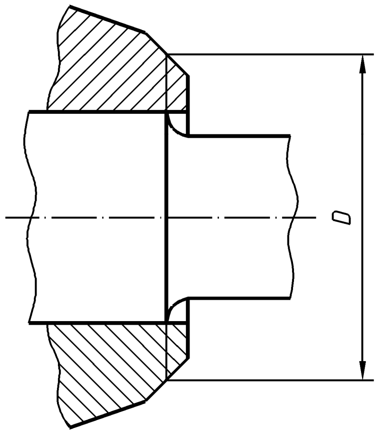


Figure 7c [8]

The design shall be verified taking into account the minimum diameters associated with the dimensional tolerances and including the authorized maintenance machining.

6.2 Determination of the diameter of journals and axle bodies

In selecting the diameters of the journals and axle body, reference should be made initially to existing sizes of associated components (e.g. bearings).

The maximum stresses in the axle should then be calculated using the following formulae:

— for a solid axle : $\sigma = \dfrac{K \times 32 \times MR}{\pi d^3}$

— for a hollow axle : $\sigma = \dfrac{K \times 32 \times MR \times d}{\pi (d^4 - d'^4)}$

The selection of diameters is then verified as shown in clause 7, the calculated stresses being compared to the maximum permissible stresses. A very shallow groove (0,1 mm to 0,2 mm) shall be provided so that the end of the inner bearing ring does not cause any notch effect on the journal (see Figure 8).

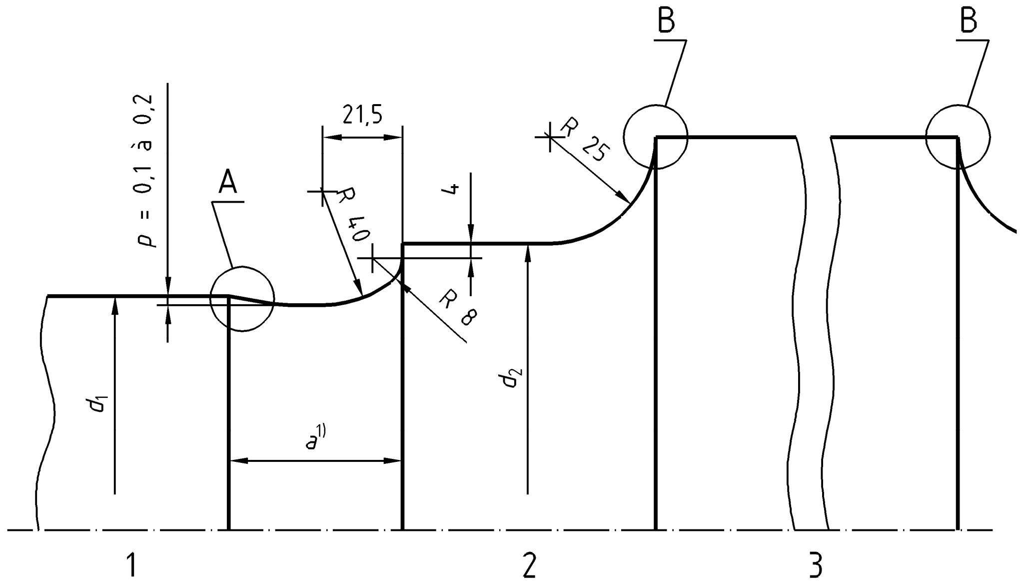
---

[8] For very thick hubs (certain cog wheels, for instance), diameter D is assumed to be that of the outer face of the hub perpendicular to the seat.

6.3 Determination of the diameter of the various seats from the diameter of the axle body or from the journals

6.3.1 Collar bearing surface

In order to standardize, whenever possible, the diameter of the collar bearing surface (d_2) should be 30 mm greater than that of the journal (d_1). The transition between the journal and the collar bearing surface is then provided as specified in Figure 8 and Figure 11.

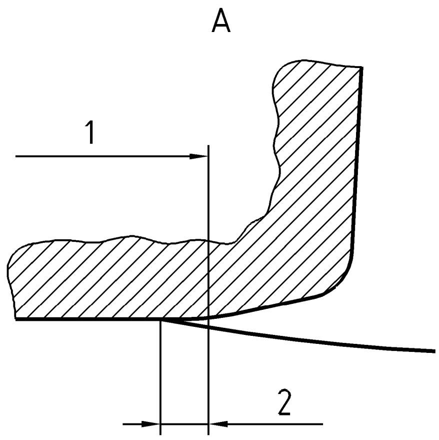


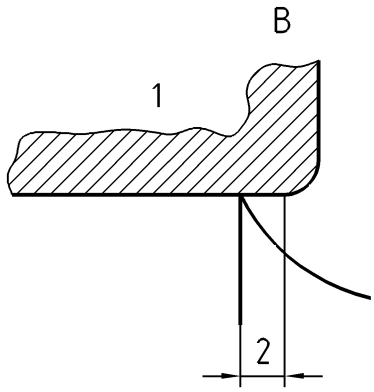
Key

1 journal
2 collar bearing
3 wheel seat
X *p* = 0,1 to 0,2

1) Variant when a is too large for maintaining the depth p with a single radius of 40 mm

Figure 8 — Transition areas between: journal and collar bearing — collar bearing and wheel seat





