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Railway applications — Calculation of braking performance (stopping, slowing and stationary braking) —

Part 1: General algorithms utilizing mean value calculation

Applications ferroviaires — Calcul des performances de freinage (freinage d'arrêt, de ralentissement et d'immobilisation) —

Partie 1: Algorithmes généraux utilisant le calcul par la valeur moyenne

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Foreword

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

The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation on the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT) see the following URL: www.iso.org/iso/foreword.html.

This document was prepared by Technical Committee ISO/TC 269, *Railway applications*.

A list of all parts in the ISO 20138 series can be found on the ISO website.

Any feedback or questions on this document should be directed to the user's national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

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Introduction

This document describes methodologies for calculation of braking performance, such as stopping distance, deceleration, power and energy for railway rolling stock. The calculations can be used at any stage of the assessment process (design, manufacture, testing, verification, investigation, etc.) of railway rolling stock.

The objective of this document is to enable the railway industry and operators to work with common calculation methods.

This document is published in two separate parts (ISO 20138-1 and ISO 20138-2), which will complement each other and can be used separately, depending on the requirements of the user.

The first part of the standard describes a common calculation method for railway applications applicable to all countries. It describes the general algorithms/formulae using mean value inputs to perform calculations of brake equipment and braking performance, in terms of stopping and slowing distances and safety for parking brake, for all types of trainsets and single vehicles. In addition, the algorithms provide a means of comparing the results of other braking performance calculation methods.

The second part of the standard details the step by step calculation methodology utilizing instantaneous values of brake force provided by each operational brake equipment type throughout the stopping/slowing time.

The two separate parts of the standard relate to each other but can be used separately, depending on the requirements of the user.

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Railway applications — Calculation of braking performance (stopping, slowing and stationary braking) —

Part 1: General algorithms utilizing mean value calculation

1 Scope

This document specifies methodologies for calculation of braking performance for railway rolling stock and is applicable to all countries.

This document describes the general algorithms/formulae using mean value inputs to perform calculations of brake equipment and braking performance in terms of stopping/slowing distances, stationary braking, power and energy for all types of rolling stock, either as single vehicles or train formations, with respect to the braking distance.

The calculations can be used at any stage of the assessment process (design, manufacture, testing, verification, investigation, etc.) of railway rolling stock. This document does not set out the specific acceptance criteria (pass/fail).

This document is not intended to be used as a design guide for selection of brake systems and does not specify performance requirements. This document does not provide a method to calculate the extension of stopping distances when the level of available adhesion is exceeded (wheel slide activity).

This document contains examples of the calculation of brake forces for different brake equipment types and calculation of stopping distance and stationary braking relevant to a single vehicle or a train.

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2 Normative references

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

ISO 20138-2[1)], *Railway applications — Calculation of braking performance (stopping, slowing and stationary braking) — Part 2: General algorithms utilizing step by step calculation*

3 Terms and definitions

For the purposes of this document, the following terms and definitions apply.

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

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

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

3.1
train
operational formation consisting of one or more units

1) Under preparation. Stage at the time of publication: ISO/DIS 20138-2:2018.

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3.2
trainset
fixed formation unit that can operate as a train

3.3
unit
element of the train formation which can be composed of one or several vehicle(s)

3.4
vehicle
individual element of a unit

EXAMPLE Locomotive, coach, wagon, driving coach.

3.5
retention force
force, which is greater or equal than the sum of external forces and downhill force due to gravity

3.6
brake force
retarding force
forces generated by a brake equipment type or external forces

Note 1 to entry: The dynamic mass (sum of static and rotating masses) is braked by brake forces.

Note 2 to entry: Retarding forces are stopping or decelerating a vehicle or unit.

Note 3 to entry: In some cases (e.g. magnetic track brake), the brake force is equal to the retarding force. In other cases, a differentiation in brake force and retarding force is needed, e.g. calculation of required adhesion.

4 Symbols

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

Table 1 — Symbols

Symbol	Definition	Unit
A_b	Contact area per brake block	m^2
A_c	Brake cylinder area	m^2
A_p	Contact area per brake pad	m^2
a	Defined level for the minimum output signal (typically 10 % or 5 %)	%
a_e	Equivalent deceleration	m/s^2
$a_{e,grad}$	Equivalent deceleration including the effect of gradient and inertia	m/s^2
$a_{e,grad\_simple}$	Equivalent deceleration neglecting inertia	m^2
$a_{e,z}$	Equivalent deceleration acting during speed range, z	m/s^2
α	Angle of slope	°
b	Defined level for the maximum output signal (typically 95 % or 90 %)	%
C_1	Characteristic coefficient of the train independent of speed	N
C_2	Characteristic coefficient of the train proportional to the speed	N/(m/s)
C_3	Characteristic coefficient of aerodynamic resistance due to pressure drag and skin friction drag	N/(m/s)^2
D	Wheel diameter	m
F	Force	N
F_{AMg}	Attraction force of one magnet	N

[a] bar or kPa is also allowed; 1 bar = 10^5 Pa.

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Table 1 *(continued)*

Symbol	Definition	Unit
$F_{AMg,st}$	Attraction force of one permanent magnet	N
F_{B}	Contribution of the friction brake	N
F_{Bd}	Blended retarding force	N
F_{BED}	Retarding force of electro-dynamic brake	N
$\bar{F}_{BED}$	Mean retarding force of electro-dynamic brake	N
$F_{BED,max}$	Maximum retarding force of electro-dynamic brake	N
F_{BFR}	Retarding force of fluid retarder	N
$\bar{F}_{BFR}$	Mean retarding force of fluid retarder	N
$F_{BFR,max}$	Maximum retarding force of fluid retarder	N
$\bar{F}_{B,n}$	Mean braking force of brake equipment type *n*	N
$F_{B,ind}$	Adhesion independent (not related to the wheel to rail contact) force, e.g. force of permanent magnetic track brake	N
$F_{B,ind,z}$	Adhesion independent (not related to the wheel to rail contact) retarding force per type of equipment	N
$F_{B,st}$	Total stationary brake force acting at the rail	N
$F_{B,ax,st}$	Stationary brake force acting on that wheelset	N
$F_{B,\tau,i}$	Adhesion dependent (related to the wheel to rail contact) retarding force generated by applied parking brake (i is an index used for sorting wheelsets)	N
$F_{B,\tau,req}$	Adhesion dependent (related to the wheel to rail contact) retarding force	N
$F_{B,\tau,req,rem}$	Remaining force of the mass to be held	N
F_{b}	Brake block force	N
$F_{b,st}$	Parking brake force acting on the tread of the wheel from a single parking brake unit	N
$F_{b,ax}$	Single brake block force	N
$F_{b,ax,st}$	Static single brake block force	N
$F_{b,tot}$	Total force acting on all disc faces or total brake block force	N
$F_{b,tot,st}$	Total static force acting on all disc faces or total static brake block force	N
$F_{Cr,H}$	Crank handle or hand wheel force	N
F_{c}	Internal cylinder force	N
F_{cl}	Clamping force	N
$F_{cl,n}$	Brake calliper clamping force	N
F_{D}	Downhill force due to gravity	N
$F_{d,ax}$	Proportion of downhill force to be resisted per wheelset with applied parking brakes	N
$\bar{F}_{ext}$	Mean external force	N
F_{ext}	External forces (e.g. wind force)	N
F_{G}	Output force of parking brake mechanism	N
F_{g}	Weight	N
$F_{g,ax}$	Static axle load	N
F_{H}	Retention force	N
$F_{Mg,st}$	Parking brake force of one permanent magnet	N
$F_{Mg,st,tot}$	Total parking brake force of all permanent magnets in a vehicle	N

[a] bar or kPa is also allowed; 1 bar = 10^5 Pa.

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Table 1 *(continued)*

Symbol	Definition	Unit
F_N	Static axle load perpendicular to the rail per wheelset with applied parking brake	N
$F_{N,ax}$	Static axle load perpendicular to the rail per wheelset with applied parking brake for a specific wheelset	N
$F_{N,i}$	Static axle load perpendicular to the rail per wheelset with applied parking brake (*i* is an index used for sorting wheelsets)	N
$F_{N,rem}$	Remaining static axle load perpendicular to the rail	N
F_{PB}	Total parking brake force acting at the rail	N
$F_{PB,ax}$	Parking brake force per wheelset acting at the rail	N
F_{Perp}	Perpendicular force	N
$F_{Perp,ax}$	Perpendicular force at wheelset	N
F_p	Piston force	N
F_{pad}	Force acting on single disc surface	N
$F_{pad,n}$	Force acting on single disc surface	N
F_{pull}	Force on application point bogie	N
$F_{pull,st}$	Static force on application point bogie	N
F_{Ra}	Retarding force by train resistance	N
$\bar{F}_{Ra}$	Mean retarding force by train resistance	N
$F_{Ra,st}$	Stationary train resistance force	N
F_r	Instantaneous retarding force acting at the rail generated by the brake equipment	N
$\bar{F}_r$	Mean retarding force acting at the rail generated by the brake equipment	N
F_{rECB}	Instantaneous retarding force of linear eddy current brake	N
$\bar{F}_{rECB}$	Mean retarding force of linear eddy current brake	N
$F_{rECB,max}$	Maximum retarding force of linear eddy current brake	N
F_{rMg}	Retarding force of one magnet	N
$F_{rMg,tot}$	Total retarding force of all magnets in a vehicle	N
$\bar{F}_{r,n}$	Mean retarding force of brake equipment type *n*	N
F_{SP}	Parking brake spring force	N
$F_{S,C}$	Restoring force of brake unit or spring applied force	N
$F_{S,R}$	Restoring force, e.g. slack adjuster	N
F_{st}	Stationary brake force of the train	N
$F_{st,ax}$	Transmittable stationary brake force acting on that wheelset	N
$F_{st,n}$	Stationary brake force acting on the wheelset of each immobilization/holding/parking brake, *n*	N
$F_{s,rig}$	Restoring force	N
F_t	Tangential force	N
F_{wind}	Wind force on the train	N
g	Standard acceleration due to gravity	m/s^2
η_{Cbl}	Cable efficiency	—
η_c	Internal efficiency of brake unit	—
η_G	Gear efficiency	—
η_R	Overall efficiency of brake rigging	—
[a] bar or kPa is also allowed; 1 bar = 10^5 Pa.		

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Table 1 *(continued)*

Symbol	Definition	Unit
$\eta_{R,st}$	Overall static efficiency of brake rigging	—
η_{rig}	Efficiency of brake rigging/calliper	—
$\eta_{rig,st}$	Static efficiency of calliper	—
I	Current	A
i	Gradient of the track (positive rising/negative falling)	—
i_{Cbl}	Cable mechanical ratio	—
i_c	Internal rigging ratio of brake unit	
i_G	Gear ratio	—
i_{max}	Maximum achievable gradient	—
$i_{max,roll}$	Maximum achievable gradient for rolling	—
$i_{max,slide}$	Maximum achievable gradient for sliding	—
i_{rig}	Rigging ratio	—
$i_{rig,ax,n}$	Lever ratio per brake beam	—
$i_{rig,C}$	Calliper lever ratio (parking brake)	—
$i_{rig,n}$	Calliper lever ratio	—
$i_{s,rig}$	Rigging ratio for restoring force	—
J	Inertia	kg·m^2
k_0, k_2	Coefficient (provided by the supplier)	—
k_1	Coefficient (provided by the supplier)	s/m
$k1_v$, $k2_v$	Factor describing an active or passive brake cylinder	—
l_a, l_b	Main brake lever length	m
$l_{a,n}$, $l_{b,n}$	Calliper lever length	m
l_e	Main brake lever length (parking brake)	m
l_c, l_d	Bogie lever length	m
M	Mass to be held of the vehicle/unit/train	kg
m_{dyn}	Dynamic mass	kg
m_{rot}	Equivalent rotating mass	kg
$m_{rot,ax}$	Equivalent rotating mass of the braked wheelset	
m_{st}	Static mass	kg
$m_{st,ax}$	Static mass per wheelset	kg
μ_{Mg}	Mean friction coefficient of magnet (pole shoe)	—
$\mu_{Mg,st}$	Static friction coefficient of permanent magnet (pole shoe)	—
μ_m	Mean friction coefficient of brake block/brake pad	—
μ_{st}	Static friction coefficient of brake block/brake pad	—
N	Number of brake equipment types	—
$n_{PB,ax}$	Number of wheelsets with applied parking brake	—
n_{Beam}	Number of brake beams	—
n_{BW}	Number of braked wheelsets	—
n_{disc}	Number of brake discs	—
n_{face}	Number of disc faces	—
n_{Mg}	Number of magnets in a vehicle	—
n_{SP}	Number of spring brake units	—
n_1, n_2	Value of power in speed range above v_{cha} normally obtained from supplier	—

[a] bar or kPa is also allowed; 1 bar = 10^5 Pa.

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Table 1 *(continued)*

Symbol	Definition	Unit
$P_{max,n}$	Maximum power of brake equipment type *n*	W
p	Pressure	N/m^2
p_{ab}	Specific pressure per brake block	N/m^2
p_{ap}	Specific pressure per brake pad	N/m^2
p_c	Brake cylinder pressure	Pa [a]
r_m	Mean swept radius of the brake pad on the disc face	m
S_H	Retention safety	—
S_R	Safety against rolling	—
S_{st}	Safety ratio for stationary brake	—
$S_{\tau,slide}$	Safety against sliding	—
s	Stopping/slowing distance	m
$s_{B,n}$	Distance travelled while the brake equipment type *n* is applied	m
s_{grad}	Stopping/slowing distance on a gradient	m
s_0	Equivalent free running distance	m
t	Time	s
t_a	Initial delay (dead time)	s
$t_{a,n}$	Initial delay (dead time) for a specific brake equipment type *n*	s
t_{ab}	Build-up time	s
$t_{ab,n}$	Build-up time for a specific brake equipment type *n*	s
t_b	Overall response time ($t_a + t_{ab}$)	s
t_e	Equivalent response time	s
$t_{e,n}$	Equivalent response time for a specific brake equipment type *n*	s
$\overline{\tau_{ax}}$	Value of the mean adhesion required between wheel/rail for the braked wheelsets	—
$\overline{\tau}_{ax,i}$	Temporary value of the mean adhesion required between wheel/rail for the braked wheelset used during iteration step *i*	—
τ_a	Available adhesion	—
$\tau_{req,st,ax}$	Coefficient of adhesion required to resist the downhill and external forces by each braked wheelset	—
$\tau_{D,req,ax}$	Coefficient of adhesion required to resist the downhill force by each braked wheelset	—
τ_{max}	Maximum permitted or available static wheel/rail adhesion	—
$\tau_{req,ax}$	Coefficient of adhesion required by each braked wheelset	—
$\tau_{req,max,ax}$	Maximum required adhesion by each braked wheelset	—
v	Speed	m/s
v_{cha}	Characteristic speed (corresponding to maximum retarding force)	m/s
v_{fin}	Final speed	m/s
v_{max}	Maximum speed	m/s
v_0	Initial speed	m/s
$v_1 \ldots v_4$	Particular speeds	m/s
$v_{0,Mg}$	Activating speed of magnetic track brake	m/s
$v_{1,Mg}$	Deactivating speed of magnetic track brake	m/s
[a] bar or kPa is also allowed; 1 bar = 10^5 Pa.		

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Table 1 *(continued)*

Symbol	Definition	Unit
W_B	Energy dissipated by the brake systems	J
$W_{B,n}$	Energy dissipated by brake equipment type n	J
W_{Ra}	Energy dissipated by the train resistance	J
W_{tot}	Total energy	J
Y	Percentage of output signal	—
z	Speed range step number	—
Z	Number of speed ranges	—
[a] bar or kPa is also allowed; 1 bar = 10^5 Pa.		

5 Stopping and slowing distances calculation

5.1 General

A summary of the methodology to establish the braking forces acting on the train is presented in Figure A.1.

The algorithms in this document use mean values and are applicable when the response time is less than 20 % of the time with the maximum braking force. For response times with a greater percentage (e.g. braking from low initial speeds) or where instantaneous values and algorithms are used or the finite time steps are preferred, ISO 20138-2 shall be used.

The mean value calculation is not intended to be used for an extreme value estimation or variation, e.g. minimum/maximum friction coefficient of friction couple. The input values for the calculation are used without tolerances.

The retarding forces expressed in this document are those acting parallel to the rail.

The brake system design parameters necessary to conduct the calculation shall be defined at the level of the wheelset, bogie, vehicle, unit or train. For the purpose of this document, the general term "vehicle" is used.

Calculations shall be performed for each brake equipment type (e.g. disc brakes, tread brakes, electro-dynamic brakes). All of the various types of brake equipment applied to the wheelset, bogie, vehicle, unit or train shall be identified and accounted for in the calculation.

When the brake equipment fitted to the train is used under different circumstances, e.g. load condition, speed range, brake demand, etc., each condition or state of the brake shall be considered together with the resultant effect on braking force.

This clause identifies how to calculate the braking force generated by each brake equipment type related to the retardation force at the rail. In general, calculations of stopping and slowing distances are based on the assumption of straight and level track.

Annex C provides examples for brake calculations of different vehicles and units.

The following subclauses consider the braking force generated by common brake equipment types. If other brake equipment types are used, e.g. new or novel types, then alternative methods of braking force calculation should be adopted.

Figure 1 gives a general overview of brake equipment types.

