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Railway applications – Wheelsets and bogies – Powered and non-powered axles with inboard bearings – Design method

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Contents

Summary of pages

This document comprises a front cover, an inside front cover, pages i to iv, pages 1 to 36, an inside back cover and a back cover.

Foreword

Publishing information

This British Standard is published by BSI Standards Limited, under licence from The British Standards Institution, and came into effect on 31 August 2011. It was prepared by Technical Committee RAE/3, *Railway rolling stock material*. A list of organizations represented on this committee can be obtained on request to its secretary.

Information about this document

The start and finish of text introduced or altered by Corrigendum No. 1 is indicated in the text by tags C1〉 〈C1.

Test laboratory accreditation. Users of this British Standard are advised to consider the desirability of selecting test laboratories that are accredited to BS EN ISO/IEC 17025 by a national or international accreditation body.

Presentational conventions

The provisions of this standard are presented in roman (i.e. upright) type. Its requirements are expressed in sentences in which the principal auxiliary verb is "shall".

Commentary, explanation and general informative material is presented in smaller italic type, and does not constitute a normative element.

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This publication does not purport to include all the necessary provisions of a contract. Users are responsible for its correct application.

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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 (Union International des Chemins de fer) started to harmonize these methods at the beginning of the 1970s. This led to an ORE (Office de Recherches et d'Essais de l'UIC) document applicable to the design of trailer stock axles [1], which was subsequently incorporated into national standards (French, German, and 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 an UIC leaflet [2].

Work continued in Europe and standards were prepared under a mandate given to CEN/CENELEC/ETSI by the European Commission and the European Free Trade Association, which supports the essential requirements of Directives 96/48/EC [3] and 2001/16/EC [4] amended by Directive 2004/50/EC [5]. These Directives have now been superseded by Directive 2008/57/EC [6].

The resulting European standards BS EN 13103 and BS EN 13104 for the design methods for outboard bearing axles were derived based largely on the UIC leaflet [2].

This British Standard takes these considerations into account and they have been adopted and amended accordingly, for the axle configurations having inboard journal bearings. UK practice, including TM/TC0001 *Design guide for the calculation of stresses in axles with inboard journals* [7], BASS 503 *Design guide for the calculation of stresses in axles driven by axle hung traction motors* [8] and BASS 504 *Design guide for the calculation of stresses in non-driving axles* [9], has also been used in the preparation of this document.

The method described in these reference documents is largely based on conventional loadings for axles and applies the beam theory for the stress calculation. The shape and stress recommendations are derived from design verification, and the outcome is validated by many years' experience on the UK mainline railway system.

1 Scope

This British Standard specifies a design method for powered and non-powered axles with inboard bearings for railway applications.

This standard:

- specifies the forces and moments to be taken into account with reference to masses, traction and braking conditions;
- specifies the stress calculation method for axles with inboard axle journals;

- specifies the maximum permissible stresses to be assumed in calculations for steel grades EA1N, EA1T and EA4T as defined in BS EN 13261 and steel grades A1N and A1T as specified in BS 5892-1;
- specifies a method for determining the diameters for the various sections of the axle. It also gives recommendations for 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;
- axles defined in BS EN 13261 and in BS 5892-1;
- UK standard gauge.

This standard is applicable to axles fitted to rolling stock intended to run under normal UK mainline 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.

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.

BS 5892-1, *Railway rolling stock materials – Part 1: Specification for axles for traction and trailing stock*

BS EN 13261, *Railway applications – Wheelsets and bogies – Axles – Product requirements*

3 Symbols

For the purposes of this British Standard, the following symbols apply.

m_1 Mass on journals (including bearings and axle boxes), in kilograms (kg)

m_2 Wheelset mass and masses on the wheelset between running surfaces (brake disc, etc.), in kilograms (kg)

$m_1 + m_2$ For the wheelset considered, portion of the mass of the vehicle on the rails, in kilograms (kg)

g Acceleration due to gravity, in metres per second squared (m/s^2)

P Half the vertical force per wheelset on the rail, $\frac{(m_1 + m_2)g}{2}$, in newtons (N)

P_0	Vertical static force per journal when the wheelset is loaded symmetrically, $\frac{m_1 g}{2}$, in newtons (N)
P_1	Vertical force on the more heavily-loaded journal, in newtons (N)
P_2	Vertical force on the less heavily-loaded journal, in newtons (N)
P'	Portion of P braked by any mechanical braking system, in newtons (N)
Y_1	Wheel/rail horizontal force perpendicular to the rail on the side of the more heavily-loaded journal, in newtons (N)
Y_2	Wheel/rail horizontal force perpendicular to the rail on the side of the less heavily-loaded journal, in newtons (N)
H	Force balancing the forces Y_1 and Y_2, in newtons (N)
Q_1	Vertical reaction on the wheel situated on the side of the more heavily-loaded journal, in newtons (N)
Q_2	Vertical reaction on the wheel situated on the side of the less heavily-loaded journal, in newtons (N)
F_i	Forces exerted by the masses of the unsprung elements situated between the two wheels [brake disc(s), pinion, etc.], in newtons (N)
F_f	Maximum force input of the brake shoes of the same shoe holder on one wheel or interface force of the brake pads on one brake disc, in newtons (N)
M_x	Bending moment due to the masses in motion, in newton millimetres (N·mm)
M'_x, M'_z	Bending moments due to braking, in newton millimetres (N·mm)
M'_y	Torsional moment due to braking, in newton millimetres (N·mm)
M''_x, M''_z	Bending moments due to traction, in newton millimetres (N·mm)
M''_y	Torsional moment due to traction, in newton millimetres (N·mm)
MX, MZ	Sum of bending moments, in newton millimetres (N·mm)
MY	Sum of torsional moments, in newton millimetres (N·mm)
MR	Resultant moment, in newton millimetres (N·mm)
$2b$	Distance between vertical force input points on axle journals, in millimetres (mm)
$2s$	Distance between wheel treads, in millimetres (mm)
h_1	Distance between the centre of the axle and the centre of gravity of the vehicle, in millimetres (mm)
y_i	Distance between the tread of one wheel and force F_i, in millimetres (mm)
y	Abscissa for any section of the axle calculated from the section subject to force P_1, in millimetres (mm)

Γ	Average friction coefficient between the wheel and the brake shoe or between the brake pads and the brake disc
σ	Stress calculated in one section, in newtons per square millimetre (N/mm^2)
σ_{max}	Maximum stress calculated in one section, due to the masses in motion plus braking or masses in motion plus traction, whichever is the greater, in newtons per square millimetre (N/mm^2)
K	Fatigue stress concentration factor
R	Nominal radius of the tread of a wheel, in millimetres (mm)
R_b	Radius of circle on brake disc at which brake force is applied, in millimetres (mm)
d	Diameter for one section of the axle, in millimetres (mm)
d'	Bore diameter of a hollow axle, in millimetres (mm)
D	Diameter used for determining K, in millimetres (mm)
r	Radius of transition fillet or groove used to determine K, in millimetres (mm)

4 General

The major phases for the design of an axle shall be undertaken as follows:

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) verification of the options taken by:

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

NOTE A sample axle calculation sheet template is given in Annex A and some example calculations are given in Annex B to Annex H.

5 Forces and moments to be taken into consideration

5.1 Types of forces

Three types of forces shall be taken into consideration as a function of:

- the masses in motion;
- the braking system;
- the traction.

5.2 Influence of masses in motion

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

Unless otherwise specified (for example by the customer), the masses ($m_1 + m_2$) to be taken into account for the main types of rolling stock shall be as specified in Table 1.

NOTE 2 For particular applications, e.g. suburban vehicles, other masses might be necessary in accordance with the specific operating requirements (see Table 1 Footnotes [A)] and [B)]).

Figure 1 **Influence of masses in motion**

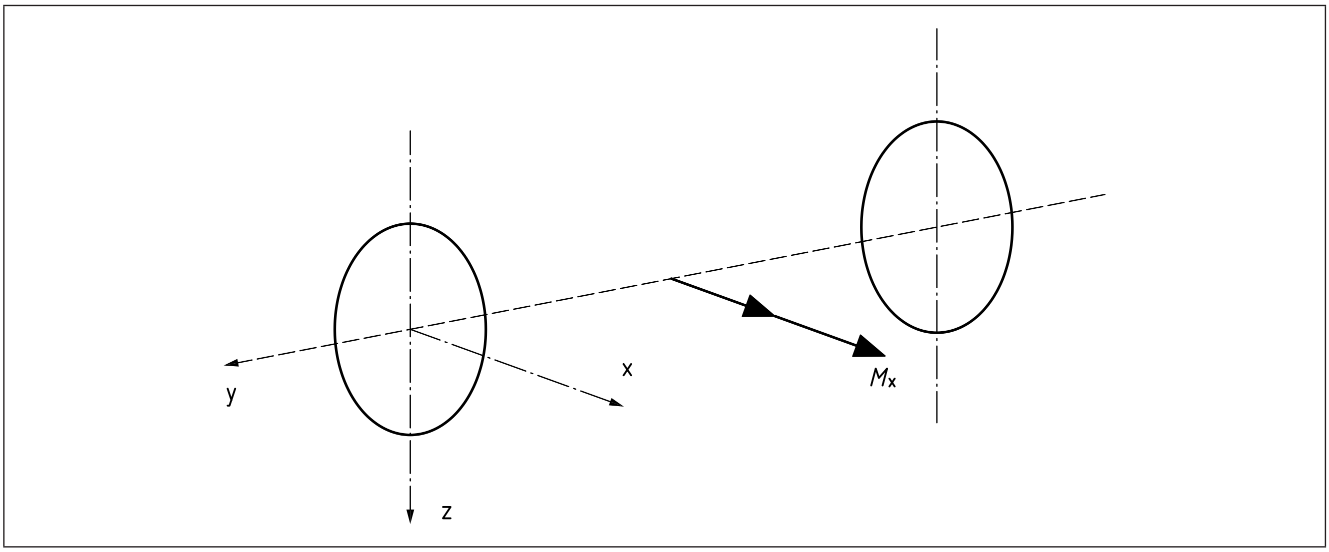


Table 1 **Masses to be taken into account for the main types of rolling stock**

Type of rolling stock units	**Mass** ($m_1 + m_2$)
Freight wagons	For the axle considered, portion of the wagon mass under maximum permissible loading in service
Traction units with no passenger accommodation, luggage areas and postal vans	For the axle considered, portion of the vehicle mass under maximum permissible loading in service
Traction units including passenger accommodation, luggage areas and postal vans	For the axle considered, portion of the vehicle mass under maximum permissible loading in service
1 – Main line vehicles [A)]	Total mass = Mass in service + 1.2 × payload Where: Mass in service is the vehicle mass without passengers, tanks full (of water, sand, fuel, etc.) Payload is the mass of passengers [B)] and luggage, taking the mass of a passenger with hand luggage as 80 kg and assuming: • 1 passenger per seat; • 2 passengers/m^2 in corridors and vestibules; • 2 passengers per attendant compartment; • 300 kg/m^2 in luggage compartments.
2 – Suburban vehicles [A) C)]	Total mass = Mass in service + 1.2 × payload Where: Mass in service is the vehicle mass without passengers, tanks full (of water, sand, fuel, etc.) Payload is the mass of passengers [B)] and luggage, taking the mass of a passenger, with little or no hand luggage, as 70 kg, and assuming: • 1 passenger per seat; • 3 passengers/m^2 in corridor areas; • 4 or 5 passengers/m^2 in vestibule areas [C)]; • 300 kg/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 square metre in corridors and vestibules.