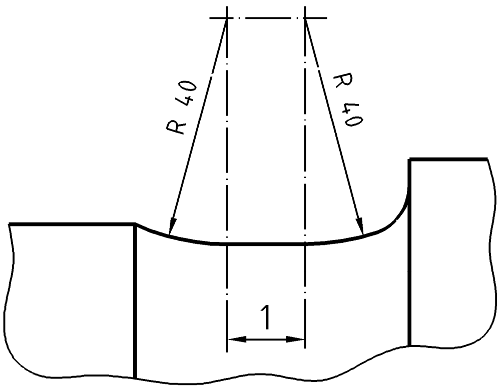
Key

1 cylindrical part of the bearing ring bore
2 ≥ 2 to ≤ 3: overlap

Figure 9 — Detail A

Key
1 wheel hub
2 ≥ 0 to ≤ 5 overlap, considering all possible tolerances and maintenance conditions should be taken into account

Figure 10 — Detail B



Key

1 bottom of cylindrical groove

Figure 11 — Transition between journal and collar bearing

6.3.2 Transition between collar bearing surface and wheel seat

In order to standardize, whenever possible, this transition should have only a single radius of 25 mm.

If this value cannot be met, the highest possible value should be selected so as to minimize the stress concentration on this area.

6.3.3 Wheel seat in the absence of an adjacent seat

The ratio between the wheel seat and the axle body diameter shall be at least equal to 1,12 at the wear limit. It is recommended that this ratio is at least 1,15 for an axle in new condition.

The transition between these two areas should be provided in such a way that the stress concentration remains at the lowest possible level.

The lengths of the wheel seat and of the cylindrical part of the wheel hub bore are selected so that the latter slightly overlaps the wheel seat, especially on the axle body side. The design shall ensure that, at the maintenance limits, there is an overlap for the limit configurations including the maintenance tolerances.

NOTE 1

1. The measurement point on the wheelset is the point of intersection of the transition radius and the surface of the entry cone.

2. The overlap criterion applies to the chamfered hubs of the brake discs and gears.

In order to have a low value of K at the transition between axle body and wheel, disc or gear seats, the value of the radius on the body side shall be at least 75 mm.

NOTE 2 Recommendations available in 4.3.2 of the ORE RP 11 report.

An example of this transition is given in Figure 12.

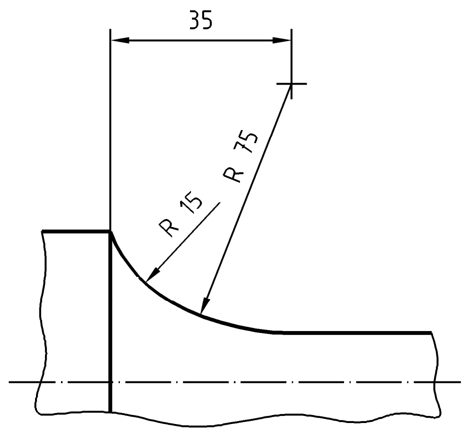


Figure 12 — Transition between body and wheel seat

6.3.4 Case of two adjacent wheel seats

Two seats shall be regarded as being adjacent if the transition of one seat to the other is by means of a single radius or a combination of radii and the fitted components are in contact.

The wheel, pinion, disc or bearing seats shall be taken into account, not the collar, deflector or cross-bar seats.

The diameter of the two seats is calculated on the basis of that of the body taking into account the requirement of 6.3.3.

A small groove (minimum depth very slightly greater than the seat wear range and minimum radius of 16 mm) is provided to separate the two wheel seats. Its main role is to prevent notches that could be produced by the bore ends of the fitted components.

In addition, the transition between the body and the wheel seats shall be as specified in 6.3.3.

6.3.5 Case of two non-adjacent wheel seats

Two wheel seats shall be regarded as not being adjacent if the transition between the two seats comprises two transition radii and the fitted components are not in contact.

The procedure is as follows:

— calculation of the diameter of each seat (see 6.3.3);

— provision of overlapping hubs (see 6.3.3);

Use the recommended transitions where possible (see 6.3.3). For designs with a diameter ratio less than 1,12, the seat fatigue limit can be less than the required value in 7.2 and 7.3. These values shall be verified on 3 axles of representative geometry (considering the lowest diameter ratio between seat and bottom of the groove).

Provide a cylindrical part between two transitions.

7 Maximum permissible stresses

7.1 General

The maximum permissible stresses are derived from:

— the fatigue limit in rotating bending for the various areas of the axle;

— the value of a security coefficient "*S*", which varies with the steel grade [A2〉 *text deleted* 〈A2].

7.2 Steel grade EA1N

The fatigue limit values [A2〉 used for the design process 〈A2] are set out below:

— for a solid axle:

  — 200 N/mm² outside the fitting;

  — 120 N/mm² beneath the fitting;

— for a hollow axle:

  — 200 N/mm² outside the fitting;

  — 110 N/mm² beneath the fitting, except the journal;

  — 94 N/mm² beneath the fitting on the journal;

  — 80 N/mm² for the surface of the bore.

Tables 7 and 8 indicate respectively for solid and hollow axles:

— the security coefficient values *S* by which the fatigue limits have to be divided to obtain the maximum permissible stresses;

— the maximum permissible stresses.