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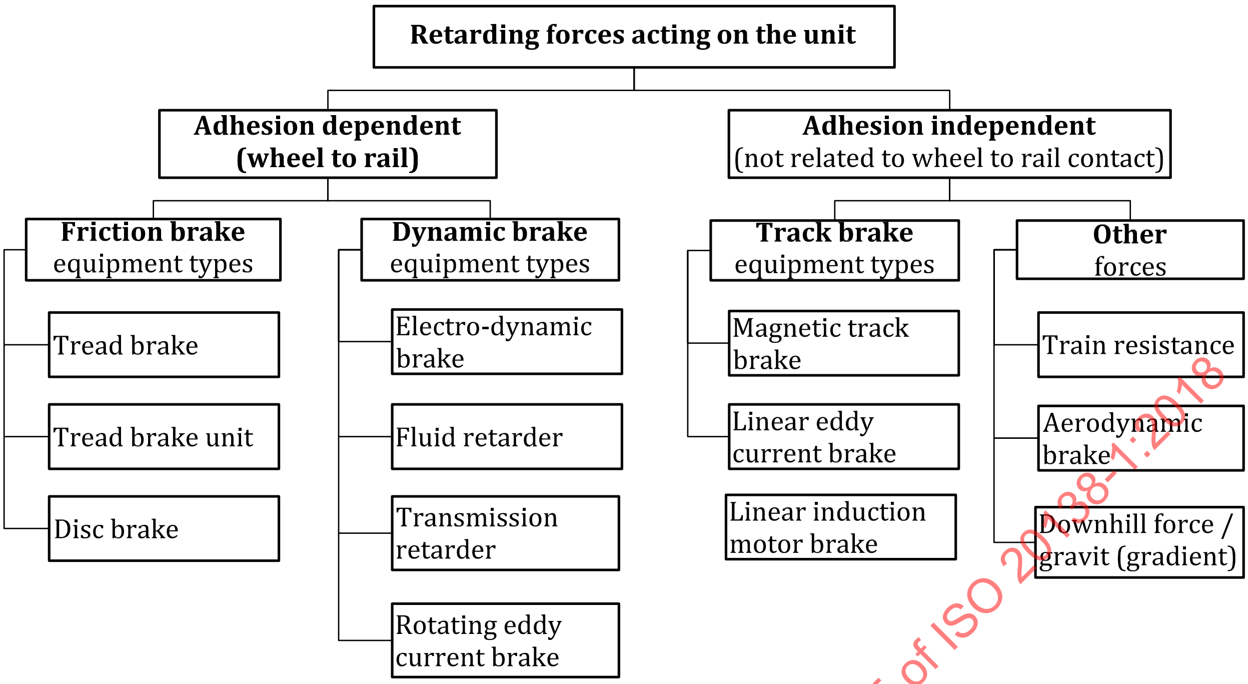

Figure 1 — General overview of retarding forces acting on the unit

5.2 Vehicle characteristics

5.2.1 Static mass, m_{st}

The static mass, m_{st}, of the vehicle and/or the static mass of a wheelset, $m_{st,ax}$, is assessed in stationary condition and shall be used to establish the braking force required or the adhesion requirements respectively for each applicable operating condition.

When there are different static masses per wheelset, $m_{st,ax}$, due to different vehicle arrangements, the braking force shall be calculated for each wheelset.

5.2.2 Equivalent rotating mass, m_{rot}

The equivalent rotating mass, m_{rot} is the linear conversion of the moment of inertia due to

— the rotation of the wheelsets, and

— the rotating parts coupled to the wheelsets during braking.

The equivalent rotating mass shall be determined using a theoretical approach or established as a result of tests. The wheel size applicable to the rotating mass shall be identified.

A value of equivalent rotating mass can be identified as a percentage of the static mass.

When there are different rotating masses, e.g. a mix of trailer and driven wheelsets, the rotating mass shall be determined for each type of wheelset.

For those wheelsets, if an inertia value, J, due to the rotating masses is known, the equivalent rotating mass using inertia is calculated in accordance with Formula (1):

$$m_{rot} = \frac{4 \cdot J}{D^2} \qquad (1)$$

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where

m_{rot} is the equivalent rotating mass, expressed in kg;

J is the inertia, expressed in kg·m²;

D is the wheel diameter, expressed in m.

NOTE The wheel diameter used for calculation of rotating masses is normally the maximum wheel diameter.

5.2.3 Dynamic mass, m_{dyn}

For the purpose of the calculation being conducted, the dynamic mass is the sum of the static mass and the equivalent rotating mass for the entity being considered, e.g. wheelset, bogie, vehicle, etc., in accordance with Formula (2):

$$m_{dyn} = m_{st} + m_{rot} \tag{2}$$

where

m_{dyn} is the dynamic mass, expressed in kg;

m_{st} is the static mass, expressed in kg;

m_{rot} is the equivalent rotating mass, expressed in kg.

5.2.4 Wheel diameter

The wheel diameter, D, is the diameter at the rolling contact point between the wheel and the rail.

When the vehicle is equipped with different sizes of wheels (by design not due to wear), each size of wheel shall be determined.

NOTE 1 The wheel diameter used for calculation of stopping and slowing distances is normally the maximum wheel diameter.

NOTE 2 The wheel diameter used to determine the required adhesion, $\tau_{req,ax}$, is normally the minimum wheel diameter.

5.3 Adhesion wheel/rail dependent brake equipment type characteristics

5.3.1 Basic brake cylinder

This subclause describes the calculation of piston force as an output force of the brake cylinder.

The first step is the calculation of the internal cylinder force, F_C [see Formula (3)].

Efficiency, ratios, friction resistances and moduli of resilience, etc. are not considered in the formula of calculation of the cylinder force.

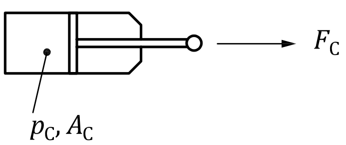

Figure 2 — Basic principle of a pressure applied cylinder

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$$F_c = p_c \cdot A_c \quad (3)$$

where

p_C is the brake cylinder pressure, expressed in Pa;

A_C is the brake cylinder area, expressed in m²;

F_C is the internal cylinder force, expressed in N.

Two types of brake cylinder exist: active version and passive version. The active version uses the brake cylinder pressure to generate the piston force. The passive version uses the spring force to generate the piston force (see Figure 3).

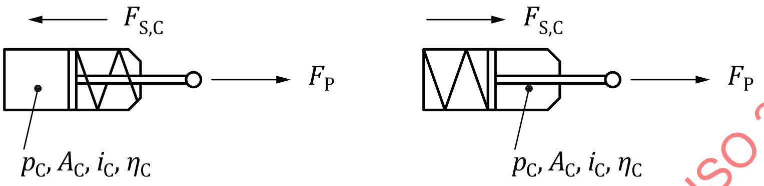

Figure 3 — Basic principle of an active (left) and passive (right) cylinder

The second step is the calculation of piston force, F_p, based on the internal brake cylinder force considering the mechanical efficiency and internal rigging ratio of brake unit. Formula (4) enables the calculation of the piston force for active and passive cylinders:

$$F_P = k1_V \cdot |p_c \cdot A_c \cdot \eta_c \cdot i_c| + k2_V \cdot |F_{S,C}| \quad (4)$$

where

F_p is the piston force, expressed in N;

p_C is the brake cylinder pressure, expressed in Pa;

A_C is the brake cylinder area, expressed in m²;

η_c is the internal efficiency of brake unit;

i_C is the internal rigging ratio of brake unit;

$F_{S,C}$ is the restoring force of brake unit or spring applied force, expressed in N.

Table 2 — Factors $k1_V$ *and* $k2_V$ for active and passive brake cylinder

	Active cylinder (pressure-applied brake)	Passive cylinder (spring-applied brake)
$k1_V$	1	–1
$k2_V$	–1	1

For simplification, a general brake cylinder, shown in Figure 4, is used throughout this document instead of active and passive cylinder.

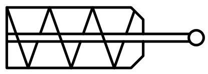

Figure 4 — Illustration of general brake cylinder

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5.3.2 Tread brake

The brake cylinder piston force is transferred to the brake blocks, taking into account the specific mechanical rigging ratios, efficiencies of brake rigging and counter forces. The brake block force for an arrangement as shown in Figure 5 is calculated as set out in Formula (6).

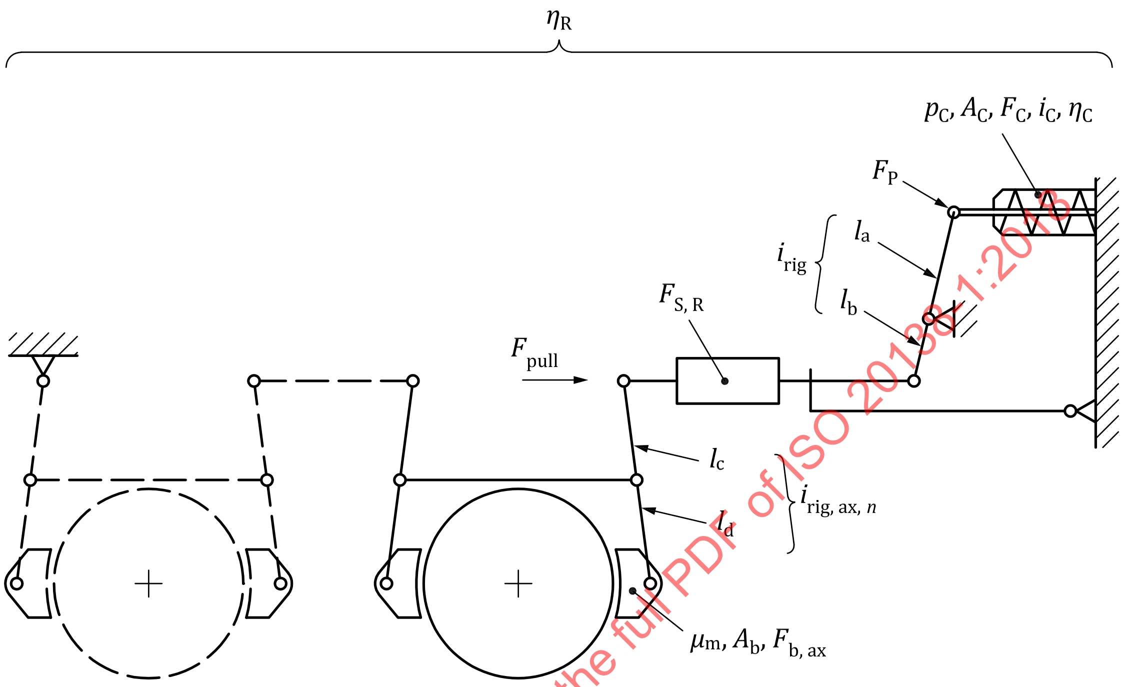

Figure 5 — Typical clasp tread brake arrangement

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The theoretical maximum of the force on application point bogie, F_{pull}, is calculated as set out in Formula (5):

$$F_{pull} = F_P \cdot i_{rig} - F_{S,R} \quad (5)$$

Assuming the bogie is fitted with brake beams, the theoretical maximum of the force acting on a single block, $F_{b,ax}$, is calculated as set out in Formula (6):

$$F_{b,ax} = \frac{1}{2} \cdot F_{pull} \cdot i_{rig,ax,n} \quad (6)$$

where

Table 3 — Formulae for single-sided and double-sided brake block arrangement

	Brake block arrangement	
	Single-sided	**Double-sided (clasp brake)**
$i_{rig,ax,n}$	$i_{rig,ax,n} = \frac{l_{c,i} + l_{d,i}}{l_{d,i}}$ (7) see Figure 6	$i_{rig,ax,n} = \frac{l_{c,i}}{l_{d,i}}$ (8) see Figure 5

NOTE 1 Due to the brake beam, there are two brake blocks per wheelset. Therefore, the multiplier ½ in Formula (6) is used to calculate the single brake block force, $F_{b,ax}$,. For Figure 5, four brake beams per bogie means eight brake blocks per bogie. For Figure 6, two brake beams per bogie means four brake blocks per bogie.

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The total brake block force is calculated as set out in Formula (9):

$$F_{b,tot} = \sum F_{b,ax} \cdot \eta_R \tag{9}$$

NOTE 2 The overall efficiency of brake rigging, η_R , covers all particular efficiencies of the components and the rigging.

The theoretical maximum of the specific pressure per brake block is calculated as set out in Formula (10):

$$p_{ab} = \frac{F_{b,ax}}{A_b} \tag{10}$$

where

F_{pull}	is the force on application point bogie, expressed in N;
$F_{b,ax}$	is the single brake block force, expressed in N;
$F_{b,tot}$	is the total brake block force, expressed in N;
F_p	is the piston force [see Formula (4)], expressed in N;
i_{rig}	is the rigging ratio;
$i_{rig,ax,n}$	is the lever ratio per brake beam;
l_a, l_b	is the main brake lever length, expressed in m;
l_c, l_d	is the bogie lever length, expressed in m;
$F_{S,R}$	is the restoring force, e.g. slack adjuster, expressed in N;
η_R	is the overall efficiency of brake rigging;
p_{ab}	is the specific pressure per brake block, expressed in N/m^2;
A_b	is the contact area per brake block, expressed in m^2.

The mean retarding force acting at the rail depends on the friction coefficient of the brake block and is independent of the wheel diameter as set out in Formula (11):

$$\bar{F}_r = F_{b,tot} \cdot \mu_m \tag{11}$$

where

$\bar{F}_r$	is the mean retarding force acting at the rail generated by the brake equipment, expressed in N;
$F_{b,tot}$	is the total brake block force, expressed in N;
μ_m	is the mean friction coefficient of brake block.

NOTE 3 The methodology to determine the mean friction coefficient of the brake blocks is outside the scope of this document.

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If the internal bogie ratios are equal, the simplifications shown in Formula (12) can be done:

$$i_{\text{rig,ax,1}} = i_{\text{rig,ax,2}} = i_{\text{rig,ax},n} \quad (12)$$

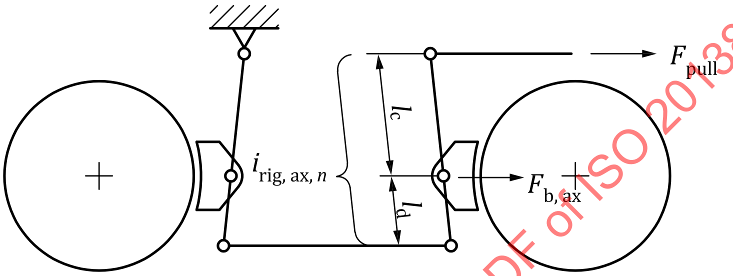
Formula (9) can be simplified with

— Formulae (6) and (7) for bogie with single-sided brake block arrangement (see Figure 6), or

— Formulae (6) and (8) for bogie with double-sided brake block arrangement (see Figure 5)

to Formula (13):

$$F_{\text{b,tot}} = \eta_{\text{R}} \cdot \sum \left(F_{\text{pull}} \cdot i_{\text{rig,ax},n} \cdot n_{\text{Beam}} \right) \quad (13)$$



EXAMPLE Three-piece bogie.

Figure 6 — Typical single-side brake block arrangement

When $l_c = l_d$, Formula (13) can be simplified to Formula (14) for a bogie with two wheelsets:

$$F_{\text{b,tot}} = 4 \cdot F_{\text{b,ax}} \cdot \eta_{\text{R}} \quad (14)$$

where

$F_{\text{b,tot}}$	is the total brake block force, expressed in N;
$F_{\text{b,ax}}$	is the single brake block force, expressed in N;
F_{pull}	is the force on application point bogie, expressed in N;
$i_{\text{rig,ax},n}$	is the lever ratio per brake beam;
n_{Beam}	is the number of brake beams;
η_{R}	is the overall efficiency of brake rigging.

5.3.3 Tread brake unit

The brake cylinder piston force is transferred to the brake blocks taking into account the specific mechanical rigging ratios, efficiencies of brake rigging and counter forces. The brake block force for an arrangement as shown in Figure 7 is calculated as set out in Formula (15).

The principle is the same for a spring applied design.

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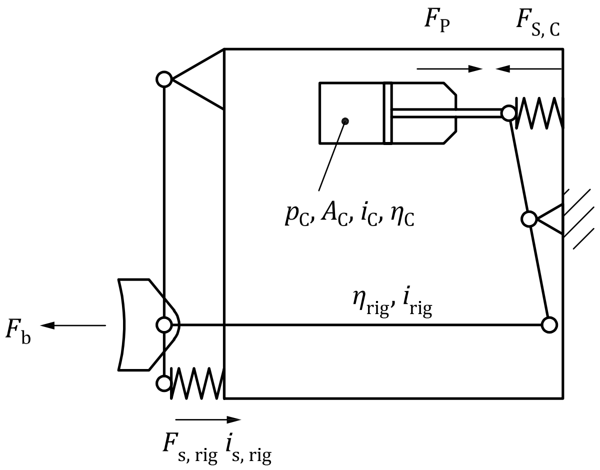

Figure 7 — Representative tread brake unit

$$F_b = F_P \cdot i_{rig} \cdot \eta_{rig} - F_{s,rig} \cdot i_{s,rig} \quad (15)$$

where

F_b is the brake block force, expressed in N;

F_P is the piston force [see Formula (4)], expressed in N;

i_{rig} is the rigging ratio;

η_{rig} is the efficiency of brake rigging;

$F_{s,rig}$ is the restoring force, expressed in N;

$i_{s,rig}$ is the rigging ratio for restoring force.

In general, $i_{s,rig}$ = 1, so Formula (15) can be simplified to Formula (16):

$$F_b = F_P \cdot i_{rig} \cdot \eta_{rig} - F_{s,rig} \quad (16)$$

The mean retarding force acting at the rail generated by the tread brake unit depends on the friction coefficient of the brake block and is independent of the wheel diameter as set out in Formula (17):

$$\bar{F}_r = F_b \cdot \mu_m \quad (17)$$

where

$\bar{F}_r$ is the mean retarding force acting at the rail generated by a tread brake unit, expressed in N;

F_b is the brake block force, expressed in N;

μ_m is the mean friction coefficient of brake block.