B) The standing passenger loadings in this table have been derived for UK application using the options available in BS EN 15663.

C) These vehicles are sometimes associated with different classes of passenger travel, i.e. 1st or 2nd class, in which there are different densities of passenger seating, and different numbers of standing passengers.

5.3 Load cases

5.3.1 General

The following load cases 1 and 2 shall be used to ensure that axles are not under designed.

NOTE It is important that these two load cases are considered independently and that the worst load case is used for the axle design against the permissible limits given in this standard.

5.3.2 Load case 1: Straight track case

*COMMENTARY ON **5.3.2***
If the lateral loads ordinarily applied for outboard journal axles in the high speed curving case were to be applied to inboard journal bearing axles, then it will reduce the overall bending moment when applied to that of the bending moment due to vertical load. Additionally, the vertical forces from the sinusoidal running on straight track should also not be applied.

Parameters for load case 1:

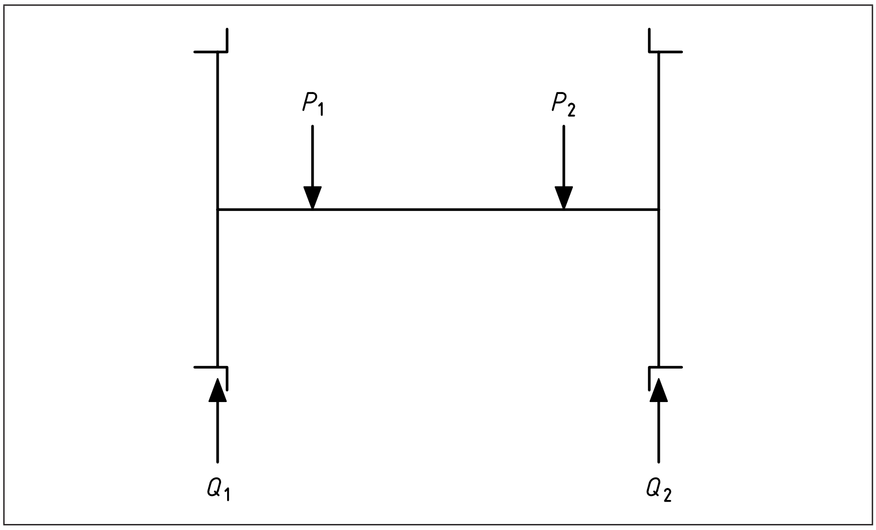
Vertical acceleration:

- Quasi static 1g
- Dynamic 0.6g
- Total 1.6g

Lateral acceleration: not applied.

$P_1 = P_2 = 0.8 m_1 g$ (for load balance see Figure 2)

Figure 2 **Load balance for load case 1**



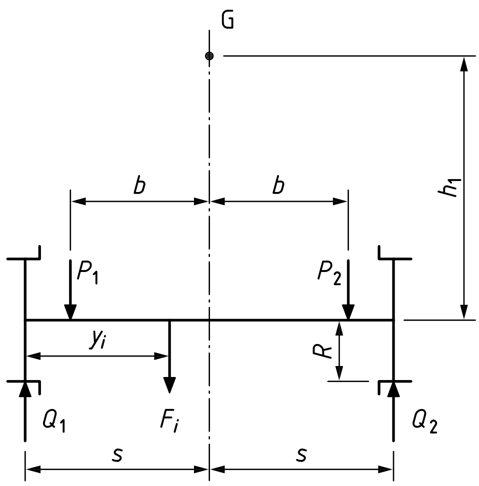
The bending moment M_x in any section shall be calculated from forces P_1, P_2, Q_1, Q_2 and F_i as shown in Figure 3.

The value of the forces F_i shall be determined from multiplying the mass of each unsprung component by an appropriate acceleration level. This will be significantly greater than 1*g* and the level shall be defined in the design.

NOTE 1 This 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.*

Figure 3 **Forces for calculating bending moment M_x**



Key

G Centre of gravity of vehicle

$P_1 = P_2 = 0.8m_1g$

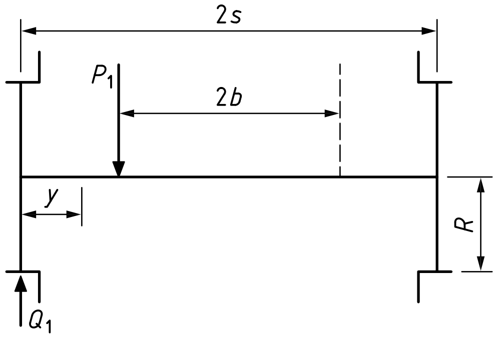
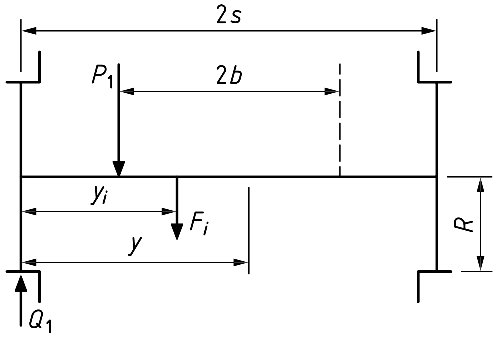
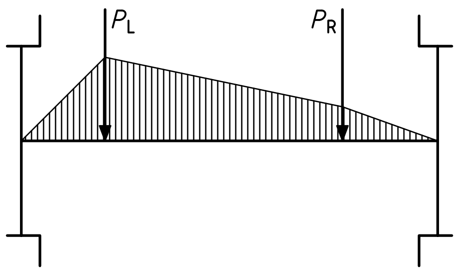
$Q_1 = [P_1(s + b) + P_2(s - b) + \sum F_i (2s - y_i)]/2s$

$Q_2 = [P_1(s - b) + P_2(s + b) + \sum F_i y_i]/2s$

The formulae to be used to calculate M_x for each zone of the axle shall be as specified in Table 2.

NOTE 2 Table 2 also gives the general outline of variations in M_x along the axle.

Table 2 **Calculation of M_x for each axle zone and general outline of variations in M_x along the axle**

Zone of axle	M_x A)
Between running surface and loading plane	$M_x = Q_1 y$ 
Between loading planes	$M_x = Q_1 y - P_1\{y-(s-b)\} - \sum F_i(y-y_i)$ 
General outline of M_x variation	

A) For a non-symmetrical wheelset, the calculations shall be carried out after applying the load alternately to the two journals to determine the worst case.

5.3.3 Load case 2: Low speed curving case

*COMMENTARY ON **5.3.3***
This load case simulates check rail contact on the flange back when negotiating curves at low speed. Therefore, dynamic factors are reduced. The height of the centre of gravity has been taken into consideration as this value varies considerably with the type of vehicle. This load case occurs infrequently but it needs to be taken into consideration.

For the calculation of Y_2, the formula from BS EN 13979-1, where $Y = 0.42P$ which equals $0.42(m_1 + m_2)g$, has been modified to use $Y_2 = 0.21m_1g$ in order to be in line with Y_1, Y_2 and H as given in BS EN 13103 and BS EN 13104.

Parameters for load case 2:

Vertical loading:

- Vertical dynamic factor 1.125 (see Note 1)

Lateral loading:

- $H = 0.075m_1g$ (corresponding to 7.5% cant deficiency) (see Note 2)
- $Y_1 = Y_2 - H$
- $Y_2 = 0.21m_1g$

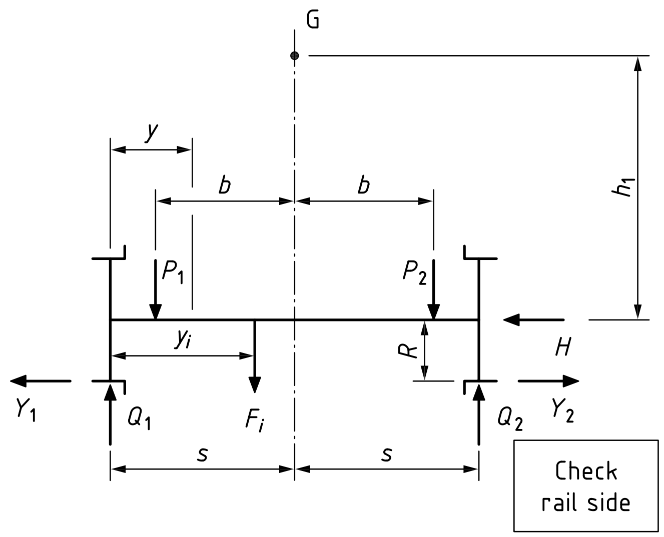
NOTE 1 Reduced from 1.25, which is the value given in BS EN 13103 and BS EN 13104, owing to low speed.

NOTE 2 Reduced from 17.5% cant deficiency which is the value given in BS EN 13103 and BS EN 13104, owing to low speed

NOTE 3 Experience has shown that for the values given for the vertical and lateral loading there is a good correlation between calculated stress and actual stress measured during track testing.

NOTE 4 The load balance for load case 2 is shown in Figure 4.