[A2] The selection of the value for coefficient S shall take into account:

— the system (if applied) for protecting the exposed areas of the axle body from, for example, impacts and corrosion;

— the associated in-service inspections and overhaul in accordance with EN 15313. [A2]

Table 7 — Maximum permissible stresses for solid axles in steel grade EA1N

Intended use of the axle	[A2] **Security coefficient** S [a] [A2]	[A2] **Zone 1**[b] [A2] N/mm²	[A2] **Zone 2**[c] [A2] N/mm²
Powered axle with press-fit driving gear or pinion	1,5	133	80
Other cases	1,3	154	92

[A2] [a] Minimum value, unless measurements exist that demonstrate the loads are more precisely defined than those defined in this standard within an appropriate maintenance regime which maintains the track conditions, whereby a lower value of security coefficient S may be used if agreed between the designer and vehicle operator. However, the security coefficient S shall not be less than 1, 2. [A2]

[b] Zone 1: axle-body, plain bearing seats, transition fillets, bottom of grooves.

[c] Zone 2: wheel seats, disc-bearing seats, rolling bearing seats, pinion seats, collar-bearing surface.

Table 8 — Maximum permissible stresses for hollow axles in steel grade EA1N[9]

Intended use of the axle	[A2] Security coefficient S [a] [A2]	[A2] Zone 1[b] [A2] N/mm²	[A2] Zone 2[c] [A2] N/mm²	[A2] Zone 3[d] [A2] N/mm²	[A2] Zone 4[e] [A2] N/mm²
Powered axle with press-fit driving gear or pinion	1,5	133	73	63	53
Other cases	1,3	154	85	72	62

[A2] [a] Minimum value, unless measurements exist that demonstrate the loads are more precisely defined than those defined in this standard within an appropriate maintenance regime which maintains the track conditions, whereby a lower value of security coefficient S may be used if agreed between the designer and vehicle operator. However, the security coefficient S shall not be less than 1,2. [A2]

[b] Zone 1: axle body, plain bearing seats, fillets.

[c] Zone 2: all seats except journals and plain bearing seats.

[d] Zone 3: journal (beneath the rolling bearing).

[e] Zone 4: bore.

7.3 Steel grades other than EA1N

The fatigue limit shall be determined:

— on the surface of the axle body;

— beneath the fitting with equivalent interference conditions to those of the wheel seats.

In the case of a hollow axle, the fatigue limit shall also be determined:

[9] The values in this table are applicable if the journal bore diameter ratio is less than 3 or if the wheel seat/bore diameter ratio is less than 4.

— on the surface of the bore;

— on the journal with interference conditions equivalent to those of the bearings on the journal.

The test procedures to determine the fatigue characteristics are specified in EN 13260 and EN 13261.

The security coefficient value *S* by which the fatigue limits have to be divided to obtain the maximum permissible stresses is equal to:

$$S = 1{,}3 \text{ (or,5) x } \frac{q(other\ steel)}{q(steel\ EA1N)}$$

with $q = \frac{R_{fL}}{R_{fE}}$

1,3 (or 1,5): is the value of security coefficient for EA1N axles;

R_{fL} is the fatigue limit under rotating bending up to 10^7 cycles for unnotched test pieces;

R_{fE} is the fatigue limit under rotating bending up to 10^7 cycles for notched test pieces.

$$q\ (\text{EA1N steel}) = \frac{250 N/mm^2}{170 N/mm^2} = 1{,}47$$

q (for other steel grades) shall be determined with unnotched or notched test pieces of about 10 mm diameter. The geometric characteristics of the notch are given below (see Figure 13):

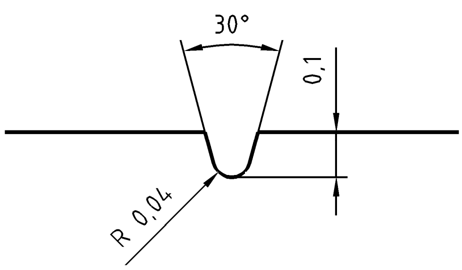


Figure 13

EXAMPLE: Steel grade EA4T (25CrMo4)

The fatigue limits for a solid axle are as follows:

— 240 N/mm² outside the fitting;

— 145 N/mm² beneath the fitting;

and for hollow axles:

— 240 N/mm² outside the fitting;

— 132 N/mm² beneath fitting, except journal;

— 113 N/mm² outside fitting on the journal;

— 96 N/mm² for the surface of the bore.

The value of security coefficient S is derived as follows:

R_{fL} = 350 N/mm²

R_{fE} = 215 N/mm²

q = 350/215 = 1,63

— powered axle with press-fit driving gear or pinion:

S = 1,5 x 1,63/1,47 = 1,66

— other cases:

S = 1,3 x 1,63/1,47 = 1,44

The maximum permissible stresses are given in the Tables 9 and 10.

Table 9 — Maximum permissible stresses for solid axles of steel grade EA4T

Intended use of the axle	[A2⟩ Security coefficient S [a] ⟨A2]	[A2⟩ Zone 1[b] ⟨A2] N/mm²	[A2⟩ Zone 2[c] ⟨A2] N/mm²
Powered axle with press-fit driving gear or pinion	1,66	145	87
Other cases	1,44	167	101

[A2⟩ [a] Minimum value, unless measurements exist that demonstrate the loads are more precisely defined than those defined in this standard within an appropriate maintenance regime which maintains the track conditions, whereby a lower value of security coefficient S may be used if agreed between the designer and vehicle operator. However the security coefficient value S shall not be less than 1,33. ⟨A2]

[b] Zone 1: axle-body, plain bearing seats, transition fillets, bottom of grooves.

[c] Zone 2: wheelseats, disc-bearing seats, rolling bearing seats, pinion seats, collar-bearing surface.