NOTE The methodology to determine the mean friction coefficient of the brake blocks is outside the scope of this document.

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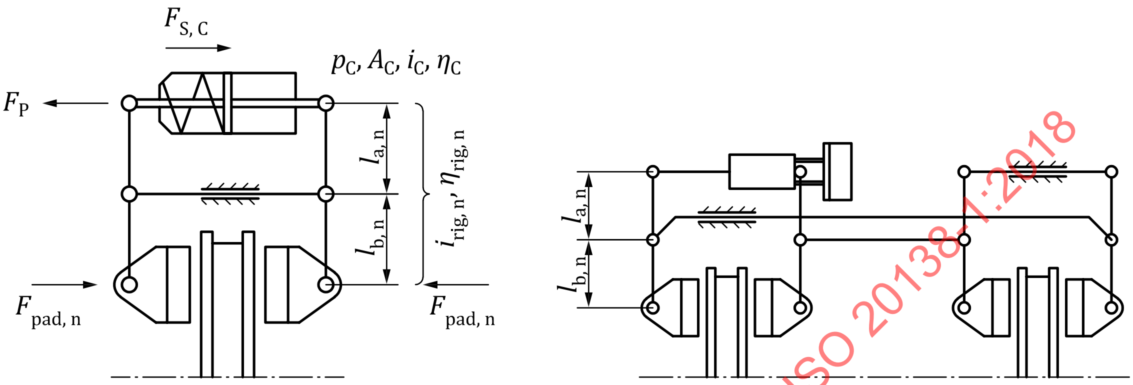
5.3.4 Disc brake

The brake cylinder piston force is transferred to the brake pads taking into account the specific mechanical calliper ratios, efficiencies and counter forces. The individual brake pad force (single face of the disc) for an arrangement as shown in Figure 8 is calculated as set out in Formula (18).

The principle is the same for a spring applied design (see 6.7.4).



a) Two disc faces

b) Four disc faces

Figure 8 — Typical arrangement of brake disc calliper

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The first step is the calculation of the piston force, F_P, considering the mechanical efficiency [see Formula (4)]. The force, $F_{S,C}$, considers the counter forces including, e.g. counter forces in the slack adjuster.

The second step is the calculation of the brake pad force, $F_{pad,n}$, acting on an individual disc face, as shown in Formula (18):

$$F_{pad,n} = F_P \cdot i_{rig,n} \cdot \eta_{rig} \tag{18}$$

With calliper lever ratio as shown in Formula (19):

$$i_{rig,n} = \frac{l_{a,n}}{l_{b,n}} \tag{19}$$

The specific brake pad contact pressure, p_{ap}, is calculated as set out in Formula (20):

$$p_{ap} = \frac{F_{pad}}{A_p} \tag{20}$$

The brake calliper clamping force, $F_{cl,n}$, acting on an individual disc with two disc faces is set out in Formula (21):

$$F_{cl,n} = 2 \cdot F_{pad,n} \tag{21}$$

where

F_{pad} is the force acting on single disc surface, expressed in N;

$F_{pad,n}$ is the force acting on single disc surface, expressed in N;

F_P is the piston force [see Formula (4)], expressed in N;

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$i_{rig,n}$ is the calliper lever ratio;

η_{rig} is the efficiency of calliper;

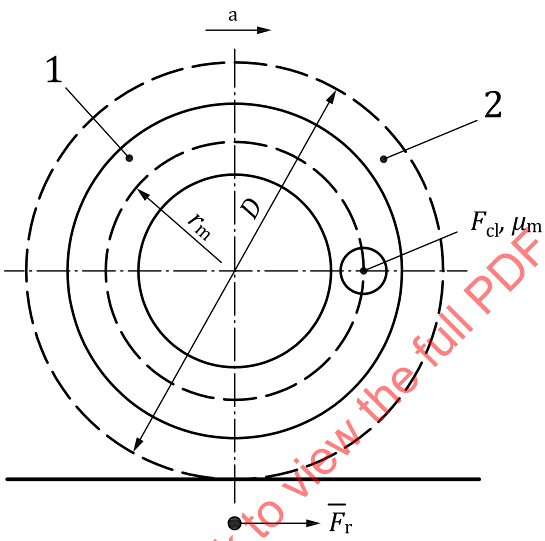
$l_{a,n}$, $l_{b,n}$ are the calliper lever lengths, expressed in m;

$F_{cl,n}$ is the brake calliper clamping force, expressed in N;

p_{ap} is the specific pressure per brake pad, expressed in N/m^2;

A_p is the contact area per brake pad, expressed in m^2.

The retardation force is acting on the rail contact point and is dependent on wheel diameter, the mean swept radius and the friction coefficient of the brake pad as shown in Figure 9.



Key

1 disc

2 wheel

a Driving direction.

Figure 9 — Diagram to show relationship between mean swept radius and wheel diameter

The mean retarding force acting at the rail generated by the brake equipment can be calculated as set out in Formula (22):

$$\bar{F}_r = F_{cl} \cdot n_{disc} \cdot \mu_m \cdot \frac{r_m}{D/2} \quad (22)$$

where

$\bar{F}_r$ is the mean retarding force acting at the rail generated by the brake equipment, expressed in N;

F_{cl} is the clamping force, expressed in N;

n_{disc} is the number of brake discs;

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μ_m is the mean friction coefficient of brake pad;

r_m is the mean swept radius of the brake pad on the disc face, expressed in m;

D is the wheel diameter, expressed in m.

NOTE 1 The methodology to determine the mean friction coefficient of the brake pads and the mean swept radius are outside the scope of this document.

NOTE 2 Mean swept radius of the brake pad on the disc face is also known as brake radius.

The tangential force, F_t, reacted by the bogie per disc can be calculated as set out in Formula (23):

$$F_t = F_{cl} \cdot \mu_m \tag{23}$$

where

F_t is the tangential force, expressed in N;

F_{cl} is the clamping force, expressed in N;

μ_m is the mean friction coefficient of brake pad.

5.3.5 Electro-dynamic brake

The braking force of an electro-dynamic brake, F_{BED}, is generally represented by a characteristic curve. Figure 10 shows an indicative braking force curve that could be achieved by an electro-dynamic brake.

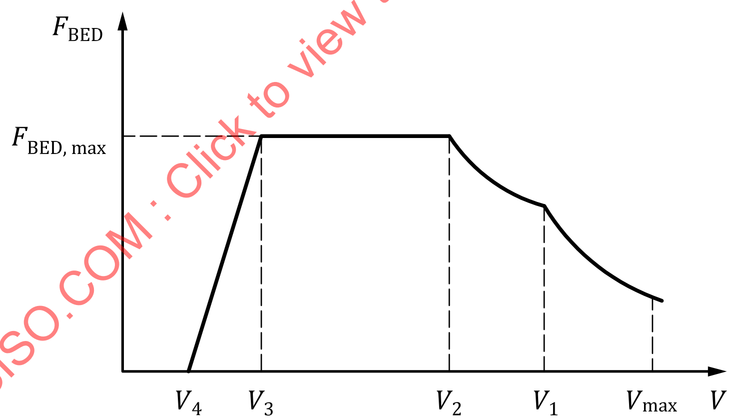

Figure 10 — Indicative characteristic braking force of an electro-dynamic brake

The section of the curve for speeds higher than v_1 (depending on $\frac{v_2 \cdot v_1}{v^2}$) is used with electro-dynamic braking and regenerative braking when the voltage has to be limited due to the receptivity of the electrical supply line.

The section of the curve from v_1 to v_2 indicates the range where available power is limiting the electro-dynamic force (hyperbolic section, depending on $\frac{v_2}{v}$).

The section of the curve from v_2 to v_3 is characterized by the constant braking force, $F_{BED,max}$, which is limited for reasons of available adhesion or longitudinal dynamics.

The section of the curve from v_3 to v_4 is the portion of the curve when the electro-dynamic force reduces due to the characteristic of the electric motor.

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An example of electro-dynamic braking force calculation is provided in D.3.

5.3.6 Fluid retarder

The retarding force of the fluid retarder, F_{BFR}, is generally represented by a characteristic curve. Figure 11 shows an indicative braking force curve that could be achieved by a fluid retarder (also known as hydraulic dynamic brake).

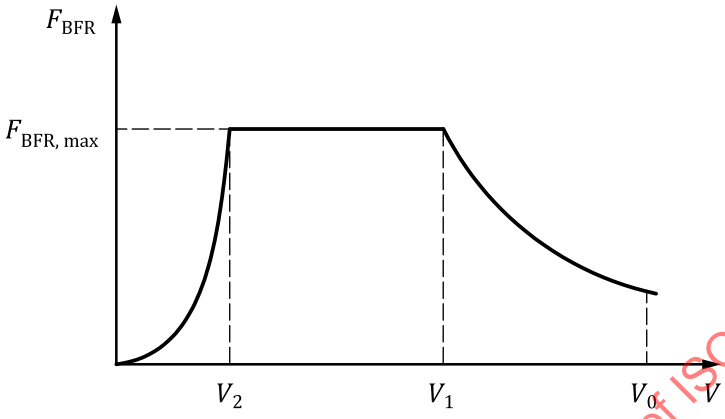

Figure 11 — Indicative characteristic retarding force of the fluid retarder

The section of the curve for speeds higher than v_1 is defined by the power limit of the cooling system of the fluid retarder.

The section of the curve from v_1 to v_2 is the maximum retarding force of the fluid retarder, $F_{BFR,max}$.

The section of the curve below v_2 is the portion of the curve when the braking force reduces due to the characteristic of the fluid retarder.

An example of retarding force of the fluid retarder, $F_{BFR,max}$, calculation is provided in D.4.

5.3.7 Transmission retarder

Reserved.

5.3.8 Rotating eddy current brake

Reserved.

5.4 Adhesion independent brake equipment type characteristics

5.4.1 Magnetic track brake

The retarding force generated by magnetic track brake, F_{rMg}, is not constant and depends on:

— the friction coefficient of the magnetic track brake segments in contact with the rail;

— the attraction force, F_{AMg}, which itself depends on the magnetic characteristics of the pole shoe and track materials.

The braking force of one magnet of a magnetic track brake can be calculated as set out in Formula (24):

$$F_{rMg} = F_{AMg} \cdot \mu_{Mg} \tag{24}$$

where

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F_{rMg} is the retarding force of one magnet, expressed in N;

F_{AMg} is the attraction force of one magnet, expressed in N;

μ_{Mg} is the mean friction coefficient of magnet (pole shoe).

Generally, the retarding force of one magnet of a magnetic track brake increases with decreasing speed. It is represented by a curve (see Figure 12).

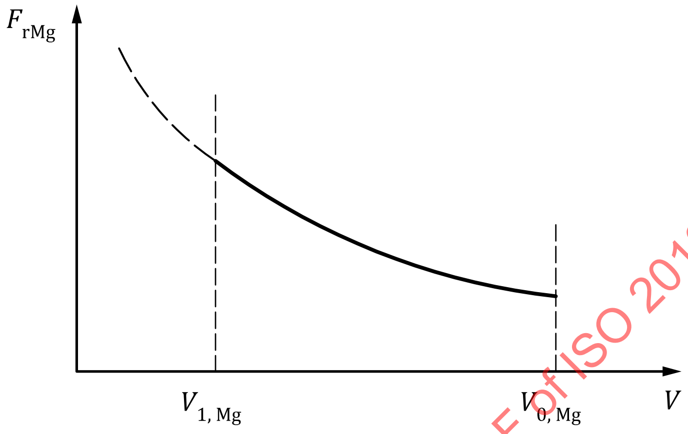

Figure 12 — Indicative retarding force of one magnet of a magnetic track brake vs. speed

The application of the magnetic track brake is often limited to a range of operating speeds between $v_{0,Mg}$ and $v_{1,Mg}$ ($v_{0,Mg}$: activating speed of magnetic track brake; $v_{1,Mg}$: deactivating speed of magnetic track brake). For speeds outside this range, the magnetic track brake is not applied.

Therefore, the mean value of the friction coefficient depends on the operating speeds of the magnetic track brake.

An example calculation for the retarding force of a magnetic track brake is provided in D.1.

NOTE 1 The selection of the operating speed range depends on the mechanical properties of the magnetic track brake.

The total retarding force of all magnets in a vehicle, $F_{rMg,tot}$, can be calculated as set out in Formula (25):

$$F_{rMg,tot} = n_{Mg} \cdot F_{rMg} \quad (25)$$

where

$F_{rMg,tot}$ is the total retarding force of all magnets in a vehicle, expressed in N;

n_{Mg} is the number of magnets in a vehicle;

F_{rMg} is the retarding force of one magnet, expressed in N.

NOTE 2 A vehicle is always equipped with magnets of the same type arranged in couples.

5.4.2 Linear eddy current brake

The retarding force of the linear eddy current brake, F_{rECB}, depends on:

— the gap between the magnets of the eddy current brake and the head of the rail;

— intensity of the magnetic field (magnet coil and excitation current);

— operating speed range including fade in/out ramps.

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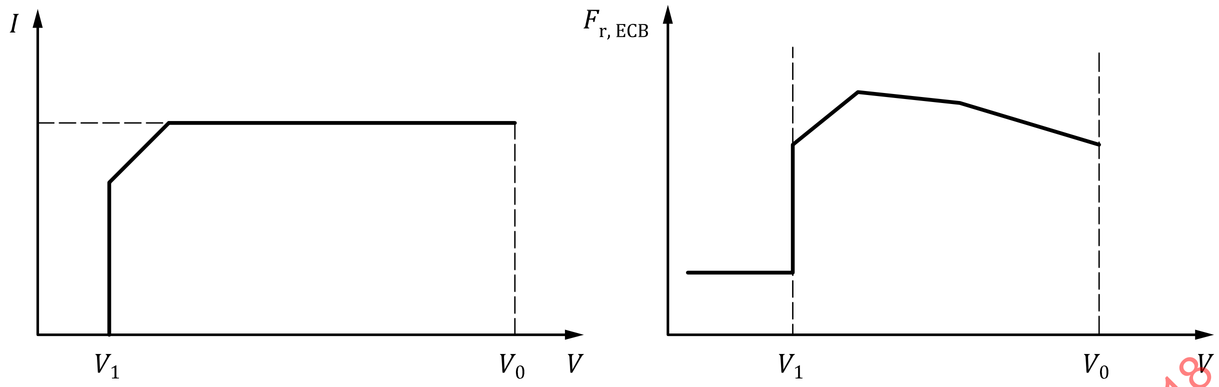

Figure 13 — Typical operating current and retarding force of linear eddy current brake vs. speed

The operating gap is generally constant during the application. Therefore, the mean value of the retarding force depends on the excitation current, *I*, and the operating speed of the eddy current brake.

An example for retarding force of linear eddy current brake calculation is provided in D.2.

NOTE The selection of the operating speed range depends on the properties of the linear eddy current brake.

5.4.3 Mean retarding force by train resistance

The mean retarding force by train resistance is a component of the train decelerating force provided by the structure/shape of the train. The characteristic of train resistance shall be determined by analogy to a similar existing train, or based on a specific calculation or test. When the values are established as a result of tests, the test conditions shall be similar to the expected operating conditions.

NOTE 1 The mean retarding force by train resistance can be neglected for brake calculations which do not take into account rotating masses, such as freight wagons equipped with tread brakes.

The instantaneous retarding force by train resistance, F_{Ra}, for a particular speed, *v*, can be determined as set out in Formula (26):

$$F_{Ra} = C_1 + C_2 \cdot v + C_3 \cdot v^2 \tag{26}$$

where

F_{Ra}	is the instantaneous retarding force by train resistance, expressed in N;
$C_1 + C_2 \cdot v$	is the forces of mechanical resistance and momentum drag due to airflow for traction and auxiliary equipment and the air-conditioning system, expressed in N;
$C_3 \cdot v^2$	is the force of aerodynamic resistance due to pressure drag and skin friction drag, expressed in N;
v	is the speed, expressed in m/s.

The approximation of the mean retarding force by train resistance, $\overline{F}_{Ra}$, for an initial speed, v_0, to final speed, v_{fin}, can be derived from Formula (26) and can be calculated as set out in Formula (27):

$$\overline{F}_{Ra} = C_1 + \frac{2}{3} \cdot C_2 \cdot \frac{v_0^2 + v_0 \cdot v_{fin} + v_{fin}^2}{v_0 + v_{fin}} + \frac{1}{2} \cdot C_3 \cdot \left(v_0^2 + v_{fin}^2\right) \tag{27}$$

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where

$\overline{F}_{\mathrm{Ra}}$ is the mean retarding force by train resistance, expressed in N;

C_1 is the characteristic coefficient of the train independent of speed, expressed in N;

C_2 is the characteristic coefficient of the train proportional to the speed, expressed in $\frac{\mathrm{N}}{\mathrm{m/s}}$;

C_3 is the characteristic coefficient of aerodynamic resistance due to pressure drag and skin friction drag, expressed in $\frac{\mathrm{N}}{(\mathrm{m/s})^2}$;

v_0 is the initial speed, expressed in m/s;

v_{fin} is the final speed, expressed in m/s.

NOTE 2 The determination of the value of the factors C_1, C_2, C_3 is outside the scope of this document.

NOTE 3 The determination of train resistance in tunnels is outside the scope of this document.

5.4.4 Aerodynamic brake

Reserved.

5.4.5 Electro dynamic brake generated by linear induction motor

Reserved.

5.5 Time characteristics

5.5.1 Derivation of brake equipment time characteristics

The time characteristic of a brake equipment type shall be declared for the purposes of the brake calculation.