[C1› Figure 4 **Load balance for load case 2**



Key

G Centre of gravity of vehicle

$P_1 = [(1.125/2) + (0.075h_1/2b)]m_1g$

$P_2 = [(1.125/2) - (0.075h_1/2b)]m_1g$

$H = 0.075m_1g$

$Y_1 = Y_2 - H$

$Y_2 = 0.21m_1g$

$Q_1 = [P_1(s + b) + P_2(s - b) + (Y_2 - Y_1)R + \sum F_i\,(2s - y_i)]/2s$

$Q_2 = [P_1(s - b) + P_2(s + b) + (Y_1 - Y_2)R + \sum F_i\, y_i]/2s$

‹C1]

5.4 Effects due to braking

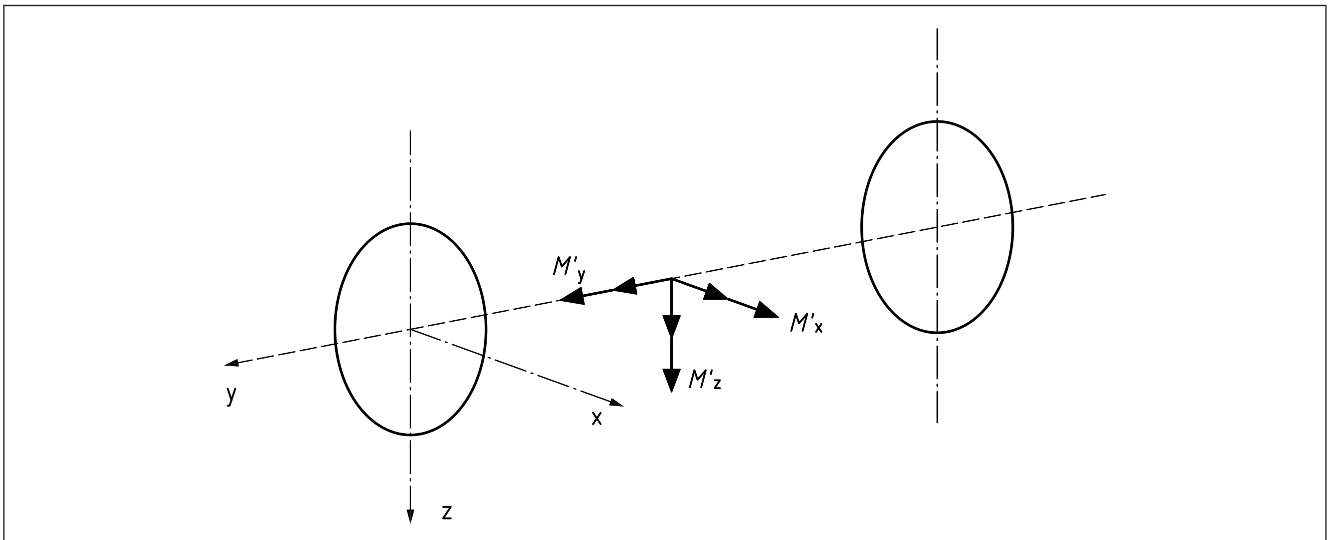
Braking generates moments which shall be considered to be represented by the three components: M'_x, M'_y and M'_z (see Figure 5).

The calculation of the values of the components M'_x, M'_y and M'_z for different methods of braking shall be in accordance with Table 3.

If several methods of braking are superimposed, the values corresponding to each of the methods shall be added together e.g. friction braking and electric braking.

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 3. Special attention should be paid to the calculation of the M'_x component, which should be added directly to the M_x component representing masses in motion.

Figure 5 **Moments generated by braking**



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.

5.5 Effects due to curving and wheel geometry

For an unbraked wheelset, the torsional moment M'_y shall be taken to be equal to 0.2*PR* to account for possible differences in wheel diameters and the effect of passing through curves.

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

5.6 Effects due to traction

*COMMENTARY ON **5.6***
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 maximum braking moments do not occur simultaneously.

5.6.1 General

The axle design shall 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 as given in **5.2**, **5.3**, **5.4** and **5.5**;

b) with the following starting conditions:

    1) effects due to masses in motion as given in **5.6.2**;

    2) effects due to starting driving torque.

The effects of the conditions specified in b1) and b2) shall be combined.

The more severe of the two sets of conditions specified in a) and b) shall be used to specify the axle.

5.6.2 Effects due to masses in motion

The effects due to masses in motion shall be calculated from the following formulae:

$$P_1 = 0.55 m_1 g$$

$$P_2 = 0.55 m_1 g$$

5.7 Calculation of the resultant moment

In every section, the maximum stresses shall be calculated from the resultant moment *MR* (see Note 2), which is given by the formula:

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

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

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

$$MY = \sum M'_y$$

$$MZ = \sum M'_z$$

NOTE 1 The values of M'_x, M'_y and M'_z may be replaced by M''_x, M''_y and M''_z, respectively, if the moments due to traction are greater than the moments due to braking.

NOTE 2 At a point on the outer surface of a solid cylinder (also in the case of a hollow cylinder) of diameter d, the components MX, MY and MZ generate:

- *a direct stress for MX and MZ;*
- *a shear stress for MY.*

The value of the direct stress, σ_n, is calculated from the following formula (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, σ_t, is calculated from the following formula (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 from the following formulae:

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

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

Since the direct stress has a much higher absolute value (10 to 20 times greater) than the shear stress, the diameter of the largest Mohr's circle, σ, 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 resultant moment, MR, is given by the formula:

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

Table 3 **Values of components M'_x, M'_z and M'_y for different methods of braking**

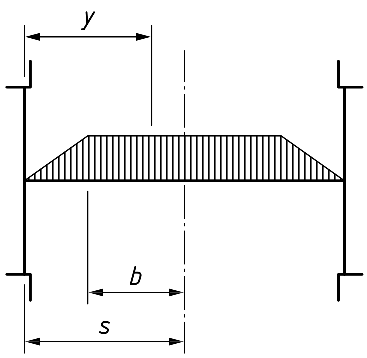
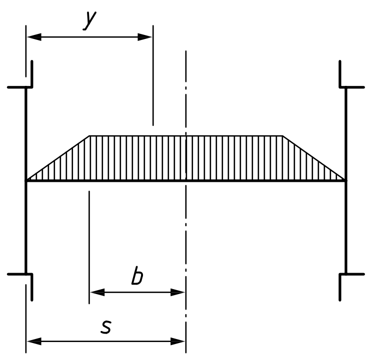
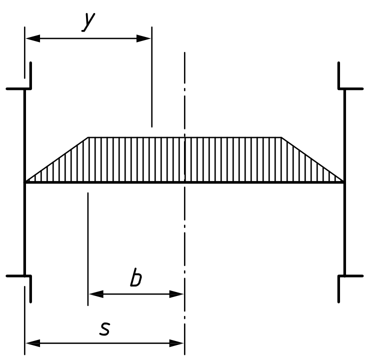
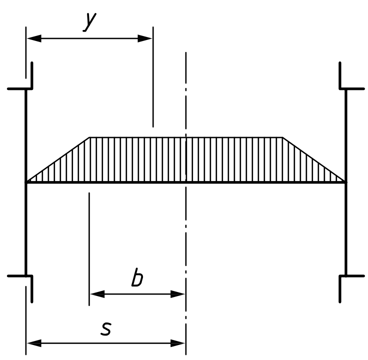
<table>
<tr><th rowspan="3">Components
M'_x, M'_z, M'_y</th><th colspan="4">Method of braking used</th></tr>
<tr><th colspan="2">Friction brake blocks on both sides of each wheel</th><th colspan="2">Friction brake block on one side only of each wheel</th></tr>
<tr><th>Between running surface and loading plane</th><th>Between loading planes</th><th>Between running surface and loading plane</th><th>Between loading planes</th></tr>
<tr><td rowspan="2">M'_x</td><td>$M'_x = 0.3F_f\Gamma y$
A), B)</td><td>$M'_x = 0.3F_f\Gamma(s-b)$
A), B)</td><td>$M'_x = F_f\Gamma y$
B)</td><td>$M'_x = F_f\Gamma(s-b)$
B)</td></tr>
<tr><td colspan="2">
</td><td colspan="2">
</td></tr>
<tr><td rowspan="2">M'_z</td><td>$M'_z = F_f(0.3+\Gamma)y$
A)</td><td>$M'_z = F_f(0.3+\Gamma)(s-b)$
A)</td><td>$M'_z = F_f(1+\Gamma)y$
B)</td><td>$M'_z = F_f(1+\Gamma)(s-b)$
B)</td></tr>
<tr><td colspan="2">
</td><td colspan="2">
</td></tr>
<tr><td>M'_y</td><td>$M'_y = 0.3P'R$
C), D)</td><td>$M'_y = 0.3P'R$
C), D)</td><td>$M'_y = 0.3P'R$
C), D)</td><td>$M'_y = 0.3P'R$
C), D)</td></tr>
</table>

Table 3 **Values of components M'_x, M'_z and M'_y for different methods of braking** *(continued)*