Table 10 — Maximum permissible constraints for hollow axles of steel grade EA4T[10]

Intended use of the axle	[A2⟩ Security coefficient S [a] ⟨A2]	[A2⟩ Zone 1[b] ⟨A2] N/mm²	[A2⟩ Zone 2[c] ⟨A2] N/mm²	[A2⟩ Zone 3[d] ⟨A2] N/mm²	[A2⟩ Zone 4[e] ⟨A2] N/mm²
Powered axle with press-fit driving	1,66	145	80	68	58

[10] The values in this table are applicable if the journal bore diameter ratio is less than 3 or if the wheel seat/bore diameter ratio is less than 4.

gear or pinion					
Other cases	1,44	167	92	78	67

[A2] [a] Minimum value, unless measurements exist that demonstrate the loads are more precisely defined than those defined in this standard within an appropriate maintenance regime which maintains the track conditions, whereby a lower value of security coefficient S may be used if agreed between the designer and vehicle operator. However the security coefficient value S shall not be less than 1,33. [A2]

[b] Zone 1: axle body, plain bearing seats, fillets.

[c] Zone 2: all seats except journals and plain bearing seats.

[d] Zone 3: journal (beneath the rolling bearing).

[e] Zone 4: bore.

Annex A
(informative)

Model of axle calculation sheet

Type
Drawing of axle N°
Drawing of wheel N°
Allocation
Material

Mass of wheelset (kg)
Axle
Wheels
Motor axle
Discs
Miscellaneous
Total (m_2)

Mass on rail per axle: $m_1 + m_2$ (kg)

Dimensions (mm)

$h =$

$s =$

$R =$

h_1

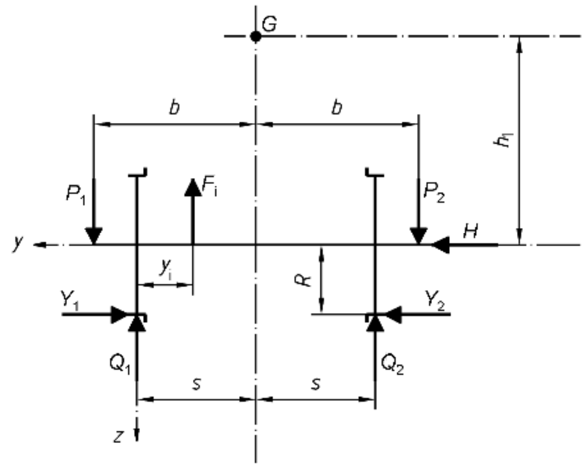
Forces (N)

$P_1 =$

$P_2 =$

$Y_1 =$

$Y_2 =$



Key

G centre of gravity of the vehicle

Figure A.1

$$Q_1 = \frac{1}{2s}\left[P_1(b+s) - P_2(b-s) + (Y_1 - Y_2)R - \sum F_i(2s - y_i)\right]$$

y_i (mm)	F_i (N)	Part

Method of braking		
P' (N)		
F_f (N)		
Γ		

1) for hollow axles:

on the surface: $\dfrac{32K10^6 d}{\pi(d^4 - d'^4)}$

in the bore: $\dfrac{32K10^6 d'}{\pi(d^4 - d'^4)}$

Section	y mm	d mm	d' mm	D mm	r mm	$\frac{r}{d}$	$\frac{D}{d}$	K	1) $\frac{32K10^6}{\pi d^3}$	M_x (Nmm) × 10^-6	M'_x (Nmm) × 10^-6	M'_z (Nmm) × 10^-6	M'_y (Nmm) × 10^-6	MR (Nmm) × 10^-6	σ (N/mm²)	σ_{max} (N/mm²)

Annex B
(informative)

Procedure for the calculation of the load coefficient for tilting vehicles

According to Table 3, $H = \beta m_1 g = 0{,}175 m_1 g$.

In general terms, factor $\beta = 0{,}175$ comprises a quasi-static centrifugal force percentage due to the unbalanced transverse acceleration a_q and a thrust factor f_q .

The usual unbalanced transverse acceleration of a_q =1,0 m/s^2 results in a transverse force factor of 0,1 (g , rounded up to 10 m/s^2) to take into account the quasi-static centrifugal force.

For the analysis performed for ORE B 136, an unbalanced transverse acceleration of a_q = 1,0 m/s^2 was used by DB and 1,3 m/s^2 by SNCF.

The result of these tests led to a value being derived of $f_q = 0{,}075$;

The following is an example for vehicles with curved track dependent superstructure control.

The traction unit will be designed for a transverse acceleration of a_q = 2,0 m/s^2 resulting from a cant deficiency. Thus, the following coefficient results for every axle in the scope of this standard:
:

$$\beta = a_q / 10 + f_q = 0{,}2 + 0{,}075 = 0{,}275$$

NOTE The dynamic part of the factor β in the formula does not differ between tilting and non-tilting vehicles. However, the dynamic factor varies as a function of the track speed and quality. Since $Y_2 = 0{,}175 m_1 g$ remains true - as Y_2 takes into account the transverse friction on the curved track inner wheel - it results from the relationship $Y_1 = Y_2 + H$ that:

$Y = 0{,}45 m_1 g$;

(The guiding force between the wheel and rail does not change, whether the tilting method is used or not).

The following formulae (see Table B.1) result from this for calculation of the forces.