For the purpose of this document, typically, the time characteristic is considered for each brake equipment type when the braking force of this equipment becomes greater than zero.

Only the application of the brake is considered, the calculations do not consider release characteristics. For specific calculations of slowing distance, the use of release characteristics may be considered.

5.5.2 Equivalent response time, t_e

When a brake application is requested, there is generally an initial delay followed by a period when the braking force builds-up before achieving the maximum braking force for this application. For the purpose of calculation, this response of the brake equipment is generally simplified (see Figure 14) to

— a period when the brake equipment is not applied, and

— a period when the brake equipment is fully applied.

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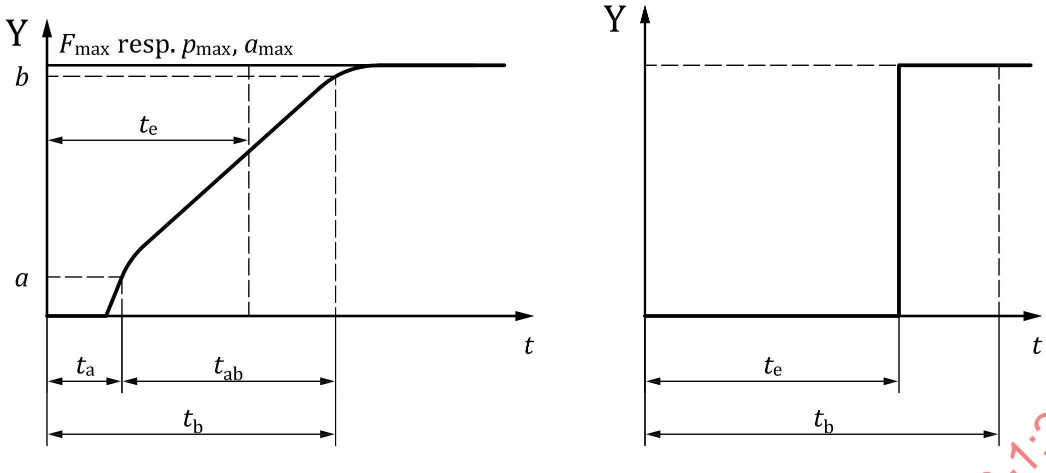

Figure 14 — Actual response of brake equipment vs. simplified equivalent response

One method to determine the equivalent response time, t_e, is set out in Formula (28):

$$t_e = t_a + \frac{t_{ab}}{2} \tag{28}$$

where

t_e is the equivalent response time, expressed in s;

t_a is the initial delay (dead time), expressed in s;

t_{ab} is the build-up time, expressed in s;

t_b is the overall response time ($t_a + t_{ab}$), expressed in s;

a is the defined level for the minimum output signal, (typically 10 % or 5 %), expressed in %;

b is the defined level for the maximum output signal (typically 95 % or 90 %), expressed in %;

Y is the percentage of output signal.

NOTE 1 The determination of the build-up time, t_{ab}, is outside the scope of this document.

NOTE 2 Other methods to determine the equivalent response time, t_e, are shown in ISO/TR 22131.

For units or trainsets with a combination of different brake equipment types, it is necessary to consider system build-up times for brake equipment types which are not pressure systems, e.g. dynamic brake, magnetic track brake, eddy current brake.

The brake performance formulae in this document use mean values and are applicable when the equivalent response time is less than 20 % of the time with the maximum braking force.

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For such units or trainsets with a combination of different brake equipment types, Formula (28) is transferred into Formula (29):

$$t_{e,n} = t_{a,n} + \frac{t_{ab,n}}{2} \quad (29)$$

where

$t_{e,n}$ is the equivalent response time for a specific brake equipment type n, expressed in s;

$t_{a,n}$ is the initial delay (dead time) for a specific brake equipment type n, expressed in s;

$t_{ab,n}$ is the build-up time for a specific brake equipment type n, expressed in s.

$t_{e,n}$ is the equivalent response time of each brake equipment type active within the train. If the response time of the system is affected by the speed of propagation of the command signals, this should be considered.

The equivalent response time, t_e, of the train can be calculated as set out in Formula (30) considering the total number of each brake equipment type at the entity level being considered, e.g. total number of bogies fitted with each brake equipment type.

$$t_e = \frac{\sum_{n=1}^{N} \left(t_{e,n} \cdot \bar{F}_{B,n} \right)}{\sum_{n=1}^{N} \bar{F}_{B,n}} \quad (30)$$

where

$\bar{F}_{B,n}$ is the mean braking force of brake equipment type n, expressed in N;

t_e is the equivalent response time, expressed in s;

$t_{e,n}$ is the equivalent response time for a specific brake equipment type n, expressed in s.

NOTE 3 The equivalent train response time, t_e, can also be determined from the deceleration curve obtained during dynamic braking tests of the train.

5.6 Initial and operating characteristics

5.6.1 Forces on the slope

The components of the weight, F_g [see Formula (31)], on the slope are the downhill force due to gravity, F_D, and the perpendicular force, F_{Perp}, and can be calculated as set out in Formulae (32) and (33).

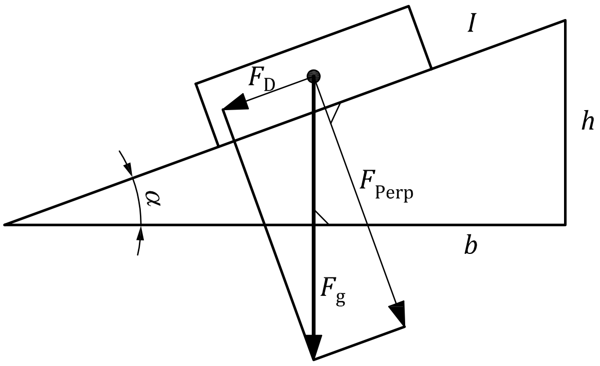

Figure 15 — Forces on the slope

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$$F_g = m_{st} \cdot g \tag{31}$$

$$F_D = F_g \cdot \sin(\alpha) \tag{32}$$

$$F_{Perp} = F_g \cdot \cos(\alpha) \tag{33}$$

where

F_g	is the weight, expressed in N;
m_{st}	is the static mass, expressed in kg;
g	is the standard acceleration of gravity, expressed in m/s^2;
F_D	is the downhill force due to gravity, expressed in N;
F_{Perp}	is the perpendicular force, expressed in N;
α	is the angle of slope, expressed in degree (°).

5.6.2 Downhill force due to gravity depending on the gradient

The gradient is defined in Figure 15 by Formula (34):

$$\tan(\alpha) = \frac{h}{b} \rightarrow h = i \cdot b \tag{34}$$

The relationship between l, b, h is given by Formula (35):

$$l^2 = h^2 + b^2 \rightarrow l = \sqrt{h^2 + b^2} \tag{35}$$

Substituting for h and l, the relationship between $\sin(\alpha)$ and the gradient of the track, i, is set out in Formula (36):

$$\sin\alpha = \frac{h}{l} = \frac{h}{\sqrt{h^2 + b^2}} = \frac{i \cdot b}{\sqrt{i^2 \cdot b^2 + b^2}} = \frac{i \cdot b}{\sqrt{b^2 \left(i^2 + 1\right)}} = \frac{i \cdot b}{b \cdot \sqrt{i^2 + 1}} = \frac{i}{\sqrt{i^2 + 1}} \tag{36}$$

The downhill force due to gravity, F_D, can be calculated using Formulae (32) and (36) as set out in Formula (37):

$$F_D = F_g \cdot \frac{i}{\sqrt{i^2 + 1}} \tag{37}$$

Substituting in Formula (37) with Formula (31), the downhill force due to gravity is set out in Formula (38):

$$F_D = \frac{m_{st} \cdot g \cdot i}{\sqrt{i^2 + 1}} \tag{38}$$

where

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F_D is the downhill force due to gravity, expressed in N;

F_g is the weight, expressed in N;

m_{st} is the static mass, expressed in kg;

g is the standard acceleration of gravity, expressed in m/s²;

i is the gradient of the track.

For gradients not steeper than 1/100 (10 ‰), the gradient is also given by Formula (39):

$$i = \tan\alpha = \frac{h}{b} \approx \frac{h}{l} = \sin\alpha \quad (39)$$

For gradients not steeper than 1/100 (10 ‰), the downhill force due to gravity can be simplified to Formula (40):

$$F_D = m_{st} \cdot g \cdot i \quad (40)$$

where

F_D is the downhill force due to gravity, expressed in N;

m_{st} is the static mass, expressed in kg;

g is the standard acceleration of gravity, expressed in m/s²;

i is the gradient of the track.

Where the gradient of the track changes during the brake performance calculation, it is necessary to determine an equivalent mean gradient.

NOTE The methodology to determine the gradient of the track is outside the scope of this document.

5.6.3 Blending

To achieve the required braking performance, it is common to use the combination of the retardation forces of the different brake systems. When the performance of individual systems changes during the brake application, the concept of blending should ensure the contribution from those systems which are changed to retain the required brake performance (see Figure 16). There is no general blending concept, it is application specific.

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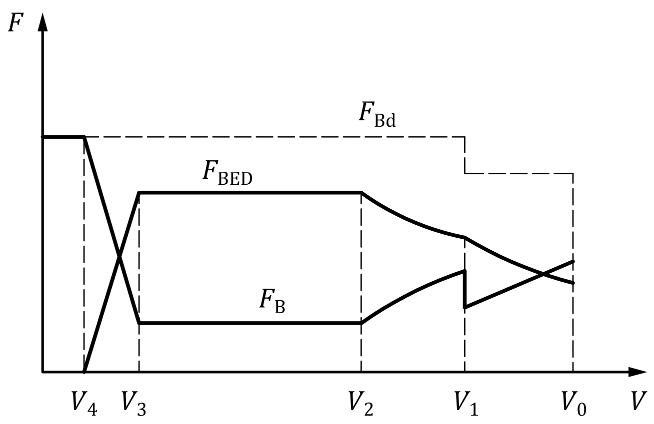

Figure 16 — Idealized example of electro-dynamic brake and friction brake blending vs. speed

The blended retarding force, F_{Bd}, can be calculated as set out in Formula (41):

$$F_{Bd} = F_{BED} + F_B \tag{41}$$

where

F_{Bd}	is the blended retarding force, expressed in N;
F_{BED}	is the retarding force of electro-dynamic brake, expressed in N;
F_B	is the contribution of the friction brake, expressed in N.

When the blended retarding force, F_{Bd}, is constant during the brake application, the calculation of the stopping distance will use this single value.

If the blended retarding force, F_{Bd}, is not constant during the brake application, the calculation of stopping distance shall be performed over a series of speed intervals corresponding to the different values of F_{Bd}.

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5.6.4 Value of the mean adhesion required between wheel/rail for the braked wheelset

The value of mean adhesion required between wheel/rail for the braked wheelset, $\overline{\tau_{ax}}$, is determined from the sum of the mean retarding forces, $\sum_{n=1}^{N} \overline{F_{r,n}}$, of the different brake equipment types, the static mass per wheelset, $m_{st,ax}$, and the gradient of the track, i.

The value of the mean adhesion required between wheel/rail for the braked wheelset can be calculated as set out in Formula (42):

$$\overline{\tau_{ax}} = \frac{\sum_{n=1}^{N} \overline{F_{r,n}} - m_{rot,ax} \cdot a_e}{m_{st,ax} \cdot g} \cdot \sqrt{i^2 + 1} \tag{42}$$

where

$\overline{\tau_{ax}}$	is the value of the mean adhesion required between wheel/rail for the braked wheelset;
N	is the number of brake equipment types;
$\sum_{n=1}^{N} \overline{F_{r,n}}$	is the sum of mean retarding forces of all brake equipment types per wheelset requiring adhesion, expressed in N;
$m_{rot,ax}$	is the equivalent rotating mass, expressed in kg;
a_e	is the equivalent deceleration, expressed in m/s^2;
$m_{st,ax}$	is the static mass per wheelset, expressed in kg;
g	is the standard acceleration of gravity, expressed in m/s^2;
i	is the gradient of the track (positive rising/negative falling).

NOTE The methodology to determine the value of permitted adhesion is outside the scope of this document.

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5.7 Stopping and slowing distance calculation based on mean values

5.7.1 Mean retarding force with respect to the distance

The mean retarding force shall be used for the calculation of braking distance. If the instantaneous retarding force is constant, it does not depend on the speed range. The mean retarding force with respect to the distance generated by all the brake equipment acting together can be calculated as set out in Formula (43):

$$\bar{F}_r = \frac{v_0^2 - v_{fin}^2}{2} \cdot 1 / \int_{v_{fin}}^{v_0} \frac{v}{F_r} dv \quad (43)$$

where

$\bar{F}_r$ is the mean retarding force acting at the rail generated by the brake equipment, expressed in N;

F_r is the instantaneous retarding force acting at the rail generated by the brake equipment, expressed in N;

v_0 is the initial speed, expressed in m/s;

v_{fin} is the final speed (= 0 in the case of a stopping distance), expressed in m/s;

v is the speed, expressed in m/s;

with

$$s = m_{dyn} \cdot \int_{v_{fin}}^{v_0} \frac{v}{F_r(v)} dv \quad (44)$$

and in case of constant brake force,

$$s = \frac{m_{dyn}}{\bar{F}_r} \cdot \frac{v_0^2 - v_{fin}^2}{2} \quad (45)$$

where

s is the stopping / slowing distance, expressed in m;

m_{dyn} is the dynamic mass, expressed in kg.

Equalizing Formulae (44) and (45) on the distance base creates Formula (43).

If the instantaneous retarding forces are defined in different speed ranges, the integral shall be calculated for each range of speed and finally summed. This results in the mean retarding force being obtained for all the speed ranges.

5.7.2 Equivalent deceleration, a_e, based on retarding forces

The equivalent deceleration with respect to the braking distance is calculated as the sum of the retarding forces during braking, divided by the dynamic mass as set out in Formula (46) and is applicable to the whole train and takes into account external forces acting on the train. This equivalent deceleration represents the mean deceleration with fully applied brake forces for all applied brake equipment types.

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The mean external forces, $\overline{F}_{ext}$, are forces acting on the vehicle/train and are not generated by the vehicle/train itself (e.g. wind).

$$a_e = \frac{\sum_{n=1}^{N} \overline{F}_{r,n} + \sum \overline{F}_{ext}}{m_{dyn}} \quad (46)$$

where

a_e is the equivalent deceleration, expressed in m/s^2;

$\overline{F}_{r,n}$ is the mean retarding force of brake equipment type n, expressed in N;

$\overline{F}_{ext}$ is the mean external force respecting the direction of action, expressed in N;

m_{dyn} is the dynamic mass, expressed in kg.

5.7.3 Equivalent free running distance, s_0

The equivalent free running distance, s_0, is a calculated theoretical distance without deceleration or acceleration on level track as set out in Formula (47):

$$s_0 = v_0 \cdot t_e \quad (47)$$

where

s_0 is the equivalent free running distance, expressed in m;

v_0 is the initial speed, expressed in m/s;

t_e is the equivalent response time, expressed in s.

5.7.4 Stopping and slowing distance on level track

The stopping or slowing distance (s) on level track is calculated as the distance run between the initial brake demand and achieving the final speed as set out in Formula (48):

$$s = v_0 \cdot t_e + \frac{v_0^2 - v_{fin}^2}{2 \cdot a_e} \quad (48)$$

where

s is the stopping/slowing distance, expressed in m;

t_e is the equivalent response time, expressed in s;

v_0 is the initial speed, expressed in m/s;

v_{fin} is the final speed (=0 in the case of a stopping distance), expressed in m/s;

a_e is the equivalent deceleration, expressed in m/s^2.

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To take into account changes in retarding forces due to different speed ranges, the stopping/slowing distance, s, is calculated as set out in Formula (49):

$$s = v_0 \cdot t_e + \frac{v_0^2 - v_z^2}{2 \cdot a_{e,z}} + \frac{v_z^2 - v_{z+1}^2}{2 \cdot a_{e,z+1}} + \ldots + \frac{v_{Z-1}^2 - v_Z^2}{2 \cdot a_{e,Z}} \quad (49)$$

where

$a_{e,z}$ is the equivalent deceleration acting during speed range z, expressed in m/s^2;

z is the speed range step number;

Z is the number of speed ranges.

NOTE Other methods to determine the stopping and slowing distance are shown in ISO/TR 22131.

5.7.5 Stopping and slowing distance based on different gradients

5.7.5.1 Stopping and slowing distance on a gradient

The stopping and slowing distance on a gradient, s_{grad}, can be calculated as set out in Formula (50). This calculation takes full account of the effects of the gradient of the track and rotating inertia.

$$s_{grad} = v_0 \cdot t_e - \frac{1}{2} \cdot \frac{m_{st}}{m_{dyn}} \cdot g \cdot i \cdot t_e^2 + \frac{\left(v_0 - \frac{m_{st}}{m_{dyn}} \cdot g \cdot i \cdot t_e\right)^2 - v_{fin}^2}{2 \cdot a_{e,grad}} \quad (50)$$

with

$$a_{e,grad} = a_e + \frac{m_{st}}{m_{dyn}} \cdot g \cdot i \quad (51)$$

where

s_{grad} is the stopping/slowing distance on a gradient, expressed in m;

t_e is the equivalent response time, expressed in s;

v_0 is the initial speed, expressed in m/s;

v_{fin} is the final speed (= 0 in case of a stopping distance), expressed in m/s;

m_{st} is the static mass, expressed in kg;

m_{dyn} is the dynamic mass, expressed in kg;

g is the standard acceleration of gravity, expressed in m/s^2;

i is the gradient of the track (positive rising/negative falling);

a_e is the equivalent deceleration, expressed in m/s^2;

$a_{e,grad}$ is the equivalent deceleration including the effect of gradient and inertia, expressed in m/s^2.