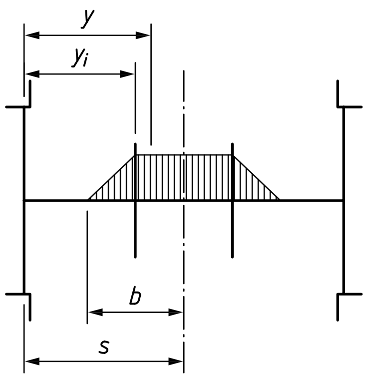
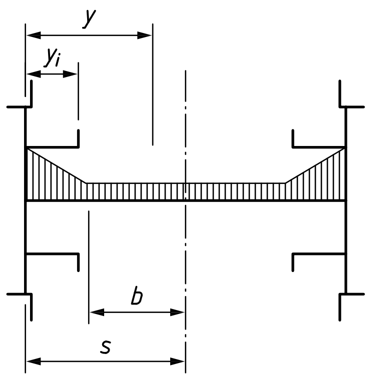
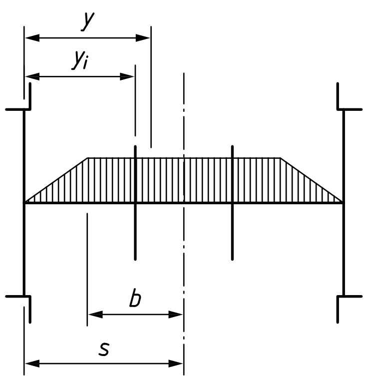
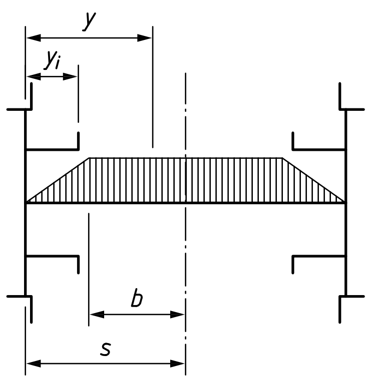
<table>
<tr><th rowspan="3">Components
M'_x, M'_z, M'_y</th><th colspan="5">Method of braking used</th></tr>
<tr><td colspan="3">Two brake discs mounted on the axle</td><td colspan="2">Two brake discs attached inboard to the wheel hub [F)]</td></tr>
<tr><td>Between running surface and loading plane</td><td>Between loading plane and brake disc</td><td>Between brake discs</td><td>Between running surface and loading plane</td><td>Between loading planes</td></tr>
<tr><td rowspan="2">M'_x</td><td>$M'_x = 0$</td><td>$M'_x = F_f \Gamma (b - s + y)$
B)</td><td>$M'_x = F_f \Gamma (b - s + y_i)$
B)</td><td>$M'_x = F_f \Gamma (y_i - y)$
B)</td><td>$M'_x = F_f \Gamma (b - s + y_i)$
B)</td></tr>
<tr><td colspan="3"></td><td colspan="2"></td></tr>
<tr><td rowspan="3">M'_z</td><td>Between running surface and loading plane</td><td colspan="2">Between loading planes</td><td>Between running surface and loading plane</td><td>Between loading planes</td></tr>
<tr><td>$M'_z = F_f \Gamma \frac{R_b}{R} y$
B)</td><td colspan="2">$M'_z = F_f \Gamma \frac{R_b}{R} (s - b)$
B)</td><td>$M'_z = F_f \Gamma \frac{R_b}{R} y$
B)</td><td>$M'_z = F_f \Gamma (s - b) \frac{R_b}{R}$
B)</td></tr>
<tr><td colspan="3"></td><td colspan="2"></td></tr>
<tr><td>M'_y</td><td>$M'_y = 0.3P'R$
D), E)</td><td colspan="2">$M'_y = 0.3P'R$
D), E)</td><td>$M'_y = 0.3P'R$
D), E)</td><td>$M'_y = 0.3P'R$
D), E)</td></tr>
</table>

Table 3 **Values of components M'_x, M'_z and M'_y for different methods of braking** *(continued)*

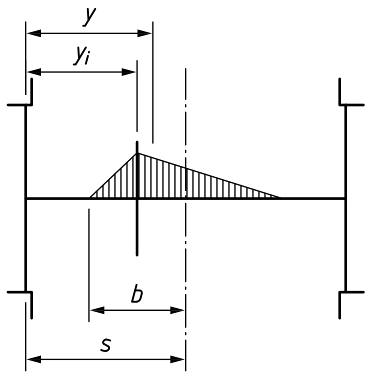
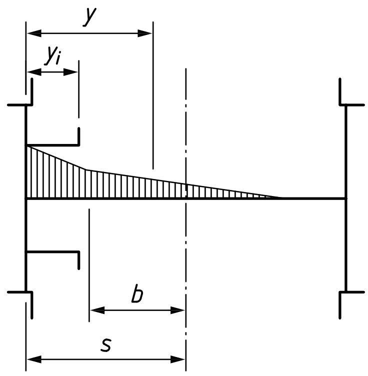
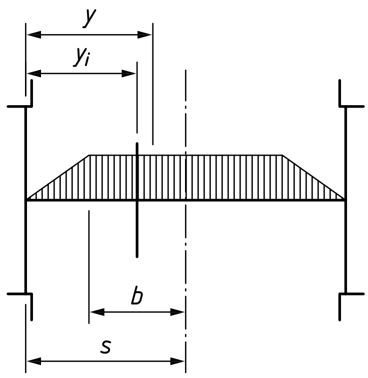
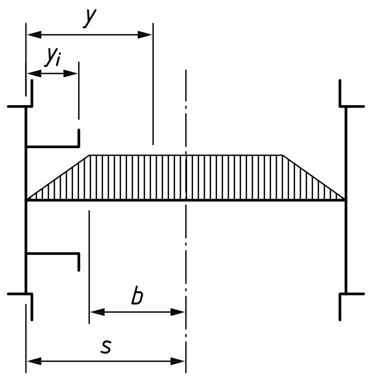
Components M'_x, M'_z, M'_y	**Method of braking used**			
	One brake disc mounted on the axle		One brake disc attached inboard to the wheel hub F)	
	Between first loading plane and brake disc	Between brake disc and second loading plane	Between running surface and first loading plane	Between loading planes
M'_x	$M'_x = F_f \Gamma \dfrac{(b+s-y_i)(b-s+y)}{2b}$ B)	$M'_x = F_f \Gamma \dfrac{(b-s+y_i)(b+s-y)}{2b}$ B)	$M'_x = F_f \Gamma (y_i - y)$ B)	$M'_x = F_f \Gamma \dfrac{(b-s+y_i)(b+s-y)}{2b}$ B)
				
M'_z	Between running surface and loading planes	Between loading planes	Between running surface and loading planes	Between loading planes
	$M'_z = \frac{1}{2} F_f \Gamma \frac{R_b}{R} y$	$M'_z = \frac{1}{2} F_f \Gamma \frac{R_b}{R} (s-b)$	$M'_z = \frac{1}{2} F_f \Gamma \frac{R_b}{R} y$	$M'_z = \frac{1}{2} F_f \Gamma \frac{R_b}{R} (s-b)$
				
M'_y	Between running surface and loading planes	Between loading planes	Between running surface and loading planes	Between loading planes
	$M'_y = 0.3P'R$ D), E)	$M'_y = 0.3P'R$ D), E)	$M'_y = 0.3P'R$ D), E)	$M'_y = 0.3P'R$ D), E)

Table 3 **Values of components M'_x, M'_z and M'_y for different methods of braking** *(continued)*

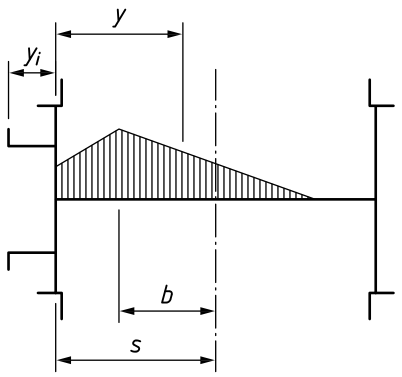
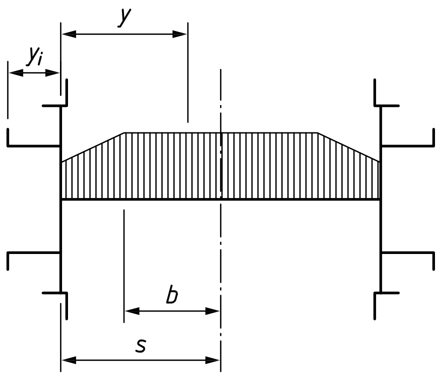
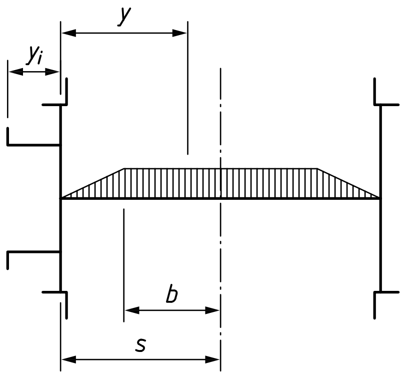
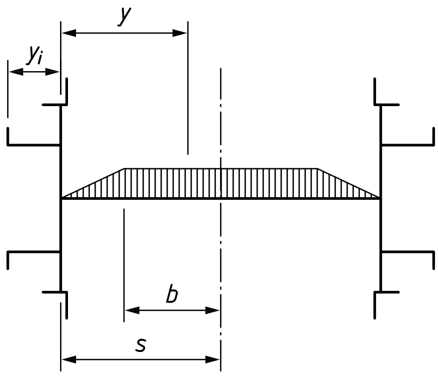
Components	**Method of braking used**			
M'_x, M'_z, M'_y	One brake disc attached outboard to the wheel hub F)		Two brake discs attached outboard to the wheel hub F)	
	Between running surface and first loading plane	Between loading planes	Between running surface and first loading plane	Between loading planes
M'_x	$M'_x = F_f \Gamma (y_i + y)$ B)	$M'_x = F_f \Gamma \left[(y_i - y) - \frac{(s+b+y_i)}{2b} \times (y - s + b) \right]$ B)	$M'_x = F_f \Gamma (y_i + y)$ B)	$M'_x = F_f \Gamma (y_i + s - b)$ B)
				
M'_z	Between running surface and loading planes	Between loading planes	Between running surface and loading planes	Between loading planes
	$M'_z = \frac{1}{2} F_f \Gamma \frac{R_b}{R} y$	$M'_z = \frac{1}{2} F_f \Gamma \frac{R_b}{R} (s - b)$	$M'_z = \frac{1}{2} F_f \Gamma \frac{R_b}{R} y$	$M'_z = \frac{1}{2} F_f \Gamma \frac{R_b}{R} (s - b)$
				
M'_y	Between running surface and loading planes	Between loading planes	Between running surface and loading planes	Between loading planes
	$M'_y = 0.3P'R$ D), E)	$M'_y = 0.3P'R$ D), E)	$M'_y = 0.3P'R$ D), E)	$M'_y = 0.3P'R$ D), E)

Table 3 **Values of components M'_x, M'_z and M'_y for different methods of braking** *(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:

Γ = 0.1 for cast iron blocks;

Γ = 0.17 for all blocks with a low-friction coefficient excluding cast iron;

Γ = 0.25 for all blocks with a high-friction coefficient excluding cast iron.

for brake pads:

Γ = 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 0.3*P'*. It includes the torsional moment as specified in **5.4**.

D) *P'* is the portion of *P* braked with the method of braking considered.

E) By convention, the torsional moment between running surfaces is given a value of 0.3*P'R*. It includes the torsional moment due to braking and the torsional moment as specified in **5.5**.

F) When the brake disc is mounted on the wheel web, then $y_i = 0$

NOTE Some example calculations are given in Annex B to Annex H.