Table B.1

<table>
<tr><td>For all wheelsets coming within the scope of this standard, for standard gauge and for vehicles with curved track dependent superstructure control</td><td>$P_1 = (0{,}625 + 0{,}275 h_1 / 2b) m_1 g$
$P_2 = (0{,}625 - 0{,}275 h_1 / 2b) m_1 g$
$Y_1 = Y_2 + H = 0{,}45 m_1 g$
$Y_2 = 0{,}175 m_1 g$
$H = 0{,}275 m_1 g$</td></tr>
<tr><td colspan="2">For all wheelsets
$$Q_1 = \frac{1}{2s}[P_1(b+s) - P_2(b-s) + (Y_1 - Y_2)R - \sum_i F_i(2s - y_i)$$
$$Q_2 = \frac{1}{2s}[P_2(b+s) - P_1(b-s) - (Y_1 - Y_2)R - \sum_i F_i y_i]$$</td></tr>
</table>

Annex C
(informative)

Values of forces to take into consideration for wheelsets for reduced gauge track (metric or close to a metre)

The following formulae (see Table C.1) are applicable for calculating forces, except for tilting vehicles.

Table C.1

For all wheelsets in the scope of this document	$P_1 = (0{,}65 + 0{,}114\ h_1 / b)\ m_1g$ $P_2 = (0{,}65 - 0{,}114\ h_1 / b)\ m_1g$ $Y_1 = 040\ m_1g$ $Y_2 = 0{,}175\ m_1g$ $H = Y_1 - Y_2 = 0{,}225\ m_1g$
For all wheelsets with wheels press-fitted onto the axle	$Q_1 = \frac{1}{2s}\ [P_1\ (b+s) - P_2\ (b-s) + (Y_1-Y_2)\ R - \Sigma F_i\ (2s-y_i)]$ $Q_2 = \frac{1}{2s}\ [P_2\ (b+s) - P_1\ (b-s) - (Y_1-Y_2)\ R - \Sigma F_i\ y_i)]$

Annex D
(normative)

Method for determination of full-scale fatigue limits for new materials

D.1 Scope

This Annex describes the requirements to be met and the procedure to be followed to characterize the fatigue limits of full-size axles for the steel grades not specified in EN 13260 and EN 13261. This procedure makes it possible to compare results from different laboratories.

The fatigue limits obtained are then used to determine the permissible stresses for the design of axles according to the procedure described in EN 13103 and this standard.

D.2 General requirements for the test pieces

The test pieces shall meet the requirements of the relevant ENs (geometry, roughness, mechanical properties etc.). All these parameters shall be verified in a summary table. The test pieces used shall be representative of axles of normal fabrication and use the same fabrication method (material quality, surface finish quality, reduction ratio, non-destructive testing, etc.). However, they can be configured specifically for the test.

D.3 General requirements for test apparatus

The test bench to be used shall allow a rotating bending moment with a constant stress amplitude to be applied to the section tested. A typical configuration is shown in Figure D.1. During the test, it shall be ensured by means of constant monitoring of the relevant measurements that the nominal stress amplitudes applied remain constant within a range of ± 5 MPa.

The main method of controlling the test bench is based on the applied load, the applied stress and the applied movement; for this parameter, it is recommended verifying the uncertainty in order to ensure that the maximum error agreed above on the nominal stress applied is not exceeded.

NOTE If a symmetrical test bench and symmetrical test piece is used, it is possible to regard two sections as having been tested (if they are correctly checked during the test).

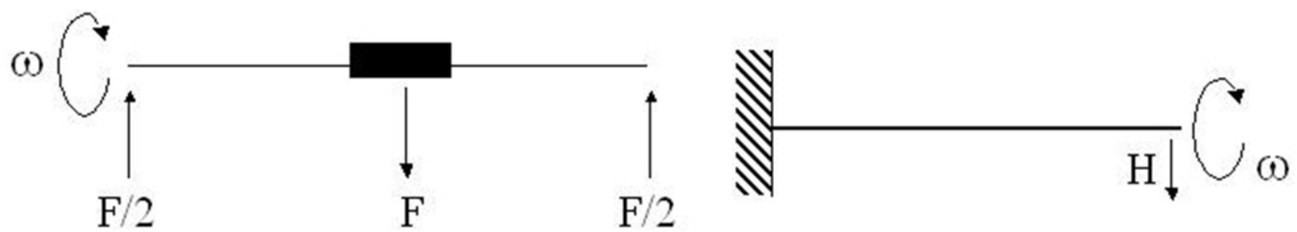


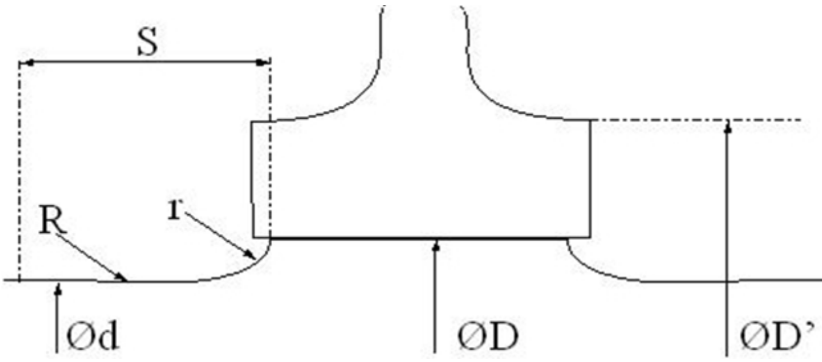
Figure D.1 — Examples of test configurations

D.4 Axle body fatigue limit ("F1")

D.4.1 Geometry

The dimensions of the test pieces shall be similar to the dimensions of the axles produced under normal conditions; the minimum dimensions are given in Figure D.2.