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5.7.5.2 Stopping and slowing distance on a gradient (simplified)

The stopping and slowing distance on a gradient, s_{grad}, set out in Formula (50) can be simplified as set out in Formula (52). In this simplified calculation, the speed changes during the free running time and the effects of rotating inertia are ignored. The simplified formula is valid under the following conditions:

— maximum gradients of the track not steeper than 1/100 (10 ‰);

— initial speed is not less than 50 km/h;

— equivalent response time of the train brake is not more than 3 s.

Within these limits, the expected error from the use of Formula (52) is <5 %.

$$s_{grad} = v_0 \cdot t_e + \frac{v_0^2 - v_{fin}^2}{2 \cdot a_{e,grad\_simple}} \tag{52}$$

with

$$a_{e,grad\_simple} = a_e + g \cdot i \tag{53}$$

where

s_{grad}	is the stopping/slowing distance on a gradient, expressed in m;
v_0	is the initial speed, expressed in m/s;
t_e	is the equivalent response time, expressed in s;
v_{fin}	is the final speed (= 0 in the case of a stopping distance), expressed in m/s;
$a_{e,grad\_simple}$	is the equivalent deceleration neglecting inertia, expressed in m/s^2;
a_e	is the equivalent deceleration, expressed in m/s^2;
g	is the standard acceleration of gravity, expressed in m/s^2;
i	is the gradient of the track.

NOTE Other methods to determine the stopping and slowing distance are shown in ISO/TR 22131.

5.8 Supplementary dynamic calculations

5.8.1 General

The following calculations are conducted in addition to stopping and slowing distances or deceleration calculation. They may be used to evaluate the design of the braking system particularly regarding the dissipation of braking energy by the brake equipment types as applicable.

The following calculations may be done using mean values of force or instantaneous values of force.

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5.8.2 Braking energy

5.8.2.1 Total energy, W_{tot}

The total energy, W_{tot}, is the sum of the dissipated energy of all applied brake equipment types and train resistance as set out in 5.4.3, which is equal to the related difference of kinetic and potential energy as set out in Formula (54):

$$W_{tot} = \frac{m_{dyn} \cdot \left(v_0^2 - v_{fin}^2\right)}{2} - m_{st} \cdot g \cdot s \cdot \frac{i}{\sqrt{i^2+1}} = W_B + W_{Ra} \quad (54)$$

where

W_{tot}	is the total energy, expressed in J;
W_B	is the energy dissipated by the brake systems, expressed in J;
W_{Ra}	is the energy dissipated by the train resistance, expressed in J;
g	is the standard acceleration of gravity, expressed in m/s²;
i	is the gradient of the track;
m_{dyn}	is the dynamic mass, expressed in kg;
m_{st}	is the static mass, expressed in kg;
v_0	is the initial speed, expressed in m/s;
v_{fin}	is the final speed, expressed in m/s;
s	is the stopping/slowing distance (distance between v_0 and v_{fin}), expressed in m.

NOTE In Formula (54), the potential energy is represented by $s \cdot \frac{i}{\sqrt{i^2+1}}$, i.e. the height difference between the start and finish of the brake demand.

5.8.2.2 Energy dissipated by each brake equipment type, $W_{B,n}$

The energy, $W_{B,n}$, dissipated by each brake equipment type n can be calculated for mean retarding forces $\overline{F_{r,n}}$ as set out in Formula (55):

$$W_{B,n} = \overline{F_{r,n}} \cdot s_{B,n} \quad (55)$$

where

$W_{B,n}$	is the energy dissipated by brake equipment type n, expressed in J;
$\overline{F_{r,n}}$	is the mean retarding force for brake equipment type n, expressed in N;
$s_{B,n}$	is the distance travelled while the brake equipment type n is applied, expressed in m.

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5.8.3 Maximum braking power of each brake equipment type, $P_{max,n}$

The maximum braking power, $P_{max,n}$, of each brake equipment type n can be calculated as set out in Formula (56), when the retarding force is constant.

$$P_{max,n} = \overline{F_{r,n}} \cdot v_0 \tag{56}$$

where

$P_{max,n}$ is the maximum power of brake equipment type n, expressed in W;

$\overline{F_{r,n}}$ is the mean retarding force of brake equipment type n, expressed in N;

v_0 is the initial speed, expressed in m/s.

NOTE If the retarding force, $F_{r,n}$, of brake equipment type n changes during the application, the maximum power, $P_{max,n}$, of brake equipment type n may not occur at the initial speed, v_0.

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6 Stationary braking

6.1 General

Stationary braking provides the following functions:

— holding brake;

— immobilization brake;

— parking brake.

6.2 Holding brake

The holding brake is used to prevent a train from moving under specified conditions and for a specified period of time when the brake system energy is replenished.

The holding brake provides the following functions:

— to secure the train at standstill during a stop in a station;

— to secure the train on a gradient during a hill start.

NOTE The holding brake is sometimes provided by the leading vehicle and/or by the locomotive alone.

6.3 Immobilization brake

The immobilization brake holds a train in a stationary position under relevant load conditions for a certain period of time and at least on a gradient. The immobilization brake is normally initiated by an application of the service brake or emergency brake using just the brake system energy stored on the train.

6.4 Parking brake

The parking brake keeps the train stationary in shut down configuration for an indefinite period of time without energy supply until intentionally released.

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6.5 Stationary brake calculation

6.5.1 General

The stationary brake calculation is generally not performed for the immobilization brake, as all wheelsets have their service/emergency brakes applied. In the case of parking brake and holding brake, only a limited quantity of wheelsets are braked, so it is necessary to calculate the stationary brake performance.

NOTE The parking brake and holding brake function may be provided by different wheelsets in the train formation.

The principle of the algorithm flow is presented in Figure B.1.

6.5.2 General characteristics

The parameters to be defined are:

— quantity of parking braked wheelsets for each adhesion dependent brake equipment type;

— quantity of holding braked wheelsets for each adhesion dependent brake equipment type;

— quantity of non-adhesion dependent brake equipment types;

— axle load of the parking and holding braked wheelsets;

— static coefficient of friction (block or pad);

— static mass of the vehicle/unit/train;

— gradient;

— wind forces;

— rolling resistance.

Each brake equipment type used for holding and/or parking shall be the subject of a specific calculation.

All of the various types of brake equipment applied to one wheelset shall be identified and considered.

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6.6 Static coefficient of friction

The static coefficient of friction is the main characteristic of friction brake equipment types to be taken into account in the stationary brake performance.

NOTE The methodology to determine the static friction coefficient of the brake blocks/pads is outside the scope of this documents.

6.7 Parking brake force provided by equipment type

6.7.1 Screw applied parking brake (tread brake)

Different technical solutions for generating the output force of parking brake exist. The example shown in Figure 17 is one typical solution using a tread brake.

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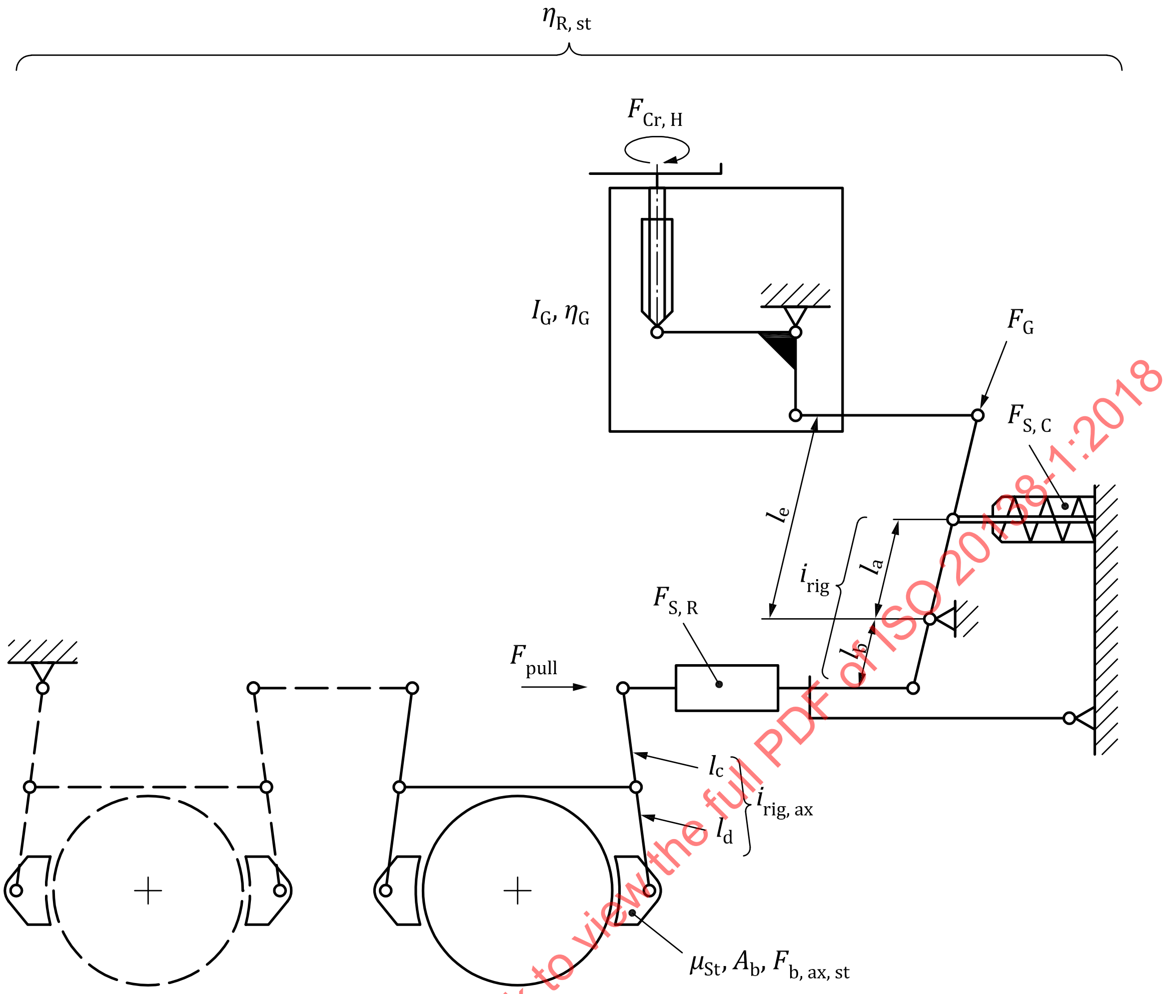

NOTE 1 The application point of the gear force, F_G, can be the application point of the piston force.

NOTE 2 Dashed lines represent additional tread brake arrangements.

Figure 17 — Typical screw hand brake (tread brake)

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The gear force, F_G, is generated by the hand wheel or a crank handle force, $F_{Cr,H}$, and is transferred to the brake blocks taking into account the specific mechanical rigging ratios (including hand brake gear), efficiencies of brake rigging and counter forces.

In the example shown in Figure 17, the output force of parking brake mechanism, F_G, can be calculated as set out in Formula (57):

$$F_G = F_{Cr,H} \cdot i_G \cdot \eta_G \tag{57}$$

where

F_G	is the output force of parking brake mechanism, expressed in N;
$F_{Cr,H}$	is the crank handle or hand wheel force, expressed in N;
i_G	is the gear ratio;
η_G	is the gear efficiency.

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The theoretical maximum of the static force on application point bogie, $F_{pull,st}$, can be calculated as set out in Formula (58):

$$F_{pull,st} = F_G \cdot \frac{l_e}{l_b} - F_{S,C} \cdot i_{rig} - F_{S,R} \quad (58)$$

The theoretical maximum of the static force acting on a single block, $F_{b,ax,st}$, can be calculated as set out in Formula (59):

$$F_{b,ax,st} = \frac{1}{2} \cdot F_{pull,st} \cdot i_{rig,ax,n} \quad (59)$$

where

Table 4 — Formulae for single-sided and double-sided brake block arrangement

	Brake block arrangement	
	Single-sided	**Double-sided (clasp brake)**
$i_{rig,ax,n}$	$i_{rig,ax,n} = \frac{l_{c,i} + l_{d,i}}{l_{d,i}}$ (60) see Figure 6	$i_{rig,ax,n} = \frac{l_{c,i}}{l_{d,i}}$ (61) see Figure 5

The total static brake block force can be calculated as set out in Formula (62):

$$F_{b,tot,st} = \sum F_{b,ax,st} \cdot \eta_{R,st} \quad (62)$$

where

$F_{pull,st}$	is the static force on application point bogie, expressed in N;
F_G	is the output force of parking brake mechanism, expressed in N;
$F_{S,C}$	is the restoring force of brake unit or spring applied force, expressed in N;
i_{rig}	is the rigging ratio;
l_a, l_b	are the main brake lever lengths, expressed in m;
l_e	is the main brake lever length (parking brake), expressed in m;
l_c, l_d	are the bogie lever lengths, expressed in m;
$F_{S,R}$	is the restoring force, e.g. slack adjuster, expressed in N;
$F_{b,tot,st}$	is the total static brake block force, expressed in N;
$F_{b,ax,st}$	is the static single brake block force, expressed in N;
$i_{rig,ax,n}$	is the lever ratio per brake beam;
$\eta_{R,st}$	is the overall static efficiency of brake rigging.

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The total stationary brake force acting at the rail applied by the parking brake can be calculated as set out in Formula (63):

$$F_{B,st} = F_{b,tot,st} \cdot \mu_{st} \tag{63}$$

where

$F_{B,st}$ is the total stationary brake force acting at the rail, expressed in N;

$F_{b,tot,st}$ is the total static brake block force, expressed in N;

μ_{st} is the static friction coefficient of brake block.

NOTE The methodology to determine the static friction coefficient of the brake blocks is outside the scope of this document.

6.7.2 Spring applied tread brake unit

The spring applied parking brake force is transferred to the brake blocks taking into account the specific mechanical rigging ratios, efficiencies of brake rigging and counter forces. The example shown in Figure 18 is one typical solution using a spring applied tread brake unit.

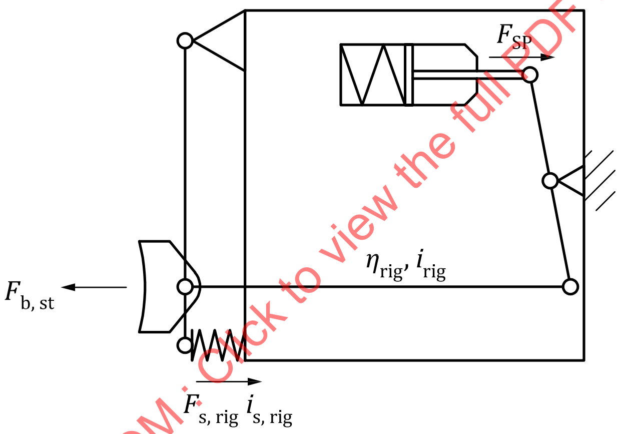

Figure 18 — Representative spring applied tread brake unit

The brake block force for an arrangement as shown in Figure 18 is calculated as set out in Formula (64):

$$F_{b,st} = F_{SP} \cdot i_{rig} \cdot \eta_{rig} - F_{s,rig} \cdot i_{s,rig} \tag{64}$$

where

$F_{b,st}$ is the parking brake force acting on the tread of the wheel from a single parking brake unit, expressed in N;

F_{SP} is the parking brake spring force, expressed in N;

i_{rig} is the rigging ratio;

η_{rig} is the efficiency of brake rigging;

$F_{s,rig}$ is the restoring force, expressed in N;

$i_{s,rig}$ is the rigging ratio for restoring force.

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In general, $i_{s,rig} = 1$, so Formula (64) can be simplified to Formula (65):

$$F_{b,st} = F_{SP} \cdot i_{rig} \cdot \eta_{rig} - F_{s,rig} \tag{65}$$

The total parking brake force acting at the rail depends on the quantity of parking brake units and the static friction coefficient of the brake block and is independent of the wheel diameter as set out in Formula (66):

$$F_{PB} = F_{b,st} \cdot \mu_{st} \cdot n_{SP} \tag{66}$$

where

F_{PB} is the total parking brake force acting at the rail, expressed in N;

$F_{b,st}$ is the parking brake force acting on the tread of the wheel from a single parking brake unit, expressed in N;

μ_{st} is the static friction coefficient of brake block;

n_{SP} is the number of spring brake units.

NOTE The methodology to determine the static friction coefficient, μ_{st}, of the brake blocks is outside the scope of this document.

6.7.3 Screw applied parking brake (disc brake)

Different technical solutions for generating the output force of parking brake exist. The example shown in Figure 19 is one typical solution using a disc brake.