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

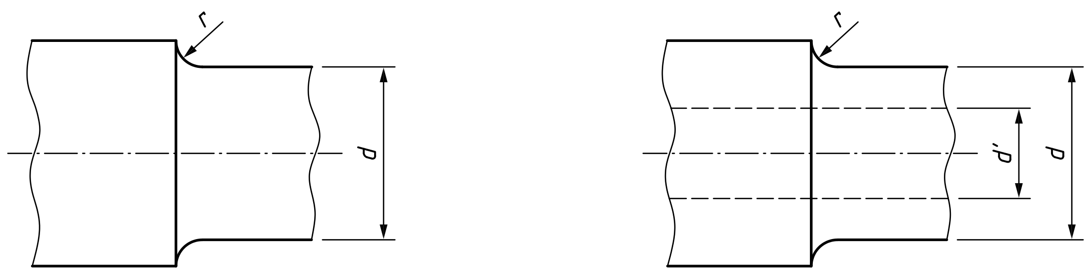
6.1 Stresses in the various sections of the axle

6.1.1 On any section of the axle of diameter *d*, the stress to be taken into account shall be as given by the following equations.

NOTE This British Standard only covers parallel wheel seats.

- For a solid axle (see Figure 6a): $\sigma = \dfrac{K \times 32 \times MR}{\pi d^3}$
- For a hollow axle (see Figure 6b):
  - on the outer surface: $\sigma = \dfrac{K \times 32 \times MR \times d}{\pi (d^4 - d'^4)}$
  - in the bore: $\sigma = \dfrac{K \times 32 \times MR \times d'}{\pi (d^4 - d'^4)}$

Figure 6 **Diameters of solid and hollow axles**



Key

d Diameter of axle

r Transition fillet radius

a) Solid axle

Key

d External diameter of axle

d' Bore diameter of axle

r Transition fillet radius

b) Hollow axle

6.1.2 In the cylindrical part, located on the surface of a solid or hollow axle, or in the bore of a hollow axle, the stress concentration factor *K* shall be taken to be equal to 1. However, account shall be taken of the fact that each change in section produces a stress increment, the maximum value of which occurs:

- at the bottom of a transition between two adjacent cylindrical parts with different diameters; and
- at the groove bottom.

6.1.3 The values of the stress concentration factor *K* used to calculate this increment shall be as shown in the nomograms in Figure 7 (transition between two adjacent cylindrical parts) and in Figure 8 (groove bottom).

NOTE 1 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 using a stress concentration factor from Figure 7.

NOTE 2 The values of K shown in Figure 7 and Figure 8 have been obtained from two ratios:

r/d and 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.

Figure 7 **Stress concentration factor *K* as a function of *D/d* and *r/d***

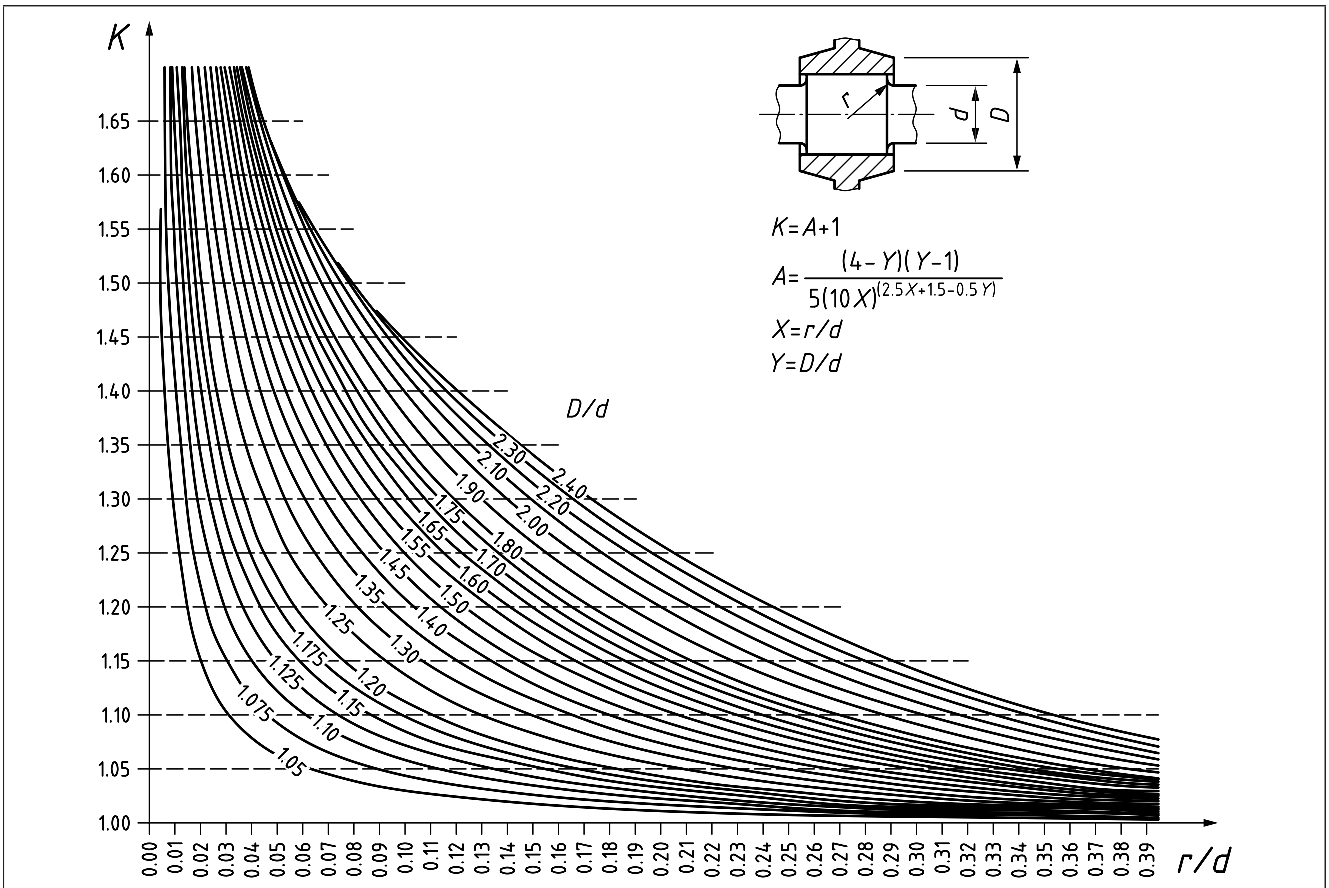
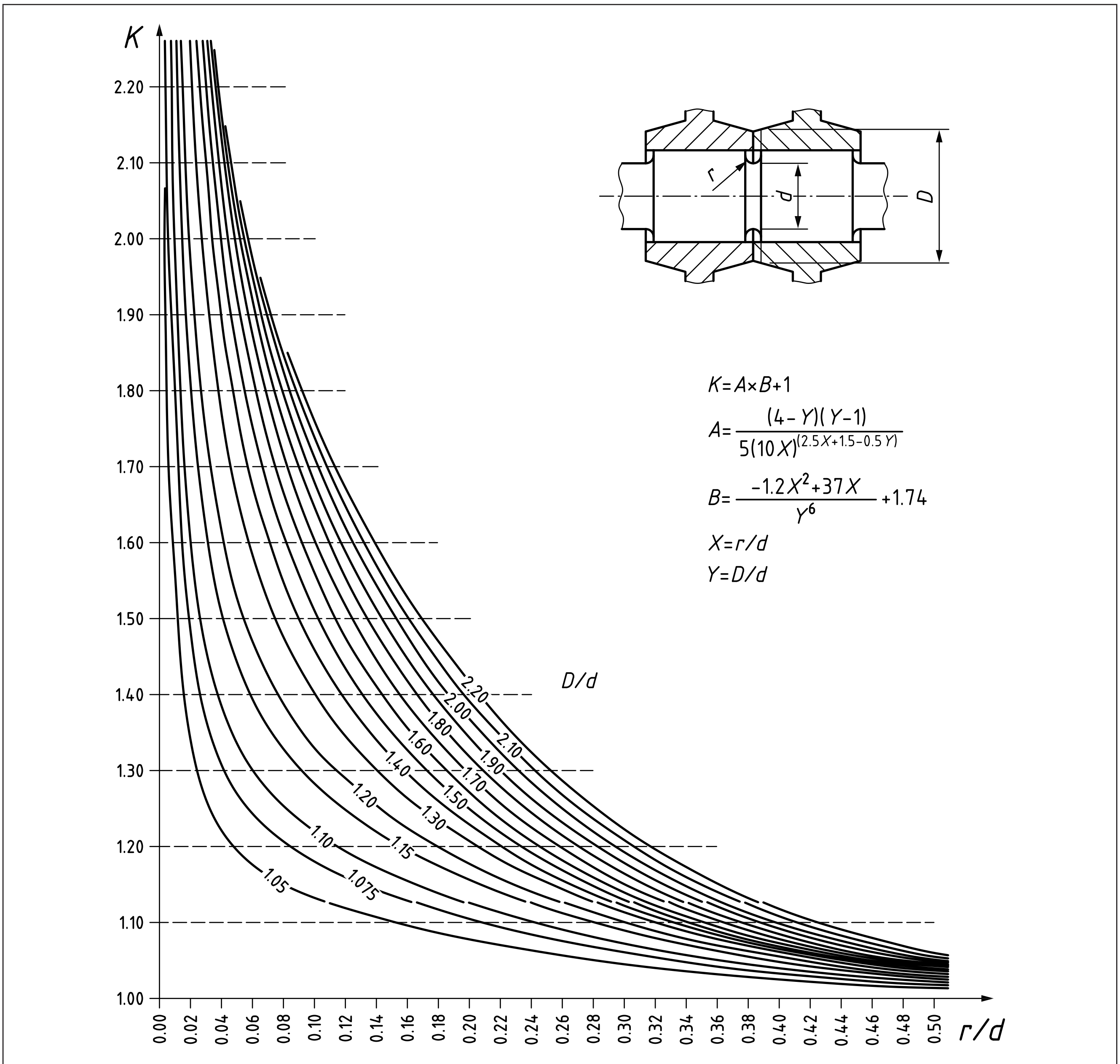
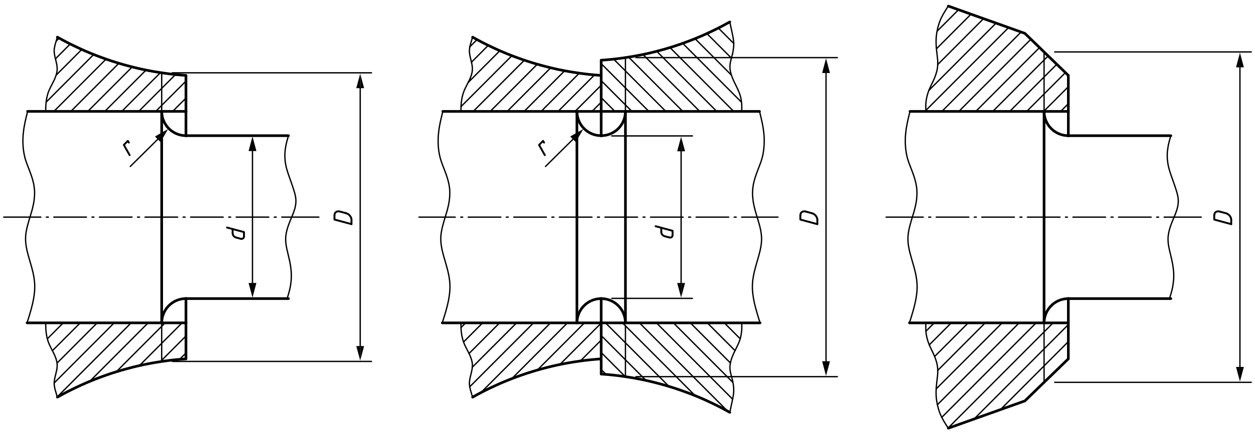