R	75
r	15
S	≥ 35
D/d	≥ 1,15
d	≥ 150
D	≤ 200
Wheel seat roughness Ra	0,8 – 1,6
Body roughness Ra	0,8 – 3,2
D`/d	1,3 – 1,5



Key
d: body diameter
D: wheel seat diameter
D': hub diameter
R and r: body-seat transition radii
S: transition fillet length

Figure D.2 — Test piece geometry

NOTE Too small a diameter ratio (D/d) would produce cracks in the wheel seat; the value at which a crack will not result in the seat but in the body depends on the fatigue strength of the axle steel (the value of the diameter ratio is higher the greater is the fatigue strength F1).

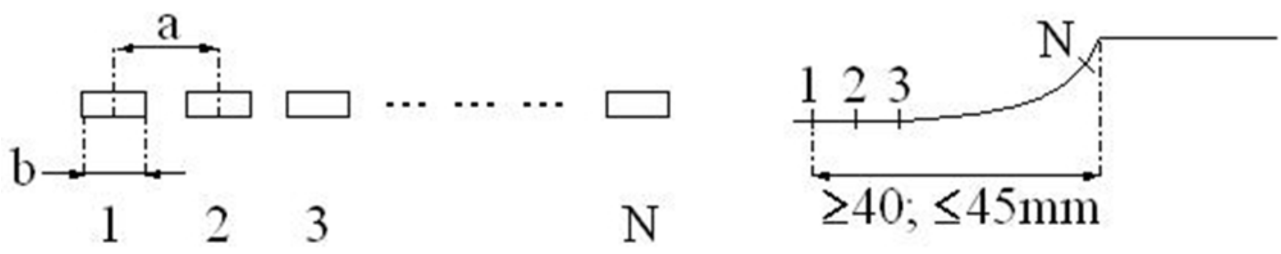
The thickness of the hub and the interference fit between the hub and seat will determine the additional stresses on the basis of the axle body fillet; therefore, the transition diameters should be similar to the typical configurations.

D.4.2 Verification of the applied stress

Regardless of the type of test bench, the maximum stress applied shall be verified experimental means with regard to the maximum value and the longitudinal position of the maximum value.

The stress values applied shall be measured by strain gauges in the zone where the initial fatigue cracks appear.

This is done by a range of strain gauges placed along the transition fillet with the axle seat supporting the maximum stress value (see Figure D.3); it is recommended that the distance between the strain gauges should not exceed 4 mm and the gauge length should not exceed 3 mm.



Key

1,2,3,…N: strain gauges
a: distance between two gauges
b: gauge length

Figure D.3 — Strain gauge instrumentation

In order to be consistent with the axle design method, the stress is determined under the assumption that the stress is mono-axial: $\sigma_{actual}=E*\varepsilon$

For the shape of the axle tested, the additional static stress factor shall be determined: $k_t=(\sigma_{actual})/\sigma_{nom}$

σ_{nom} is the nominal stress for the section where the actual stress measured is the maximum. It is determined either using the axle design method based on the beam theory if the applied force is measured or by extrapolation of the strain gauge measurements over two sections of the axle where the longitudinal stresses vary linearly.

The fatigue limit is determined both for the stress actually measured and for the nominal stress that depends strictly on the axle geometry (D, d, r).

D.4.3 End of test criterion

For each limit, it shall be verified that no crack was observed after 10^7 cycles under load, creating a surface stress equal to the test value.

D.4.4 Détermination of the fatigue limit

The statistical method to be applied to determine the fatigue limit is the STAIR CASE method.

It is recommended that the number of axles to be tested should be 15 from at least three different melts.

The stress interval is 10 MPa.

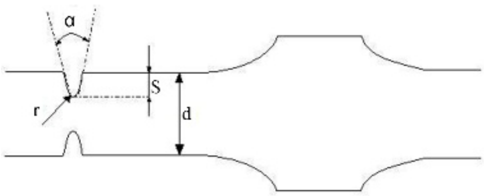
The probability of non-cracking shall be calculated and indicated in the test report. In all cases, this value should be comparable to those used for the usual materials.

D.5 Axle bore fatigue limit ("F2")

D.5.1 Geometry

The axle used for the test is notched to simulate the worst scratch that the bore-making procedure may leave. The notch is machined on the external body with a special cutting tool according to the geometric parameters given in Figure D.4.

d	≤ 140
α	30
S	1
r	0,04



Key
α : notch angle
S : notch depth
R : radius at notch bottom
D : test piece diameter

Figure D.4 — Test piece geometry

D.5.2 Verification of the applied stress

The stress to be considered is the nominal stress (σ_{nom}) in the section where the notch is located.

The stress shall be determined by experimental means on the tested axle either using the axle design method based on the beam theory if the applied force is measured or by extrapolation of the strain gauge measurements over the two sides of the notch where the longitudinal stresses vary linearly.

D.5.3 End of test criterion

For each limit, it shall be verified that no crack has appeared after 10^7 cycles of a load creating a surface stress equal to the value under test.

D.5.4 Determination of the fatigue limit

The statistical method to be applied to determine the fatigue limit is the STAIR CASE method.

It is recommended that the number of axles to be tested should be 15 from at least three different melts.

The stress interval is 10 MPa.

The probability of the absence of a defect shall be calculated and indicated in the test report. In all cases, this value should be comparable to those used for the usual materials.