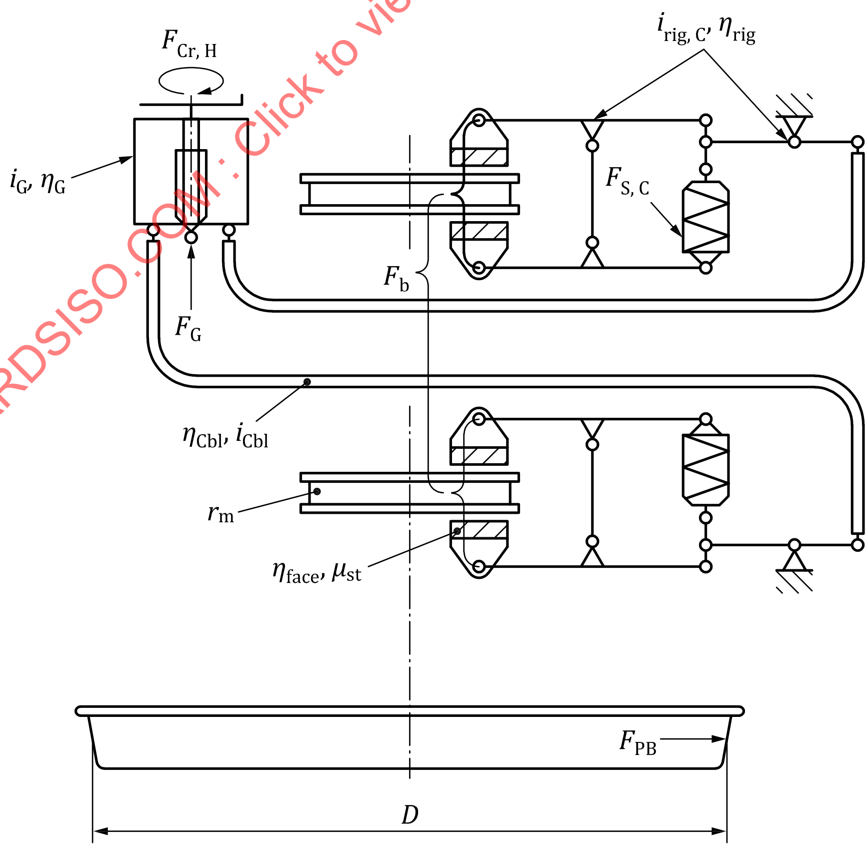

Figure 19 — Typical screw brake arrangement of disc brakes

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The parking brake force, F_G, is transferred to the brake pads taking into account the specific mechanical calliper ratios, efficiencies and counter forces. The example shown in Figure 19 uses a screw hand brake and the output force of parking brake mechanism, F_G, is set out in Formula (67):

$$F_G = F_{Cr,H} \cdot i_G \cdot \eta_G \tag{67}$$

where

F_G is the output force of parking brake mechanism, expressed in N;

$F_{Cr,H}$ is the crank handle or hand wheel force, expressed in N;

i_G is the gear ratio;

η_G is the gear efficiency.

The total force, $F_{b,tot}$, acting on all disc faces of the arrangement as shown in Figure 19 can be calculated as set out in Formula (68):

$$F_{b,tot} = \left(F_G \cdot \eta_{Cbl} \cdot i_{Cbl} - F_{S,C}\right) \cdot n_{face} \cdot i_{rig,C} \cdot \eta_{rig} \tag{68}$$

where

$F_{b,tot}$ is the total force acting on all disc faces, expressed in N;

F_G is the output force of parking brake mechanism, expressed in N;

η_{Cbl} is the cable efficiency;

i_{Cbl} is the cable mechanical ratio;

$F_{S,C}$ is the restoring force of brake unit or spring applied force, expressed in N;

n_{face} is the number of disc faces;

$i_{rig,C}$ is the calliper lever ratio (parking brake);

η_{rig} is the efficiency of brake calliper.

The total parking brake force acting at the rail, F_{PB}, can be calculated as set out in Formula (69):

$$F_{PB} = F_{b,tot} \cdot \mu_{st} \cdot \frac{r_m}{D/2} \tag{69}$$

where

F_{PB} is the total parking brake force acting at the rail, expressed in N;

$F_{b,tot}$ is the total force acting on all disc faces, expressed in N;

μ_{st} is the static friction coefficient of brake pad;

r_m is the mean swept radius of the brake pad on the disc face, expressed in m;

D is the wheel diameter, expressed in m.

NOTE The methodology to determine the static friction coefficient, μ_{st}, of the brake pads is outside the scope of this document.

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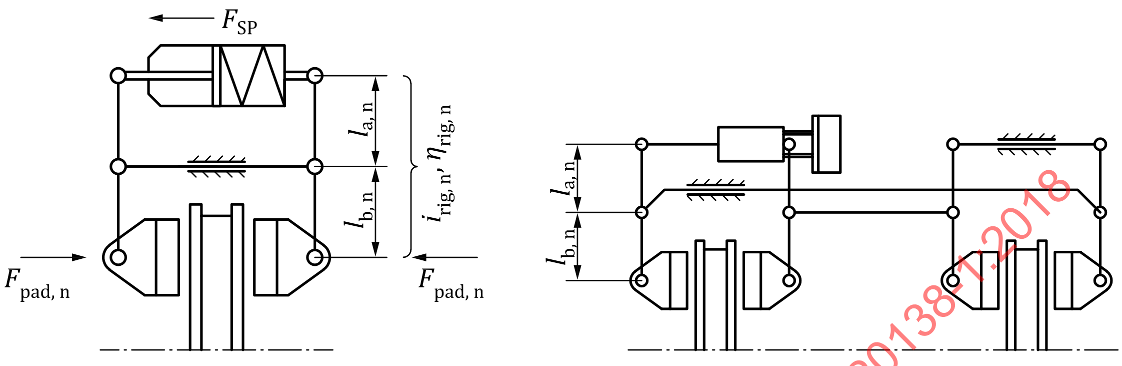
6.7.4 Spring applied disc brake unit arrangement

The spring applied parking brake force is transferred to the brake pads taking into account the specific mechanical calliper ratios, efficiencies and counter forces. The individual brake pad force (single face of the disc) for an arrangement as shown in Figure 20 can be calculated as set out in Formula (70).



a) Two disc faces

b) Four disc faces

Figure 20 — Typical arrangement of spring applied brake disc calliper

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The force, $F_{pad,n}$, acting on an individual disc face can be calculated as set out in Formula (70):

$$F_{pad,n} = F_{SP} \cdot i_{rig,n} \cdot \eta_{rig,st} \tag{70}$$

where

$$i_{rig,n} = \frac{l_{a,n}}{l_{b,n}} \tag{71}$$

$F_{pad,n}$ is the force acting on single disc surface, expressed in N;

F_{SP} is the parking brake spring force, expressed in N;

$i_{rig,n}$ is the calliper lever ratio;

$\eta_{rig,st}$ is the static efficiency of calliper;

$l_{a,n}$, $l_{b,n}$ are the calliper lever lengths, expressed in m.

The sum of the spring applied forces acting on the disc faces can be calculated as set out in Formula (72):

$$F_{b,tot.st} = n_{face} \cdot F_{pad,n} \tag{72}$$

where

$F_{b,tot,st}$ is the parking brake forces acting on the disc faces, expressed in N;

$F_{pad,n}$ is the force acting on single disc surface, expressed in N;

n_{face} is the number of disc faces.

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Total parking brake force acting at the rail can be calculated as set out in Formula (73):

$$F_{PB} = F_{b,tot,st} \cdot \mu_{st} \cdot \frac{r_m}{D/2} \quad (73)$$

where

F_{PB} is the total parking brake force acting at the rail, expressed in N;

$F_{b,tot,st}$ is the parking brake forces acting on the disc faces, expressed in N;

μ_{st} is the static friction coefficient of brake pad;

r_m is the mean swept radius of the brake pad on the disc face, expressed in m;

D is the wheel diameter, expressed in m.

NOTE The methodology to determine the static friction coefficient, μ_{st}, of the brake pads is outside the scope of this document.

6.7.5 Force of a permanent magnetic track brake

The parking brake force of one permanent magnet can be calculated as set out in Formula (74):

$$F_{Mg,st} = F_{AMg,st} \cdot \mu_{Mg,st} \quad (74)$$

where

$F_{Mg,st}$ is the parking brake force of one permanent magnet, expressed in N;

$F_{AMg,st}$ is the attraction force of one permanent magnet, expressed in N;

$\mu_{Mg,st}$ is the static friction coefficient of permanent magnet (pole shoe).

NOTE The methodology to determine the attraction force and the static friction coefficient of the permanent magnet is outside the scope of this document.

The total parking brake force of all permanent magnets in a vehicle, $F_{Mg,stat,tot}$, can be calculated as set out in Formula (75):

$$F_{Mg,st,tot} = n_{Mg} \cdot F_{Mg,st} \quad (75)$$

where

$F_{Mg,st,tot}$ is the total parking brake force of all permanent magnets in a vehicle, expressed in N;

n_{Mg} is the number of magnets in a vehicle;

$F_{Mg,st}$ is the parking brake force of one permanent magnet, expressed in N.

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6.8 Stationary brake force for each wheelset

The stationary force for each wheelset, $F_{B,ax,st}$, is the sum of the stationary forces of each adhesion dependent immobilization/holding/parking brake equipment type acting on that wheelset, $F_{st,n}$, as set out in Formula (76).

$$F_{B,ax,st} = \left(\sum_{0}^{N} F_{st,n} \right)_{ax} \tag{76}$$

where

$F_{B,ax,st}$ is the stationary brake force acting on that wheelset, expressed in N;

$F_{st,n}$ is the stationary brake force acting on the wheelset of each immobilization/holding/parking brake type n, expressed in N.

The individual stationary brake forces may be limited by the available adhesion. Therefore, the transmittable force shall be established. The individual stationary brake force is the lower value of either the adhesion transmittable force or the stationary brake applied force. It can be calculated as set out in Formula (77):

$$F_{st,ax} = \min\left[\left(F_{B,ax,st} \right) \text{or} \left(\tau_a \cdot m_{st,ax} \cdot g \cdot i \right) \right] \tag{77}$$

where

$F_{st,ax}$ is the transmittable stationary brake force acting on that wheelset, expressed in N;

$F_{B,ax,st}$ is the stationary brake force on that wheelset, expressed in N;

g is the standard acceleration of gravity, expressed in m/s^2;

$m_{st,ax}$ is the static mass per wheelset, expressed in kg;

τ_a is the available adhesion;

i is the gradient of the track.

NOTE The methodology to determine the available adhesion value, τ_a, is outside the scope of this document.

6.9 Total stationary brake force per train

The stationary brake force of the train is the sum of all stationary brake forces including the forces of adhesion dependent and adhesion independent brake equipment types; it can be calculated as set out in Formula (78):

$$F_{st} = \sum F_{B,ind} + \sum_{ax} F_{st,ax} \tag{78}$$

where

F_{st} is the stationary brake force of the train, expressed in N;

$F_{st,ax}$ is the transmittable stationary brake force acting on that wheelset, expressed in N;

$F_{B,ind}$ is the adhesion independent (not related to the wheel to rail contact) force, e.g. force of permanent magnetic track brake, expressed in N.

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6.10 Stationary brake safety calculation

The stationary brake safety calculation is performed to check whether the train is equipped with enough parking brake/holding brake forces to prevent the train from moving in a first step (see 6.11) and whether the distribution of brake forces prevents those wheelsets from sliding in a second step (see 6.12). Both issues shall be fulfilled to keep the train stationary.

The capability to keep the train stationary can also be expressed by the maximum gradient (see 6.13).

If the vehicle/unit/train has different axle loads and/or the parking brake equipment generates different parking brake forces, then the calculation of safety against rolling and safety against sliding should use the methods set out in 6.14.

6.11 Safety ratio for stationary brake

The ratio of the stationary brake force on the train to the forces that would accelerate the train shall be greater than one and can be calculated as set out in Formula (79):

$$S_{st} = \frac{F_{st} + F_{Ra}}{F_D + F_{wind}} \tag{79}$$

where

S_{st}	is the safety ratio for stationary brake;
F_{st}	is the stationary brake force of the train, expressed in N;
F_D	is the downhill force due to gravity, expressed in N;
F_{wind}	is the wind force on the train, expressed in N;
F_{Ra}	is the retarding force by train resistance, expressed in N.

For downhill and/or wind force and train resistance force, refer to 5.6.2 and 5.4.3.

6.12 Coefficient of adhesion required by each disc braked wheelset

The coefficient of adhesion, $\tau_{req,ax}$, required to ensure that each braked wheelset will not slide is determined from the parking brake force, the static axle load and the effect of gradient. It can be calculated as set out in Formula (80):

$$\tau_{req,ax} = \frac{F_{PB,ax}}{m_{st,ax} \cdot g / \sqrt{i^2 + 1}} \tag{80}$$

with

$$F_{PB,ax} = F_{cl} \cdot n_{disc} \cdot \mu_m \cdot \frac{r_m}{D/2} \tag{81}$$

$$i = \tan \alpha \tag{82}$$

$$\cos(\alpha) = \frac{1}{\sqrt{i^2 + 1}} \tag{83}$$

where

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$\tau_{req,ax}$ is the coefficient of adhesion required by each braked wheelset;

$F_{PB,ax}$ is the parking brake force per wheelset acting at the rail, expressed in N;

$m_{st,ax}$ is the static mass per wheelset, expressed in kg;

g is the standard acceleration of gravity, expressed in m/s^2;

i is the gradient of the track;

F_{cl} is the clamping force, expressed in N;

n_{disc} is the number of brake discs;

μ_m is the mean friction coefficient of brake block/brake pad;

r_m is the mean swept radius of the brake pad on the disc face, expressed in m;

D is the wheel diameter, expressed in m;

α is the angle of slope (see Figure 15), expressed in degree (°);

Generally, there is a maximum adhesion value, τ_{max}, that is specified for the design of the stationary braking.

The calculated value, $\tau_{req,ax}$, shall be less than or equal the maximum permitted value of τ_{max}.

NOTE 1 The methodology to determine the maximum permitted adhesion value is outside the scope of this document.

NOTE 2 Formula (80) can be adapted to assess the required adhesion for the holding brake.

NOTE 3 For block brake or tread brake, Formula (81) is replaced by Formula (63) or (66), respectively.

6.13 Maximum achievable gradient

The maximum achievable gradient is derived from the available stationary braking force, wind force, and downhill force.

An increase of the gradient causes a higher downhill force and can result in an insufficient holding force. If the gradient on the track is steeper than i_{max}, the vehicle starts moving by sliding or rolling. Therefore, the maximum achievable gradient shall be calculated for both cases, $i_{max,slide}$ and $i_{max,roll}$.

The availability of adhesion to prevent sliding and the limit conditions for rolling shall be checked for each individual wheelset.

Where all wheelsets are braked, the maximum achievable gradient for sliding, $i_{max,slide}$, can be calculated as set out in Formula (84):

$$i_{max,slide} = \frac{1}{\sqrt{\left(\frac{m_{st} \cdot g}{F_{PB} - F_{wind}}\right)^2 - 1}} \quad (84)$$

Where all wheelsets are braked, the maximum achievable gradient for rolling, $i_{max,roll}$, can be calculated as set out in Formula (85):

$$i_{max,roll} = \frac{1}{\sqrt{\left(\frac{m_{st} \cdot g}{F_{PB} + F_{Ra,st} - F_{wind}}\right)^2 - 1}} \quad (85)$$

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where

$i_{max,slide}$ is the maximum achievable gradient for sliding;

F_{PB} is the total parking brake force acting at the rail, expressed in N;

F_{wind} is the wind force on the train, expressed in N;

m_{st} is the static mass, expressed in kg;

g is the standard acceleration of gravity, expressed in m/s^2;

$i_{max,roll}$ is the maximum achievable gradient for rolling;

$F_{Ra,st}$ is the stationary train resistance force, expressed in N.

NOTE The methodology to determine the stationary train resistance force and wind forces is outside the scope of this document.

6.14 Method for safety calculation for vehicles with a different relationship between brake force and load per wheelset

6.14.1 General

For parking and holding brake calculation, the proportion of the mass of the vehicle/unit/train which has to be held by those wheelsets fitted with parking/holding brake equipment shall be taken into account. In vehicles/units there are wheelsets with and without stationary brake equipment.

Analogously to 5.6.1, the parking brake forces acting on a single wheelset as shown in Figure 21 are:

— proportion of downhill force to be resisted by the wheelsets with applied parking brake, $F_{d,ax}$;

— static axle load perpendicular to the rail per wheelset with applied parking brake, $F_{N,ax}$;

— static axle load, $F_{g,ax}$, due to gravity,

and can be calculated as set out in Formulae (86), (87) and (88).

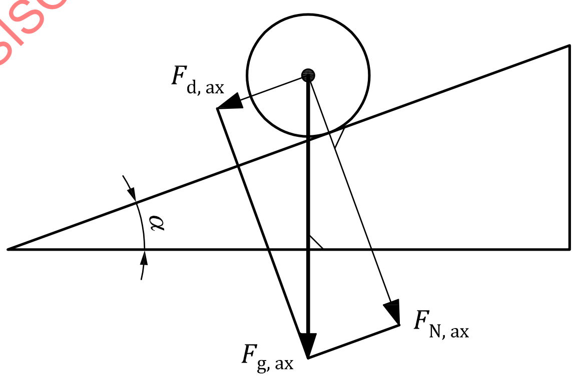

Figure 21 — Effect of the gradient on the static axle load

$$F_{d,ax} = F_{g,ax} \cdot \sin(\alpha) \quad (86)$$

$$F_{N,ax} = F_{g,ax} \cdot \cos(\alpha) \quad (87)$$

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$$F_{g,ax} = m_{st,ax} \cdot g \tag{88}$$

where

$F_{g,ax}$ is the static axle load, expressed in N;

$F_{d,ax}$ is the proportion of downhill force to be resisted by the wheelsets with applied parking brakes, expressed in N;

$F_{N,ax}$ is the static axle load perpendicular to the rail per wheelset with applied parking brake for a specific wheelset, expressed in N;

α is the angle of slope, expressed in degree (°);

$m_{st,ax}$ is the static mass per wheelset, expressed in kg;

g is the standard acceleration of gravity, expressed in m/s^2.