Figure 8 **Stress concentration factor *K* as a function of *D/d* and *r/d* (groove bottom)**



6.1.4 When a wheel, a brake disc, a pinion or a bearing is press-fitted (cold or hot) onto a seat, *D* shall be assumed to be equal to the diameter of the hub or the bearing ring (see Figures 9a, 9b and 9c). For a collar or deflector or cross-bar, *D* shall be assumed to be equal to the diameter of the bearing seat, since the interference fit of these parts is very small.

Figure 9 **Wheel, brake disc, pinion and bearing ring press-fitted onto a seat**



a) Wheel, brake disc or pinion
b) Abutting hubs (wheel/brake disc/pinion/bearing)
c) Bearing ring

Key

r Transition fillet or groove radius

d Diameter of the cylindrical part in which the stress concentration is calculated

D Diameter of the other cylindrical part

6.1.5 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 shall be made initially to existing sizes of associated components (e.g. bearings).

The maximum stresses in the axle shall 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 shall then be verified as specified in Clause **7**, the calculated stresses being compared to the maximum permissible stresses.

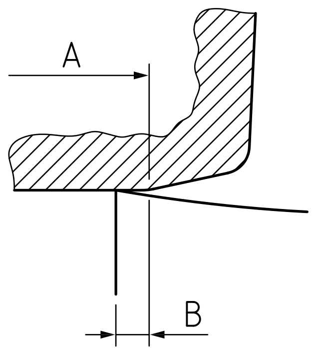
6.3 Determination of the diameter of the various seats from the diameter of the axle body or from the journals

6.3.1 Wheel seat and bearing seat interface requirements

For the wheel and bearing assembly, the overhang of the components shall be as shown in Figure 10 and Figure 11.

Figure 10 **Journal showing bearing overhang**

Dimensions in millimetres



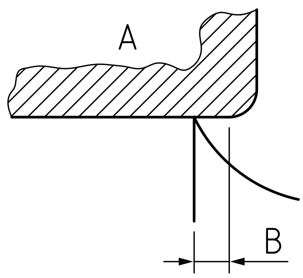
Key

A Cylindrical part of the bearing ring bore

B Overhang: >0 to ≤3

Figure 11 **Wheel seat showing wheel hub overhang**

Dimensions in millimetres



Key

A Wheel hub

B Overhang >0 to ≤5

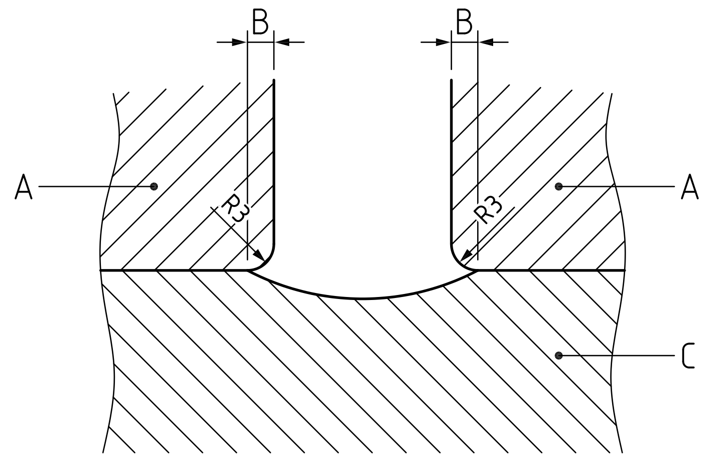
6.3.2 Transition between bearing seat and wheel seat

Whenever possible, the transition between the bearing seat and wheel seat shall have only a single radius as shown in Figure 12.

The highest possible value of this radius shall be selected so as to minimize the stress concentration on this area.

Figure 12 **Transition between bearing seat and wheel seat**

Dimensions in millimetres



Key

A Hub or bearing

B Overhang for bearings: >0 to ≤3

Overhang for hub: >0 to ≤5

C Axle

NOTE Where the wheel hub is used to hold the bearing in position, the maximum overhangs given may be exceeded.

6.3.3 Wheel seat in the absence of an adjacent seat

6.3.3.1 The allowable fatigue stress limits given in Table 4 and Table 5 shall only be applied in cases where the ratio of the wheel seat diameter to the axle body diameter is at least 1.12 at the wear limit. If design requirements would result in a ratio of less than 1.12 then the fatigue limit on the fitted seat shall be determined on at least three axles of representative geometry.

NOTE 1 It is recommended that this ratio is at least 1.15 for an axle in the new condition.

NOTE 2 The transition between these two areas should be provided in such a way that the stress concentration remains at the lowest possible level.

6.3.3.2 The lengths of the wheel seat and of the cylindrical part of the wheel hub bore shall be 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. In addition, at the maintenance limits, the design shall ensure that the stress limits in the seats and stress relief grooves are within the permitted design limits.

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

NOTE 2 The overlap criterion applies for the overlap and chamfered hubs on gearwheels and brake discs on their respective seats.

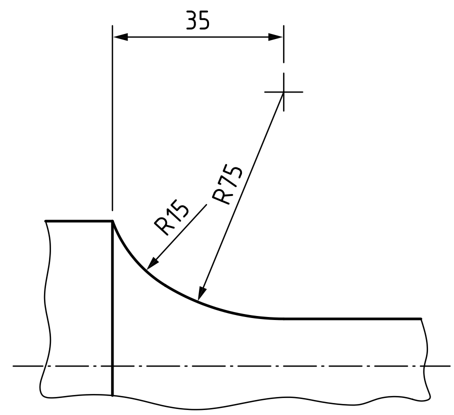
6.3.3.3 In order to have a low value of *K* at the transition between axle body and bearing, brake disc or gear seats, the value of the radius on the body side shall be at least 75 mm.

NOTE 1 Recommendations given in 4.3.2 of ORE Report No. 11 [1].

NOTE 2 An example of this transition is given in Figure 13.

Figure 13 **Transition between axle body and bearing, brake disc or gear seat**

Dimensions in millimetres



6.3.4 Case of two adjacent seats

Two seats shall be regarded as being adjacent if the transition from one seat to the other is by means of a single radius or a combination of radii and the fitted components are in contact (e.g. see Figure 9b).

All adjacent seats (wheel, gearwheel, labyrinth ring, axle drive bearing, shrink collar, distance ring, brake disc etc.) shall be considered.

A small groove (minimum depth very slightly greater than the seat wear range and minimum radius 16 mm) shall be provided to separate the two seats.

NOTE Its main role is to prevent notches that could be produced by the bore ends of the fitted components.

6.3.5 Case of two non-adjacent seats

Two 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 for these seats shall be as follows:

- calculation of the diameter of each seat;
- provision of overlapping hubs;
- determination of transitions;
- provision of a cylindrical part between two transitions.

6.3.6 Effect of interference fits on adjacent seats

For assembly of components prior to fitting wheels on the axle, the following shall be taken into account:

- allowance for shrinkage at axle seats due to other nearby interference fits (e.g. gearwheels and other bearings);
- checking the level of interference between bearings and their shafts or housings is within the specified tolerances.

7 Maximum permissible stresses

7.1 Steel grades EA1N, EA1T and EA4T

The maximum permissible stresses for EA1N and EA1T steels shall be as specified in Table 4 and for EA4T steel shall be as specified in Table 5.

Table 4 **Maximum permissible stresses for axles in steel grades EA1N and EA1T**

Intended use of the axle	**Zone 1** A) N/mm^2	**Zone 2** B) N/mm^2	**Zone 3** C) N/mm^2	**Zone 4** D) N/mm^2
Powered and non-powered, solid and hollow axles	110 E)	65	65	70 F)

A) Zone 1: axle body, plain bearing seats, transition fillets, bottom of grooves.

B) Zone 2: all seats except journals and plain bearing seats, i.e. wheel seats, brake disc seats, gearwheel seats and gearbox bearing seats where shallow grooves exist between the seats.

C) Zone 3: journal (beneath the rolling bearing).

D) Zone 4: bore.

E) For the main body of the axle where it is protected from impact damage and corrosion, (e.g. inside gearboxes and similar such protected areas) a stress of 133 N/mm^2 may be used.

F) This value is higher than the BS EN 13103 and BS EN 13104 permissible stress, but is valid for the bore surface finish as defined in BS EN 13261.

Table 5 **Maximum permissible stresses for axles in steel grade EA4T**

Intended use of the axle	**Zone 1** A) N/mm^2	**Zone 2** B) N/mm^2	**Zone 3** C) N/mm^2	**Zone 4** D) N/mm^2
Powered and non-powered, solid and hollow axles	120 E)	65	65	70 F)

A) Zone 1: axle body, plain bearing seats, transition fillets, bottom of grooves.

B) Zone 2: all seats except journals and plain bearing seats, i.e. wheel seats, brake disc seats, gearwheel seats and gearbox bearing seats where shallow grooves exist between the seats.

C) Zone 3: journal (beneath the rolling bearing).

D) Zone 4: bore.

E) For the main body of the axle where it is protected from impact damage and corrosion, (e.g. inside gearboxes and similar such protected areas) a stress of 145 N/mm^2 may be used.

F) This value is higher than the BS EN 13103 and BS EN 13104 permissible stress, but is valid for the bore surface finish as defined in BS EN 13261.