D.6 Wheel seat fatigue limit ("F3 and F4")

D.6.1 Geometry

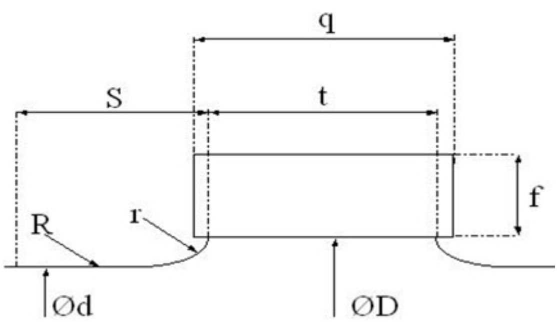
F3 refers to solid axles (without bore) and F4 to bored axles.

The test piece dimensions shall be similar to the dimensions of normally-produced axles; the range of dimensions is given in Figure D.5.

The actual fatigue limit of the fitting zones on the axle depends on the various geometric parameters, in particular the diameter ratio D/d: for a given nominal stress applied to the end of the seat, the increase in the diameter ratio reduces the actual longitudinal stress at the end of the seat. Therefore, the nominal fatigue limit also increases. Beyond a certain diameter ratio value, the cracks appear on the body and no longer on the seat (see Figure D.6).

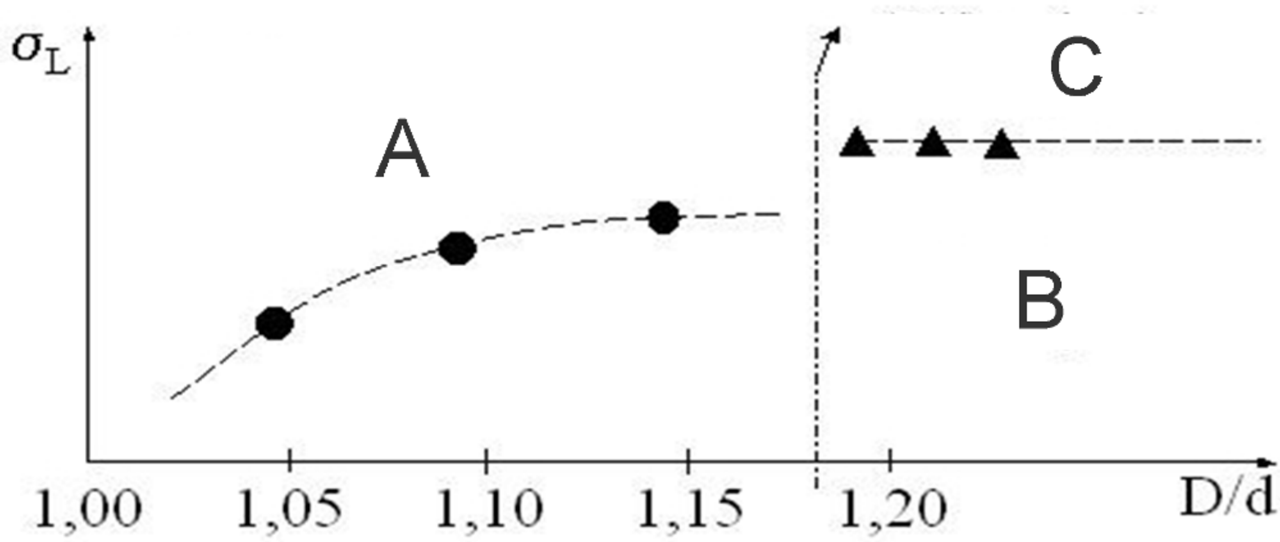
To obtain an overall view of the fatigue limits F3 and F4, it would be useful to carry out tests for different diameter ratios (at least three). By interpolating these values and by knowing the fatigue limit of the body F1, it is possible to determine the critical ratio D/d beyond which the cracks appear on the body and below which they appear on the seat. This is important information for the design of axles made of new materials ensuring that the cracks appear on the body rather than on the seat where it is more difficult to detect them by ultrasonic inspection.

D	≤ 140
Ø	30 °
S	1
R	0,04



Key:
f :ring thickness
t :seat length
q : ring length

Figure D.5 — Geometric parameters for F3 and F4



Key

A cracks in the wheel seat
B cracks in the body
C (D/d) optional

Figure D.6 — Effect of diameter ratio D/d

D.6.2 Verification of the applied stress

To be consistent with the axle design method, the stress to be considered is the nominal stress (σ_{nom}) 10 mm from the end of the wheel seat.

The stress shall be determined by experimental means on the tested axle either using the axle design method based on the beam theory if the applied force is measured or by extrapolation of the strain gauge measurements over the two sides of the notch where the longitudinal stresses vary linearly

The stress level shall be determined using the dimension actually measured for the critical section.

D.6.3 End of test criterion

For each limit, it shall be verified that no crack has been detected after 10^7 cycles at a load creating a surface stress equal to the test value.

D.6.4 Determination of the fatigue limit

The first stage consists of determining the interpolation curve and finding the critical ratio D/d. A minimum of three test pieces may be used for each D/d value. The stress limit to be considered is the highest stress level without cracking for any test piece.

When the critical D/d value is reached, a second stage consists of applying the STAIR CASE method with 15 test pieces to determine the fatigue limit for this ratio D/d.

The stress interval is10 MPa.

The probability of the absence of a crack shall be calculated and indicated in the test report. In all cases, this value should be comparable to those used for the usual materials.