6.14.2 Mean adhesion value between wheel/rail

The value of mean adhesion required between wheel/rail for the braked wheelsets, $\overline{\tau_{ax}}$, is determined from the proportion of downhill force to be resisted per wheelset with applied parking brakes, $F_{d,ax}$, and the static axle load, $F_{N,ax}$, perpendicular to the rail per wheelset with applied parking brake and can be calculated as set out in Formula (89):

$$\overline{\tau_{ax}} = \frac{\sum F_{d,ax}}{\sum F_{N,ax}} \tag{89}$$

where

$\overline{\tau_{ax}}$ is the value of the mean adhesion required between wheel/rail for the braked wheelset;

$F_{d,ax}$ is the proportion of downhill force to be resisted per wheelset with applied parking brakes, expressed in N;

$F_{N,ax}$ is the static axle load perpendicular to the rail per wheelset with applied parking brake, expressed in N.

NOTE In case of equal distribution of axle load along the whole vehicle and equal stationary brake force acting on each wheelset, the mean adhesion value, τ, corresponds to the adhesion value of each wheelset.

6.14.3 Safety against rolling

To prevent rolling, the total parking brake force acting at the rail, F_{PB}, shall be capable of resisting the downhill force due to gravity, F_D, generated by the static mass, m_{st}, and external forces, F_{ext}, (e.g. wind), as set out in Formula (90):

$$F_{PB} > F_D + F_{ext} \tag{90}$$

The downhill force due to gravity, F_D, can be calculated as set out in Formula (91):

$$F_D = m_{st} \cdot g \cdot \sin(\alpha) \tag{91}$$

Safety against rolling, S_R, applies to the vehicle/unit/train and represents the relationship between the total parking brake force acting at the rail, F_{PB}, and the stationary train resistance force, $F_{Ra,st}$, and

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adhesion independent forces, $F_{B,ind}$, vs. the downhill force due to gravity, F_D, and external forces, F_{ext}, as set out in Formula (92):

$$S_R = \frac{F_{PB} + F_{B,ind} + F_{Ra,st}}{F_D + F_{ext}} \tag{92}$$

where

S_R	is the safety against rolling;
F_{PB}	is the total parking brake force acting at the rail, expressed in N;
$F_{Ra,st}$	is the stationary train resistance force, expressed in N;
$F_{B,ind}$	is the adhesion independent (not related to the wheel to rail contact) force, e.g. force of permanent magnetic track brake, expressed in N;
F_D	is the downhill force due to gravity, expressed in N;
F_{ext}	is the external forces (e.g. wind force), expressed in N;
m_{st}	is the static mass, expressed in kg;
g	is the standard acceleration of gravity, expressed in m/s^2;
α	is the angle of slope (see Figure 15), expressed in degree (°).

6.14.4 Safety against sliding

6.14.4.1 General consideration

At train level, the safety against sliding is the ability of the braked wheelsets to keep the vehicle/unit/train stationary vs. the downhill forces and external forces.

It is necessary to determine the proportion of the mass to be held M for each individual wheelset with applied parking brake in order to compare the required adhesion with the maximum available adhesion.

Starting with the calculation of the forces using Formula (32), which should be transferred into Formulae (37) and (38), the adhesion-dependent (related to the wheel to rail contact) retarding force, $F_{B,\tau,req}$, can be calculated as set out in Formula (93):

$$F_{B,\tau,req} = F_D - F_{B,ind} \pm F_{ext} \tag{93}$$

where

$F_{B,\tau,req}$	is the adhesion-dependent (related to the wheel to rail contact) retarding force, expressed in N;
F_D	is the downhill force due to gravity, expressed in N;
$F_{B,ind}$	is the adhesion independent (not related to the wheel to rail contact) force, e.g. force of permanent magnetic track brake, expressed in N;
F_{ext}	is the external forces (e.g. wind force), expressed in N.

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with

$$F_D = \frac{m_{st} \cdot g \cdot i}{\sqrt{i^2 + 1}} \quad (38)$$

where

F_D	is the downhill force due to gravity, expressed in N;
m_{st}	is the static mass, expressed in kg;
g	is the standard acceleration of gravity, expressed in m/s²;
i	is the gradient of the track.

At wheelset level, the value of the mean adhesion required between wheel/rail for the braked wheelsets, $\overline{\tau_{ax}}$, can be calculated as set out in Formula (94):

$$\overline{\tau_{ax}} = \frac{F_{B,\tau,req}}{F_{Perp}} \quad (94)$$

where

$\overline{\tau_{ax}}$	is the value of the mean adhesion required between wheel/rail for the braked wheelsets;
$F_{B,\tau,req}$	is the adhesion-dependent (related to the wheel to rail contact) retarding force, expressed in N;
F_{Perp}	is the perpendicular force, expressed in N.

with

$$F_{Perp} = F_g \cdot \cos(\alpha) \quad (33)$$

$$F_{Perp} = \frac{m_{st} \cdot g}{\sqrt{i^2 + 1}} \quad (95)$$

where

F_{Perp}	is the perpendicular force, expressed in N;
F_g	is the weight, expressed in N;
α	is the angle of slope, expressed in degree (°);
m_{st}	is the static mass, expressed in kg;
g	is the standard acceleration of gravity, expressed in m/s².

The maximum required adhesion, $\tau_{req,max,ax}$, by each braked wheelset can be calculated as set out in Formula (96):

$$\tau_{req,max,ax} = \frac{F_{B,\tau,i}}{F_{Perp,ax}} \quad (96)$$

where

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$\tau_{req,max,ax}$ is the maximum required adhesion by each braked wheelset;

$F_{B,\tau,i}$ is the adhesion-dependent (related to the wheel to rail contact) retarding force generated by applied parking brake, expressed in N;

$F_{Perp,ax}$ is the perpendicular force, expressed in N.

The coefficient of adhesion, $\tau_{req,st,ax}$, required to resist the downhill and external forces by each braked wheelset can be determined in accordance with Formula (97):

$$\tau_{req,st,ax} = \min\left(\tau_{req,max,ax}; \overline{\tau_{ax}}\right) \quad (97)$$

where

$\tau_{req,st,ax}$ is the coefficient of adhesion required to resist the downhill and external forces by each braked wheelset;

$\tau_{req,max,ax}$ is the maximum required adhesion by each braked wheelset;

$\overline{\tau_{ax}}$ is the value of the mean adhesion required between wheel/rail for the braked wheelsets.

The safety against sliding, $S_{\tau,slide}$, specifies the relationship between the maximum permitted or available static wheel/rail adhesion, τ_{max}, and the coefficient of adhesion, $\tau_{req,st,ax}$, required to resist the downhill and external forces by each braked wheelset and can be calculated as set out in Formula (98):

$$S_{\tau,slide} = \frac{\tau_{max}}{\tau_{req,st,ax}} \quad (98)$$

where

$S_{\tau,slide}$ is the safety against sliding;

τ_{max} is the maximum permitted or available static wheel/rail adhesion;

$\tau_{req,st,ax}$ is the coefficient of adhesion required to resist the downhill and external forces by each braked wheelset.

NOTE The maximum permitted or available static wheel/rail adhesion, τ_{max}, is a given value and the methodology to determine it is outside the scope of this document.

The maximum required adhesion by each braked wheelset, $\tau_{req,max,ax}$, can be calculated as set out in Formula (99):

$$\tau_{req,max,ax} = \frac{F_{PB,ax}}{F_{N,ax}} \quad (99)$$

where

$\tau_{req,max,ax}$ is the maximum required adhesion by each braked wheelset;

$F_{PB,ax}$ is the parking brake force per wheelset acting at the rail, expressed in N;

$F_{N,ax}$ is the static axle load perpendicular to the rail per wheelset with applied parking brake for a specific wheelset, expressed in N.

If no requirements for train resistance or external forces are available and the wheelsets generate the same parking brake forces, the formulae may be simplified.

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The downhill forces to be resisted per wheelset with applied parking brakes, $F_{d,ax}$, can be calculated as set out in Formula (100):

$$F_{d,ax} = \frac{M \cdot g \cdot \sin(\alpha)}{n_{PB,ax}} \tag{100}$$

where

$F_{d,ax}$ is the proportion of downhill force to be resisted per wheelset with applied parking brakes, expressed in N;

M is the mass to be held of the vehicle/unit/train, expressed in kg;

g is the standard acceleration of gravity, expressed in m/s²;

α is the angle of slope (see Figure 21), expressed in degree (°);

$n_{PB,ax}$ is the number of wheelsets with applied parking brake.

If the individual parking brakes generate different forces, then the downhill force needs to be distributed in proportion to the contribution made by each parking brake to prevent sliding.

The available parking brake force per wheelset acting at the rail, $F_{PB,ax}$, shall be greater than the proportion of downhill force, $F_{d,ax}$, as set out in Formula (101):

$$F_{PB,ax} > F_{d,ax} \tag{101}$$

The coefficient of adhesion required to resist the downhill force by each braked wheelset, $\tau_{D,req,ax}$, can be calculated as set out in Formula (102):

$$\tau_{D,req,ax} = \frac{F_{d,ax}}{F_{N,ax}} \tag{102}$$

where

$F_{d,ax}$ is the proportion of downhill force to be resisted per wheelset with applied parking brakes, expressed in N;

$F_{N,ax}$ is the static axle load perpendicular to the rail per wheelset with applied parking brake for a specific wheelset, expressed in N;

$F_{PB,ax}$ is the parking brake force per wheelset acting at the rail, expressed in N;

$\tau_{D,req,ax}$ is the coefficient of adhesion required to resist the downhill force by each braked wheelset.

Safety against sliding, $S_{\tau,slide}$, specifies the relationship between the maximum permitted or available static wheel/rail adhesion, τ_{max}, and the coefficient of adhesion required to resist the downhill force by each braked wheelset, $\tau_{D,req,ax}$, and can be calculated as set out in Formula (103):

$$S_{\tau,slide} = \frac{\tau_{max}}{\tau_{D,req,ax}} \tag{103}$$

where

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$S_{\tau,\text{slide}}$ is the safety against sliding;

$\tau_{\max}$ is the maximum permitted or available static wheel/rail adhesion;

$\tau_{\text{D,req,ax}}$ is the coefficient of adhesion required to resist the downhill force by each braked wheelset.

The maximum required adhesion by each braked wheelset, $\tau_{\text{req,max,ax}}$, can be calculated as set out in Formula (104):

$$\tau_{\text{req,max,ax}} = \frac{F_{\text{PB,ax}}}{F_{\text{N,ax}}} \tag{104}$$

where

$F_{\text{N,ax}}$ is the static axle load perpendicular to the rail per wheelset with applied parking brake for a specific wheelset, expressed in N;

$F_{\text{PB,ax}}$ is the parking brake force per wheelset acting at the rail, expressed in N.

6.14.4.2 General process

More complex methods are required (e.g the method described in this subclause and shown in B.3) if there are significant differences in the distribution of the parking brake forces acting on the wheels or in the distribution of the overall static mass (e.g. unbraked wheelsets) over all wheelsets of the considered composition (vehicle/unit/train).

In order to hold a vehicle/unit/train safely on a slope, a value of the mean adhesion required between wheel/rail for the braked wheelsets, $\overline{\tau}_{\text{ax}}$, as calculated in Formula (89), is necessary. If all installed brake systems are able to guarantee a maximum required adhesion level higher than $\overline{\tau}_{\text{ax},i}$ at each wheelset (i is an index used for sorting wheelsets), then, as a conclusion, sufficient braking force has been installed. Automatically, the level of the coefficient of adhesion required to resist the downhill force by each braked wheelset, $\tau_{\text{D,req,ax}}$, is equal to the value of the mean adhesion required between wheel/rail for the braked wheelsets, $\overline{\tau}_{\text{ax}}$.

However, if, for one or several wheelset(s), the braking force of a brake system leads to a maximum required adhesion level less than the value of the mean adhesion required between wheel/rail for the braked wheelsets, $\overline{\tau}_{\text{ax}}$, then some amount of mass shall be taken over by other brake systems and wheelsets respectively. To identify those wheelsets, the results of the maximum required adhesion by each braked wheelset, $\tau_{\text{req,max,ax}}$, are sorted in ascending order.

In the next step, an iterative procedure shall be triggered and the following assessment starts with the wheelset requiring the lowest level of adhesion and ends with the wheelset requiring the highest level of adhesion (see Figure B.2), starting with $\overline{\tau}_{\text{ax},i} = \overline{\tau}_{\text{ax}}$.

The coefficient of adhesion required to resist the downhill force by each braked wheelset, $\tau_{\text{D,req,ax}}$, is limited (for the time being) to the temporary value of the mean adhesion required between wheel/rail for the braked wheelsets used during iteration step, i, and can be calculated as set out in Formula (105). To be clear, if $\tau_{\text{req,max,ax}}$ is greater than $\overline{\tau}_{\text{ax},i}$ at all wheelsets, then the coefficient of adhesion required to resist the downhill force by each braked wheelset, $\tau_{\text{D,req,ax}}$, is $\overline{\tau}_{\text{ax},i}$ and the iteration can be stopped.

$$\tau_{\text{D,req,ax}} = \min\left[\left(\tau_{\text{req,max,ax}}\right) \text{or} \left(\overline{\tau}_{\text{ax},i}\right)\right] \tag{105}$$

If the maximum required adhesion by each braked wheelset, $\tau_{\text{req,max,ax}}$, of the first wheelset is less than $\overline{\tau}_{\text{ax},i}$, then $\tau_{\text{D,req,ax}}$ is equal to $\tau_{\text{req,max,ax}}$ for this wheelset.

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The remaining force of the mass to be held, $F_{B,\tau,req,rem}$, can be recalculated as set out in Formula (106):

$$F_{B,\tau,req,rem} = F_{B,\tau,req} - \tau_{D,req,ax} \cdot F_N \tag{106}$$

Following the steps of the iterative procedure in the flowchart of B.3, the remaining static axle load perpendicular to the rail, $F_{N,rem}$, can be recalculated as set out in Formula (107):

$$F_{N,rem} = F_N - F_{N,ax} \tag{107}$$

To determine a new level of $\bar{\tau}_{ax,i+1}$ for the wheelsets which had a maximum required adhesion level larger than $\bar{\tau}_{ax,i}$, the remaining force of the mass to be held, $F_{B,\tau,req,rem}$, shall be distributed uniformly to all remaining wheelsets of the vehicle/unit/train and can be recalculated as set out in Formula (106).

The iteration steps in Formulae (106) and (107) shall be repeated for each axle in order to shift the mass to be held virtually to other wheelset as set out in Formula (106) and as shown in Figure B.2.

The train is considered to be moving either:

— the maximum required adhesion by each braked wheelset, $\tau_{req,max,ax}$, at all wheelsets is smaller than the initial value of the mean adhesion required between wheel/rail for the braked wheelset, $\overline{\tau_{ax}}$;

— there is at least one wheelset with a coefficient of adhesion required to resist the downhill force by each braked wheelset, $\tau_{D,req,ax}$, greater than the maximum permitted or available static wheel/rail adhesion, τ_{max}.

where

$\overline{\tau_{ax}}$	is the value of the mean adhesion required between wheel/rail for the braked wheelset;
$\bar{\tau}_{ax,i}$	is the temporary value of the mean adhesion required between wheel/rail for the braked wheelset used during iteration step i;
$\tau_{req,max,ax}$	is the maximum required adhesion by each braked wheelset;
τ_{max}	is the maximum permitted or available static wheel/rail adhesion;
$\tau_{D,req,ax}$	is the coefficient of adhesion required to resist the downhill force by each braked wheelset;
F_N	is the static axle load perpendicular to the rail per wheelset with applied parking brake, expressed in N;
$F_{N,ax}$	is the static axle load perpendicular to the rail per wheelset with applied parking brake for a specific wheelset, expressed in N;
$F_{N,rem}$	is the remaining static axle load perpendicular to the rail, expressed in N;
$F_{B,\tau,req}$	is the adhesion-dependent (related to the wheel to rail contact) retarding force, expressed in N;
$F_{B,\tau,req,rem}$	is the remaining force of the mass to be held, expressed in N.

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6.14.5 Retention force

As a minimum, the sum of the adhesion-dependent (related to the wheel to rail contact) retarding forces, $F_{B,\tau,req}$, shall be greater than the downslope gravitational force and externally action force, F_{ext}, to prevent the train from moving and can be calculated as set out in Formula (108):

$$F_{B,\tau,req} > F_D - F_{ext} - F_{B,ind} - F_{Ra} \tag{108}$$

where

$F_{B,\tau,req}$ is the adhesion-dependent (related to the wheel to rail contact) retarding force, expressed in N;

F_D is the downhill force due to gravity, expressed in N;

F_{ext} is the external force (e.g. wind force), expressed in N;

$F_{B,ind}$ is the adhesion independent (not related to the wheel to rail contact) force, e.g. force of permanent magnetic track brake, expressed in N;

F_{Ra} is the instantaneous retarding force by train resistance, expressed in N.

For brake systems that do not depend on adhesion, the entire braking force is considered. In case of brake systems depending on the wheel/rail adhesion, only the minimum transferable force is considered. In such cases, additional safety factors, S_R (safety against rolling) and $S_{\tau,slide}$ (safety against sliding), can be specified as safety margins for the retention force, F_H, calculated as set out in Formula (109):

$$F_H = \sum_z \frac{F_{B,ind,z}}{S_R} + \sum_i \min\left(\frac{F_{N,i} \cdot \tau_{max}}{S_{\tau,slide}}; \frac{F_{B,\tau,i}}{S_R}\right) \tag{109}$$

where

F_H is the retention force, expressed in N;

$F_{B,ind,z}$ is the adhesion independent (not related to the wheel to rail contact) retarding force per type of equipment, expressed in N;

$F_{N,i}$ is the static axle load perpendicular to the rail per wheelset with applied parking brake (i is an index used for sorting wheelsets), expressed in N;

τ_{max} is the maximum permitted or available static wheel/rail adhesion;

$S_{\tau,slide}$ is the safety against sliding;

S_R is the safety against rolling;

$F_{B,\tau,i}$ is the adhesion-dependent (related to the wheel to rail contact) retarding force generated by applied parking brake (i is an index used for sorting wheelsets), expressed in N.