7.2 Other materials

If other materials are proposed for use, the fatigue characteristics and permissible stress limits for the zones defined in Table 4 and Table 5 shall be determined from material tests for the required configurations.

Annex A (informative) Sample axle calculation sheet template – Load case 1

Type

Drawing of axle No.

Drawing of wheel No.

Allocation

Material

Mass of wheelset (kg)

Axle

Wheels

Motor axle

Brake discs

Miscellaneous

Total (m_2)

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

Dimensions (mm)

$h_1 =$

$s =$

$R =$

Forces (N)

Braking force, $P_B =$

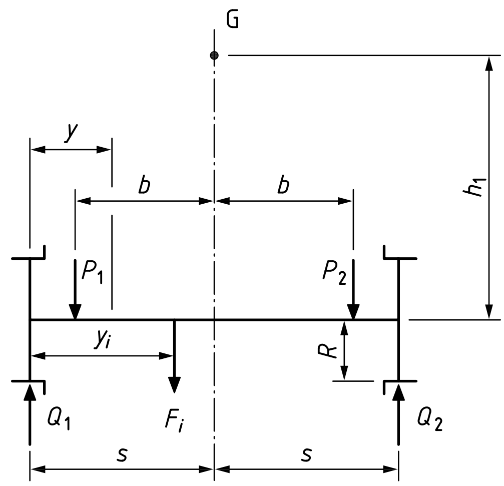
Reaction force (left), $P_L = P_1$

Reaction force (right) $P_R = P_2$

$P_1 = P_2 = 0.8m_1g$

$Q_1 = [P_1(s + b) + P_2(s – b) + \sum F_i\,(2s – y_i)]/2s$

$Q_2 = [P_1(s – b) + P_2(s + b) + \sum F_i y_i]/2s$



Key

G Centre of gravity of vehicle

Sample axle calculation sheet template – Load case 1 (*continued*)

y_i (mm)	F_i (N)	Part

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

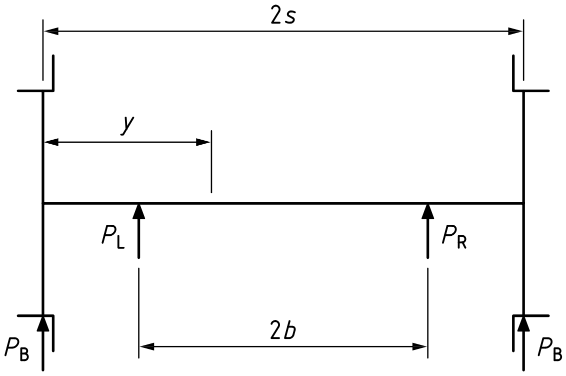
Section	y mm	d mm	d' mm	D mm	r mm	$\frac{r}{d}$	$\frac{D}{d}$	K	$\frac{32K \times 10^6}{\pi d^3}$ [1)]	M_x (N·mm) $\times 10^{-6}$	M'_x (N·mm) $\times 10^{-6}$	M'_z (N·mm) $\times 10^{-6}$	M'_y (N·mm) $\times 10^{-6}$	MR (N·mm) $\times 10^{-6}$	σ (N/mm²)	σ_{max} (N/mm²)

1) for hollow axles:

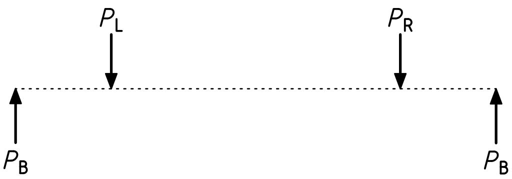
on the surface: $\frac{32K \times 10^6 d}{\pi(d^4 - d'^4)}$

in the bore: $\frac{32K \times 10^6 d'}{\pi(d^4 - d'^4)}$

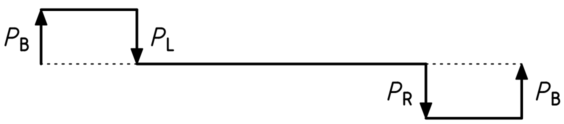
Annex B (informative) Sample axle calculation sheet – Friction blocks on each wheel



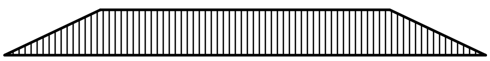
Applied forces balance



Shear force diagram



Bending moment diagram



Forces: $P_{\mathrm{L}} = P_{\mathrm{R}} = P_{\mathrm{B}}$ $= F_{\mathrm{f}}\Gamma$ (for blocks one side of wheel)
$= 0.3F_{\mathrm{f}}\Gamma$ (for blocks both sides of wheel)

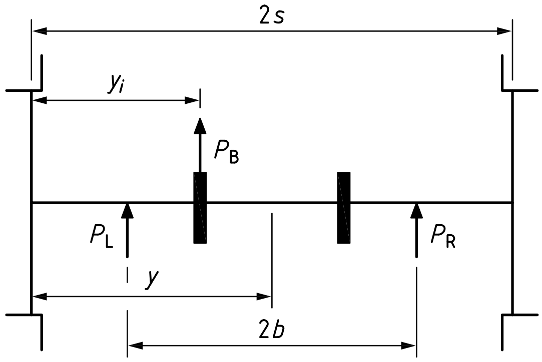
(1) Bending moment between running surface and first loading plane (journal)

$M'_{\mathrm{x}} = P_{\mathrm{B}}y$ $= F_{\mathrm{f}}\Gamma y$ (for blocks one side of wheel)
$= 0.3F_{\mathrm{f}}\Gamma y$ (for blocks both sides of wheel)

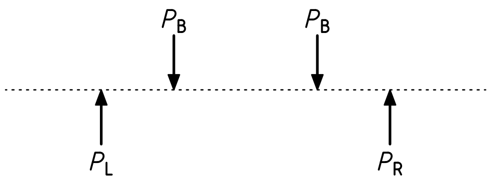
(2) Bending moment between loading planes

$M'_{\mathrm{x}} = P_{\mathrm{B}}y - P_{\mathrm{L}}(b - s + y)$ $= F_{\mathrm{f}}\Gamma[\mathrm{y} - (b - s + y)]$
$= F_{\mathrm{f}}\Gamma(s - b)$ (for blocks one side of wheel)
$= 0.3F_{\mathrm{f}}\Gamma(s - b)$ (for blocks both sides of wheel)

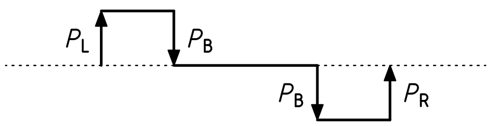
Annex C (informative) Sample axle calculation sheet – Two brake discs mounted on the axle



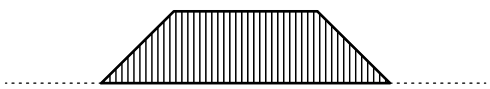
Applied forces balance



Shear force diagram



Bending moment diagram



Forces: $P_L = P_R = P_B = F_f \Gamma$

(1) Bending moment between running surface and first loading plane (journal) = 0

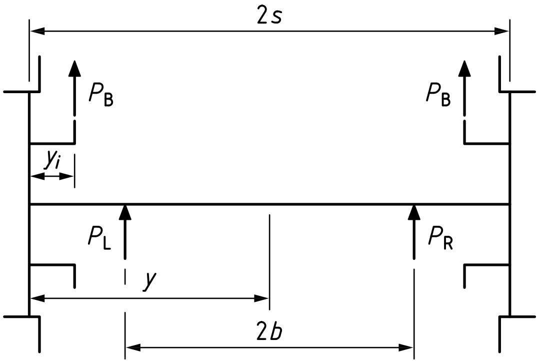
(2) Bending moment between first loading plane and brake disc

$$\begin{aligned} M'_x &= P_L(b - s + y) \\ &= F_f \Gamma (b - s + y) \end{aligned}$$

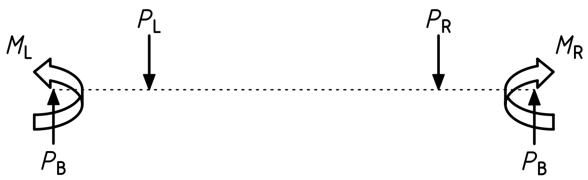
(3) Bending moment between brake discs

$$\begin{aligned} M'_x &= P_L(b - s + y) - P_B(y - y_i) \\ &= F_f \Gamma (b - s + y) - F_f \Gamma (y - y_i) \\ &= F_f \Gamma (b - s + y_i) \end{aligned}$$

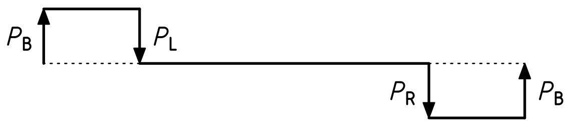
Annex D (informative) Sample axle calculation sheet – Two brake discs mounted inboard on the hub



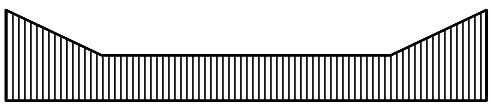
Applied forces balance



Shear force diagram



Bending moment diagram



M_L Moment on left-hand wheel

M_R Moment on right-hand wheel

Forces: $P_L = P_R = P_B = F_f\Gamma$

Moments: $M_L = M_R = P_B y_i = F_f \Gamma y_i$

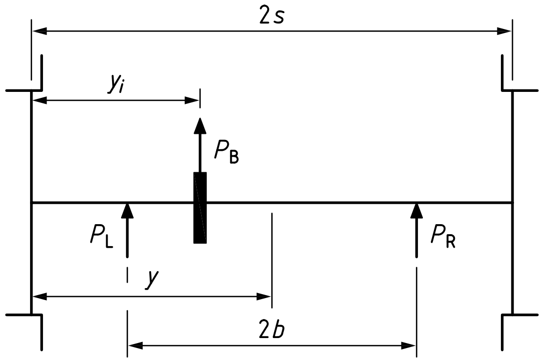
(1) Bending moment between running surface and first loading plane (journal)

$$M'_x = M_L - P_B y$$
$$= F_f \Gamma (y_i - y)$$

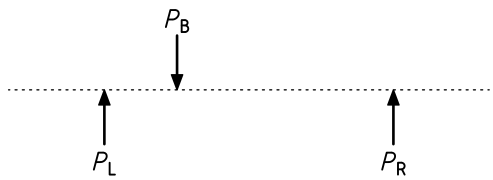
(2) Bending moment between loading planes

$$M'_x = M_L - P_B y + P_L\{y - (s - b)\}$$
$$= F_f \Gamma [y_i - y + \{y - (s - b)\}]$$
$$= F_f \Gamma (y_i - s + b)$$