D.7 Content of the test report

A test report shall be presented containing results and analysis for each fatigue limit. This report shall record all the conditions and parameters used for carrying out the tests. It shall contain the following information:

a) a description of the material subjected to the test (general mechanical properties, fabrication procedure, heat treatment, material quality, surface finish quality, reduction ratio, etc.);

b) detailed full-scale diagrams of the test piece and other elements fitted for the test (the information on the diagrams shall meet the requirements of the relevant subclauses of the standards on the component - roughness, tolerances, etc.);

c) description of the fitting procedure and results of the related tests;

d) serial number of the test piece (the serial number shall also permit identification of the melt);

e) records of the test carried out on the test pieces according to 3.4.2 and 3.5 to 3.8 of the main body of EN 13261:2009;

f) methods used to verify the stress, to measure the stress and to extrapolate the values in the critical zones (in the cases required in the above subclauses);

g) description of the full measurement chain and characteristics of the added components; indication of keeping within the measurement tolerances and accuracy level;

h) inspection report for each test piece at the end of each stress step;

i) description and analyse of the crack where a test piece has cracked.

The test report shall be part of a file including:

— records indentifying each mechanical property defined in 3.2.1, 3.2.2, 3.3 and 3.4.1 of the main body of EN 13261:2009 (from batches);

— certificate of conformity to EN ISO/IEC 17025 for the laboratory(ies) that carried out the tests.

Annex ZA
(informative)

A1 Relationship between this European Standard and the Essential Requirements of EU Directive 2008/57/EC

This European Standard has been prepared under a mandate given to CEN/CENELEC/ETSI by the European Commission to provide a means of conforming to Essential Requirements of the New Approach Directive 2008/57/EC[11].

Once this standard is cited in the Official Journal of the European Union under that Directive and has been implemented as a national standard in at least one Member State, compliance with the clauses of this standard given in Table ZA.1 for High Speed Rolling Stock, Table ZA.2 for Conventional Rail Locomotives and Passenger Rolling Stock, confers, within the limits of the scope of this standard, a presumption of conformity with the corresponding Essential Requirements of that Directive and associated EFTA regulations.

Table ZA.1 – Correspondence between this European Standard, the HS RST TSI published in the OJEU and dated 26 March 2008 and Directive 2008/57/EC

Clauses/subclauses of this European Standard	Chapters/subclauses/annexes of the TSI	Corresponding text, articles/subclauses/annexes of Directive2008/57/EC	Comments
The whole standard is applicable	4. Characteristics of the sub-system 4.2 Functional and technical specifications of the sub-system 4.2.3. Vehicle track interaction and gauging 4.2.3.4 1. Rolling stock dynamic behaviour. General	Annex III, Essential requirements 1 General requirements 1.1 Safety 1.1.1, 1.1.3 1.5 Technical compatibility §1 2 Requirements specific to each sub-system 2.3 Control-command and signalling 2.3.2 Technical compatibility §1 2.4 Rolling stock 2.4.2 Reliability and availability	

[11] The Directive 2008/57/EC adopted on 17 June 2008 is a recast of the previous Directive 96/48/EC 'Interoperability of the trans-European high-speed rail system' and 2001/16/EC 'Interoperability of the trans-European conventional rail system' and their revision by Directive 2004/50/EC of the European Parliament and of the Council of 29 April 2004 amending Council Directive 96/48/EC 'Interoperability of the trans-European high-speed rail system' and Directive 2001/16/EC of the European Parliament and of the Council 'Interoperability of the trans-European conventional rail system'

Table ZA.2 – A2⟩ Correspondence between this European Standard, the Conventional Rail TSI Locomotives and Passenger RST published in the Official Journal of the European Union on 26 May 2011 and Directive 2008/57/EC ⟨A2

Clauses/subclauses of this European Standard	Chapter/subclauses/annexes of the TSI	Corresponding text, articles/subclauses/annexes of Directive2008/57/EC	Comments
The whole standard is applicable	4. Characterization of the rolling stock sub-system 4.2 Functional and technical specification of the sub-system 4.2.3. Vehicle track interaction and gauging 4.2.3.5.2.1 Wheelsets. Mechanical and geometrical characteristics of wheelsets	Annex III, Essential requirements 1 General requirements 1.1 Safety 1.1.1, 1.1.3 1.5 Technical compatibility §1 2 Requirements specific to each subsystem 2.3 Control-command and signalling 2.3.2 Technical compatibility §1 2.4 Rolling stock 2.4.2 Reliability and availability	Clauses 4, 5, 6 and 7 of EN 13104:2009 are quoted in the TSI and therefore are regulatory in nature. A2⟩ Clauses 4, 5 and 6 of this amended version of EN 13104 remain the same, and clause 7 remains technically the same, correcting the inaccuracies of the previous version. This was the sole aim of this amended edition of EN 13104. ⟨A2

WARNING — Other requirements and other EU Directives may be applicable to the product(s) falling within the scope of this standard. ⟨A1

A1⟩ *text deleted* ⟨A1

Bibliography

[1] ORE report No.11, *Calculation of wagon and coach axles (from committee B136)*

[2] UIC 515-3, *Railway rolling stock - Bogies - Running gear “Method of calculation for designing axles”*

[3] NF F 01-118, *Railway rolling stock - Axles with outside axle journals – Design rules and calculation method*

[4] EN ISO/IEC 17025, *General requirements for the competence of testing and calibration laboratories (ISO/IEC 17025:2005)*

[5] EN 13103, *Railway applications - Wheelsets and bogies – Non-powered axles – Design method*

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