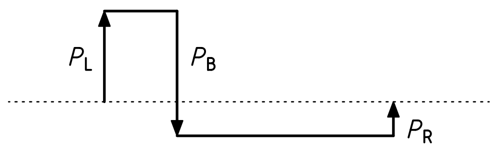
Annex E (informative) Sample axle calculation sheet – One brake disc mounted on the axle



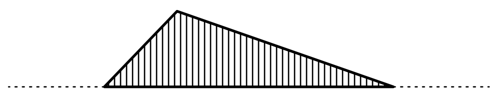
Applied forces balance



Shear force diagram



Bending moment diagram



Forces: $P_L = F_f\Gamma(b + s - y_i)/2b$

$P_R = F_f\Gamma(b - s + y_i)/2b$

(1) Bending moment between running surface and first loading plane (journal) = 0

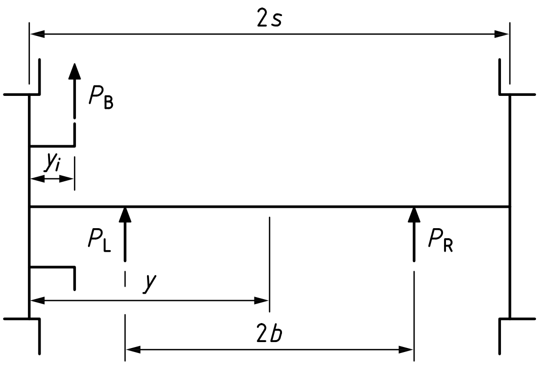
(2) Bending moment between first loading plane and brake disc

$$M'_x = P_L(b - s + y)$$
$$= F_f\Gamma(b + s - y_i)(b - s + y)/2b$$

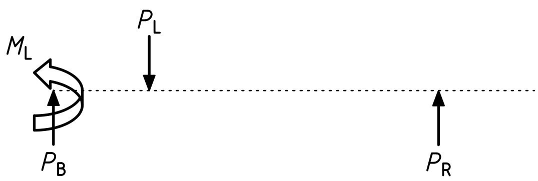
(3) Bending moment between brake disc and second loading plane

$$M'_x = P_L(b - s + y) - P_B(y - y_i)$$
$$= F_f\Gamma[(b + s - y_i)(b - s + y)/2b - (y - y_i)]$$
$$= F_f\Gamma[(b + s - y_i)(b - s + y) - (2by + 2by_i)]/2b$$
$$= F_f\Gamma[b^2 - s^2 + sy_i + sy - y_iy - by + by_i]/2b$$
$$= F_f\Gamma[(b + s)(b - s) + (b + s)y_i - (b - s + y_i)y]/2b$$
$$= F_f\Gamma[(b + s)(b - s + y_i) - (b - s + y_i)y]/2b$$
$$= F_f\Gamma[(b - s + y_i)(b + s - y)]/2b$$

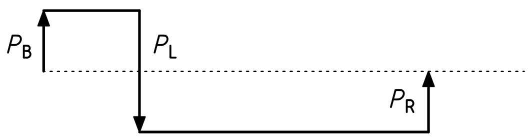
Annex F (informative) Sample axle calculation sheet – One brake disc mounted inboard on the hub



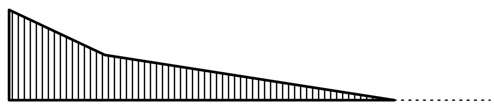
Applied forces balance



Shear force diagram



Bending moment diagram



M_L Moment on left-hand wheel

Forces: $P_B = P_L - P_R = F_f\Gamma$

$P_L 2b = P_B(s + b) - M_L = F_f\Gamma(s + b - y_i)$

$P_L = F_f\Gamma(s + b - y_i)/2b$

$P_R = F_f\Gamma(s - b - y_i)/2b$

Moments: $M_L = F_f\Gamma y_i$

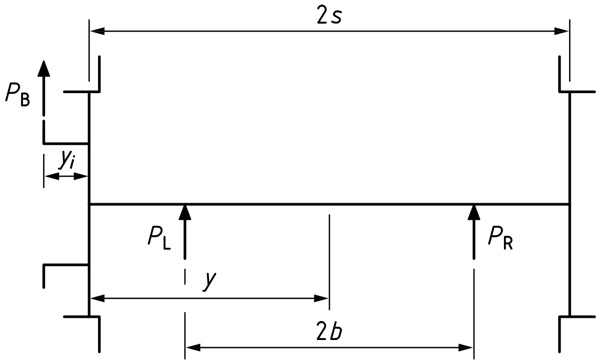
(1) Bending moment between running surface and first loading plane (journal)

$$M'_x = M_L - P_B y$$
$$= F_f\Gamma(y_i - y)$$

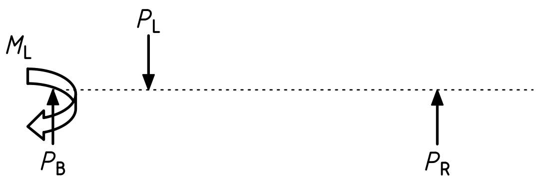
(2) Bending moment between loading planes

$$M'_x = M_L - P_B y + P_L\{y - (s - b)\}$$
$$= F_f\Gamma[(y_i - y) + (y - s + b)(s + b - y_i)/2b]$$
$$= F_f\Gamma[(2by_i - 2by) + (y - s + b)(s + b - y_i)]/2b$$
$$= F_f\Gamma[(b - s + y_i)(b + s - y)]/2b$$

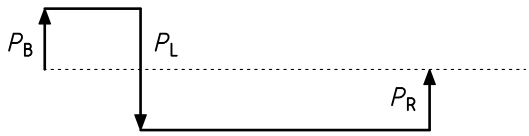
Annex G (informative) Sample axle calculation sheet – One brake disc mounted outboard on the hub



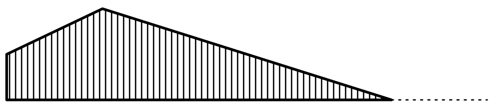
Applied forces balance



Shear force diagram



Bending moment diagram



M_L Moment on left-hand wheel

Forces: $P_B = P_L - P_R = F_f\Gamma$

$P_L 2b = M_L + P_B(s + b) = F_f\Gamma(y_i + s + b)$

$P_L = F_f\Gamma(y_i + s + b)/2b$

$P_R = F_f\Gamma(y_i + s - b)/2b$

Moments: $M_L = F_f\Gamma y_i$

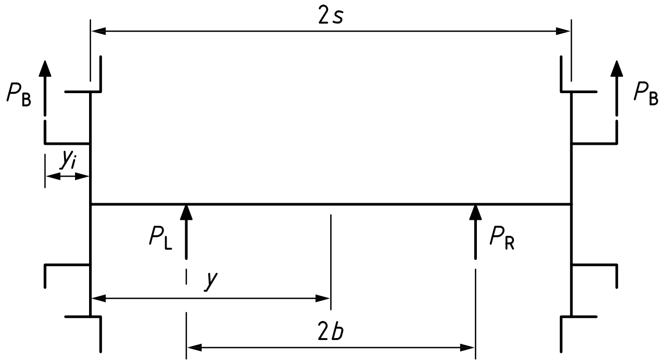
(1) Bending moment between running surface and first loading plane (journal)

$$M'_x = M_L + P_B y$$
$$= F_f\Gamma(y_i + y)$$

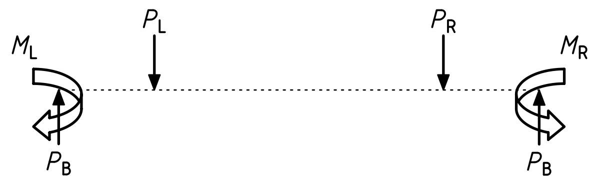
(2) Bending moment between loading planes

$$M'_x = M_L + P_B y - P_L\{y - (s - b)\}$$
$$= F_f\Gamma[(y_i + y) - (y - s + b)(y_i + s + b)/2b]$$

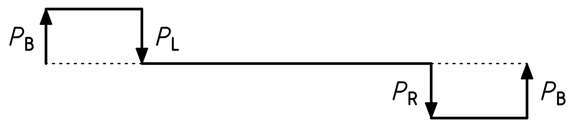
Annex H (informative) Sample axle calculation sheet – Two brake discs mounted outboard on the hub



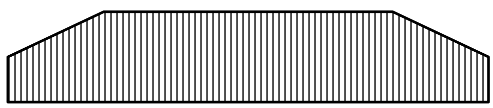
Applied forces balance



Shear force diagram



Bending moment diagram



M_L Moment on left-hand wheel
M_R Moment on right-hand wheel

Forces: $P_L = P_R = P_B = F_f \Gamma$

Moments: $M_L = M_R = P_B y_i = F_f \Gamma y_i$

(1) Bending moment between running surface and first loading plane (journal)

$$\begin{aligned} M'_x &= M_L + P_B y \\ &= F_f \Gamma (y_i + y) \end{aligned}$$

(2) Bending moment between loading planes

$$\begin{aligned} M'_x &= M_L + P_B y - P_L\{y - (s - b)\} \\ &= F_f \Gamma [y_i + y - \{y - (s + b)\}] \\ &= F_f \Gamma (y_i + s - b) \end{aligned}$$

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BS EN 13104, *Railway applications – Wheelsets and bogies – Powered axles – Design method*

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[2] UIC (Union International des Chemins de fer) Leaflet 515-3, *Railway rolling stock – Bogies – Running gear – Method of calculation for designing axles* [1)].

[3] EUROPEAN COMMUNITIES. 96/48/EC. Council Directive 96/48/EC of 23 July 1996 on the interoperability of the trans-European high-speed rail system. Luxembourg: Office for Official publications of the European Communities, 1996.

[4] EUROPEAN COMMUNITIES. 2001/16/EC. Directive 2001/16/EC of the European Parliament and of the Council of 19 March 2001 on the interoperability of the trans-European conventional rail system. Luxembourg: Office for Official publications of the European Communities, 2001.

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[7] TM/TC0001, *Design guide for the calculation of stresses in axles with inboard journals.* British Rail, 1990. Now out of print [2)].

[8] BASS 503, *Design guide for the calculation of stresses in axles driven by axle hung traction motors.* British Rail, 1996. Now out of print [2)].

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