This approach is based on the assumption of the maximum theoretical adhesion, which does not always correspond to the adhesion that really acts on the train. To obtain a realistic value, the adhesion actually used can be calculated (see B.3).

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6.14.6 Retention Safety

The retention safety, S_H, corresponds to the relationship between retention force, F_H, and external forces, F_{ext}, and can be calculated as set out in Formula (110), with F_D as set out in Formula (37):

$$S_H = \frac{F_H}{\left|F_D + F_{ext}\right|} \quad (110)$$

where

S_H is the retention safety;

F_H is the retention force, expressed in N;

F_D is the downhill force due to gravity, expressed in N;

F_{ext} is the external forces (e.g. wind force), expressed in N.

As a minimum, the safety level, $S_{H,min}$, shall be reached. The result is defined for the whole unit.

In the static case, no difference will be made between downhill and uphill gradient. External forces, F_{ext}, can reduce or raise the effect of the downhill force due to gravity, F_D.

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Annex A
(informative)

Methodology of stopping and slowing distance calculation

The calculation procedure is summarized in Figures A.1 and A.2 and assumes that the entity considered for the friction brake is an individual wheelset.

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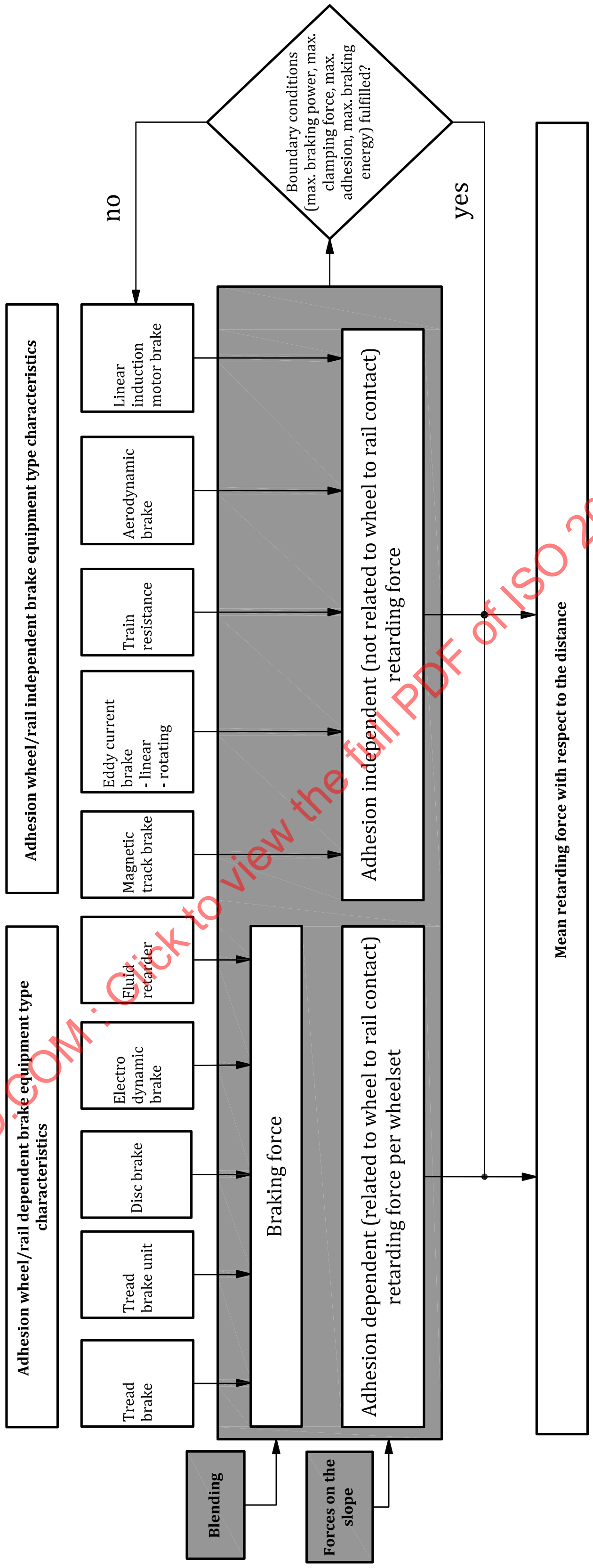

Figure A.1 — Methodology of stopping and slowing distance calculation (Part 1)

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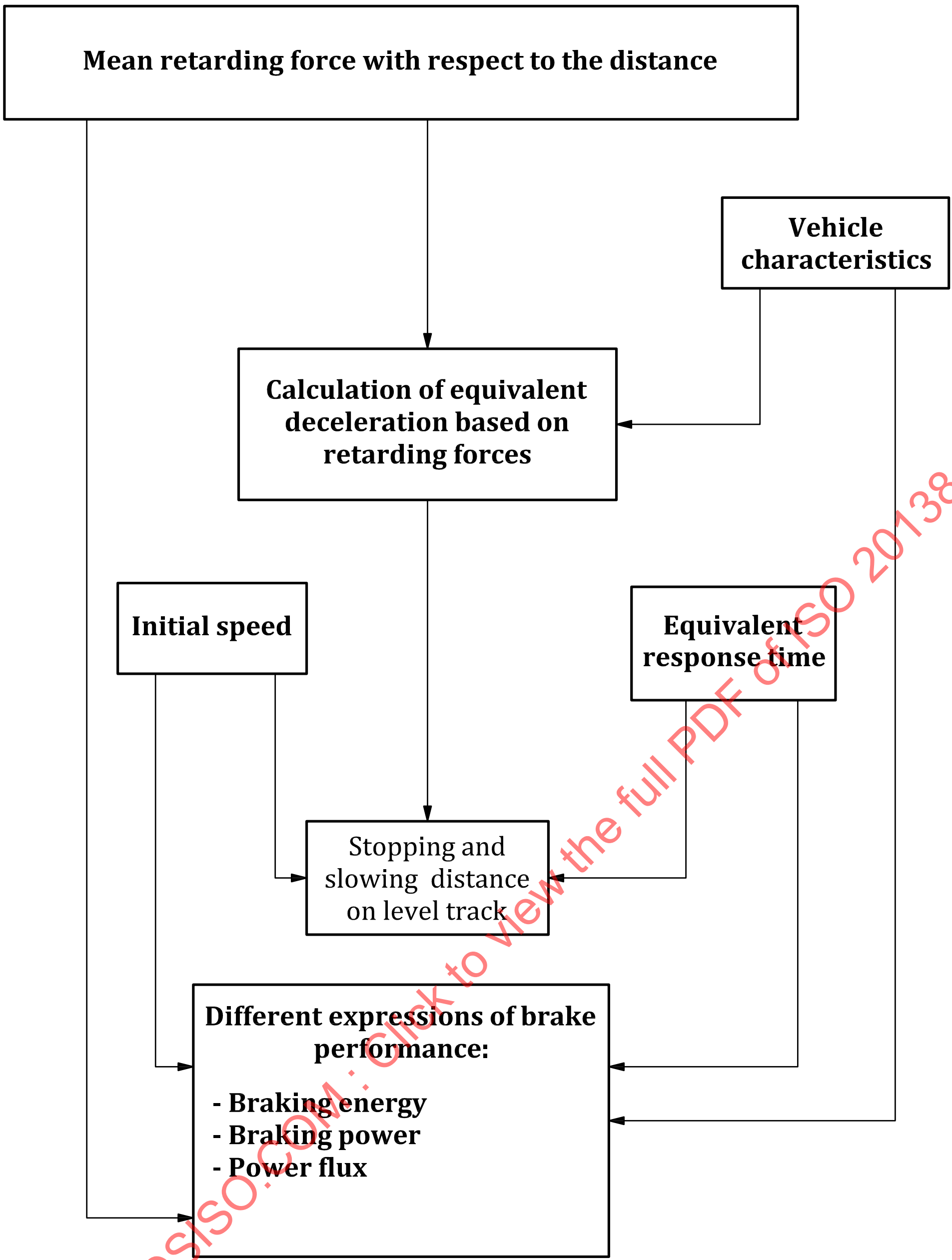

Figure A.2 — Methodology of stopping and slowing distance calculation (Part 2)

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Annex B
(informative)

Workflow for stationary brake calculations

B.1 General

The calculation procedure is summarized in Figures B.1 and B.2 and assumes that the entity considered for the friction brake is an individual wheelset.

B.2 Methodology for stationary brake calculation

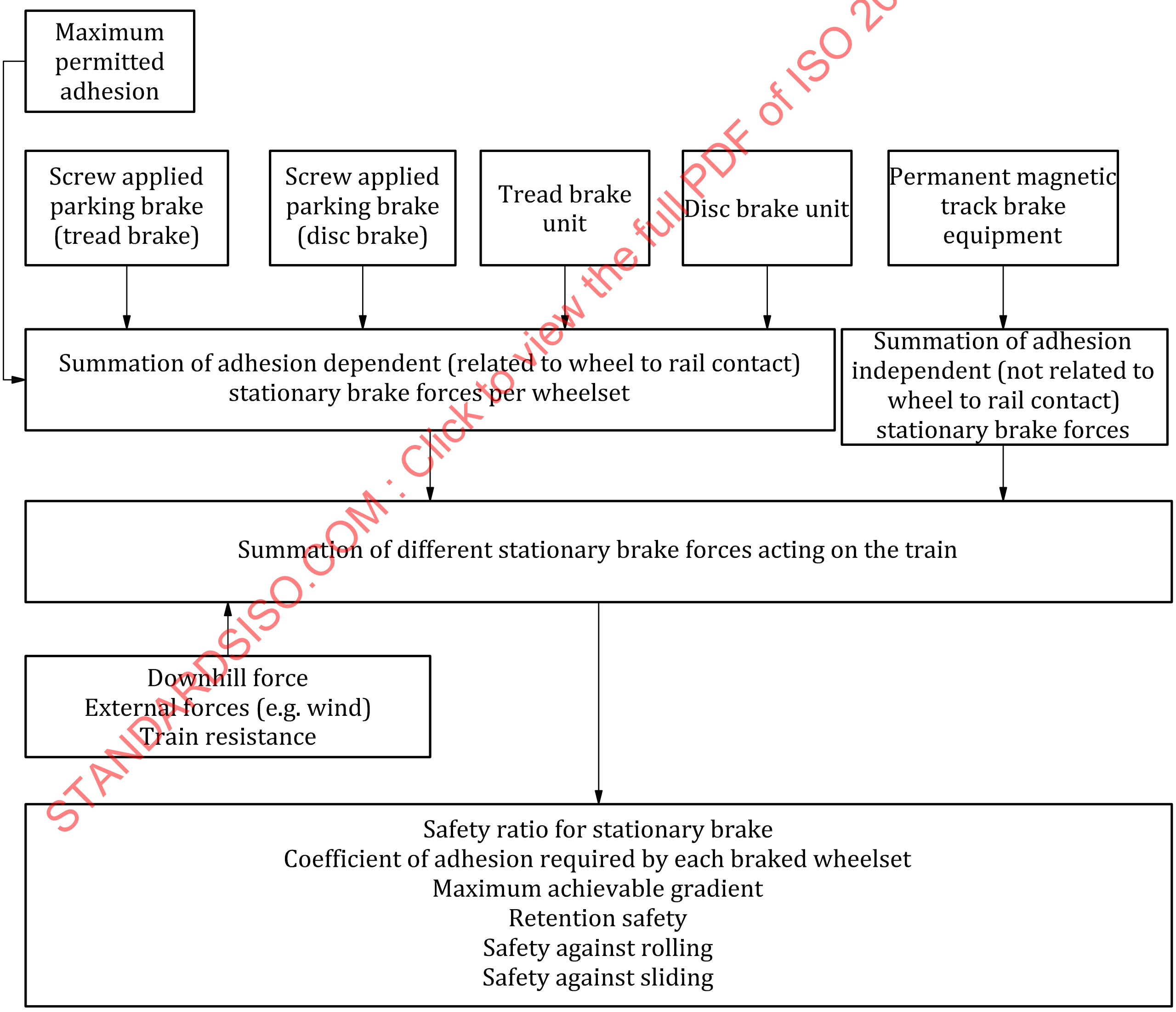

Figure B.1 — Workflow for stationary brake calculation

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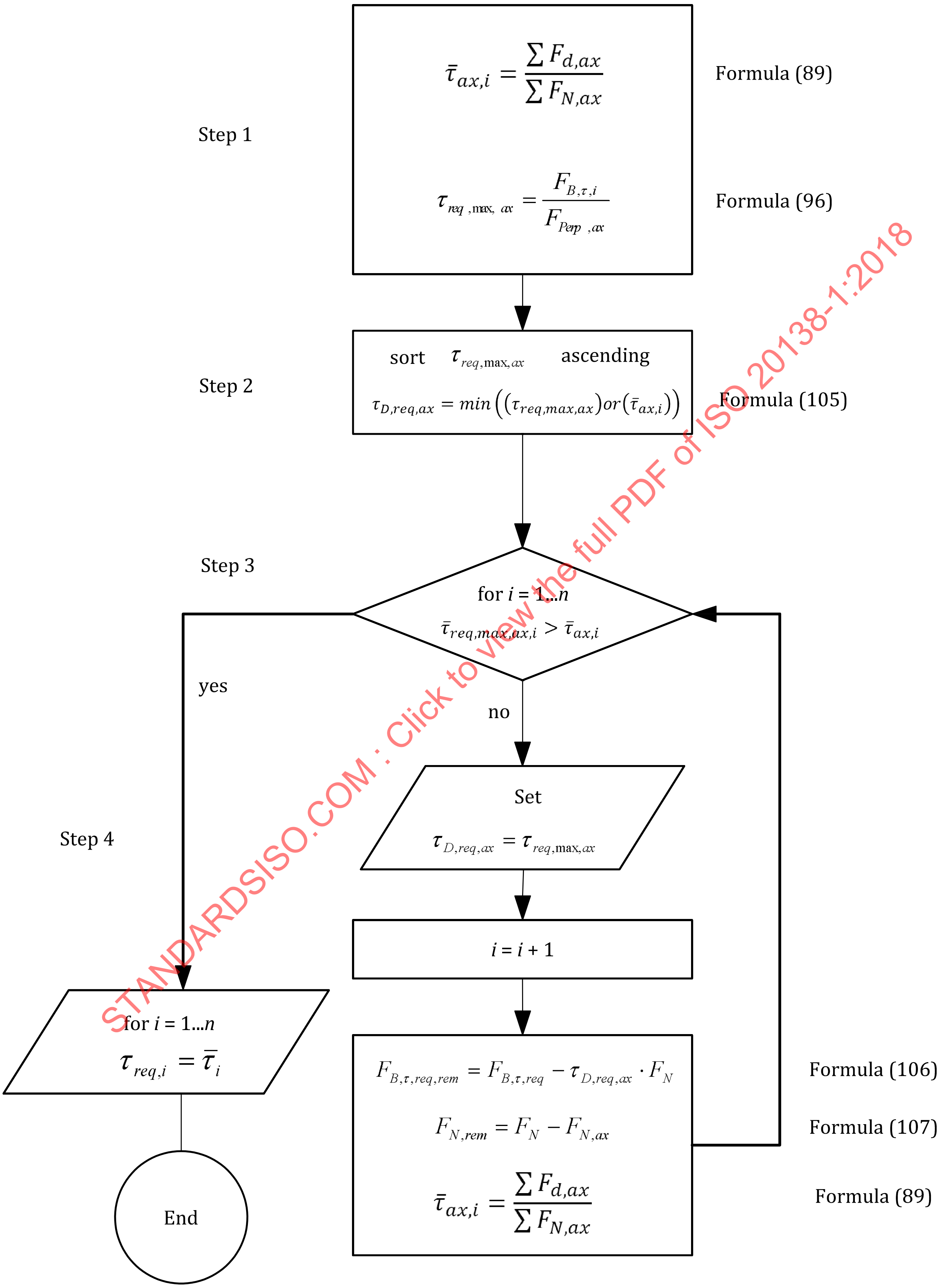
B.3 Workflow for the retention calculation



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Key

$F_{d,ax}$	proportion of downhill force to be resisted per wheelset with applied parking brakes, expressed in N
$\overline{\tau}_{ax}$	value of the mean adhesion required between wheel/rail for the braked wheelset
$\tau_{req,max,ax}$	maximum required adhesion by each braked wheelset
$\tau_{D,req,ax}$	coefficient of adhesion required to resist the downhill force by each braked wheelset
F_{Perp}	perpendicular force, expressed in N
F_{N}	static axle load perpendicular to the rail per wheelset with applied parking brake, expressed in N
$F_{N,ax}$	static axle load perpendicular to the rail per wheelset with applied parking brake for a specific wheelset, expressed in N
$F_{N,rem}$	remaining static axle load perpendicular to the rail, expressed in N
$F_{B,\tau,req}$	adhesion-dependent (related to the wheel to rail contact) retarding force, expressed in N
$F_{B,\tau,req,rem}$	remaining force of the mass to be held, expressed in N
$F_{B,\tau,i}$	adhesion dependent (related to the wheel to rail contact) retarding force generated by applied parking brake (*i* is an index used for sorting wheelsets), expressed in N

Figure B.2 — Workflow for the retention calculation

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Annex C
(informative)

Examples for brake calculation

C.1 Freight wagon with two bogies/four wheelsets/central brake rigging with one brake cylinder and 16 brake blocks

C.1.1 General

See Figure C.1.

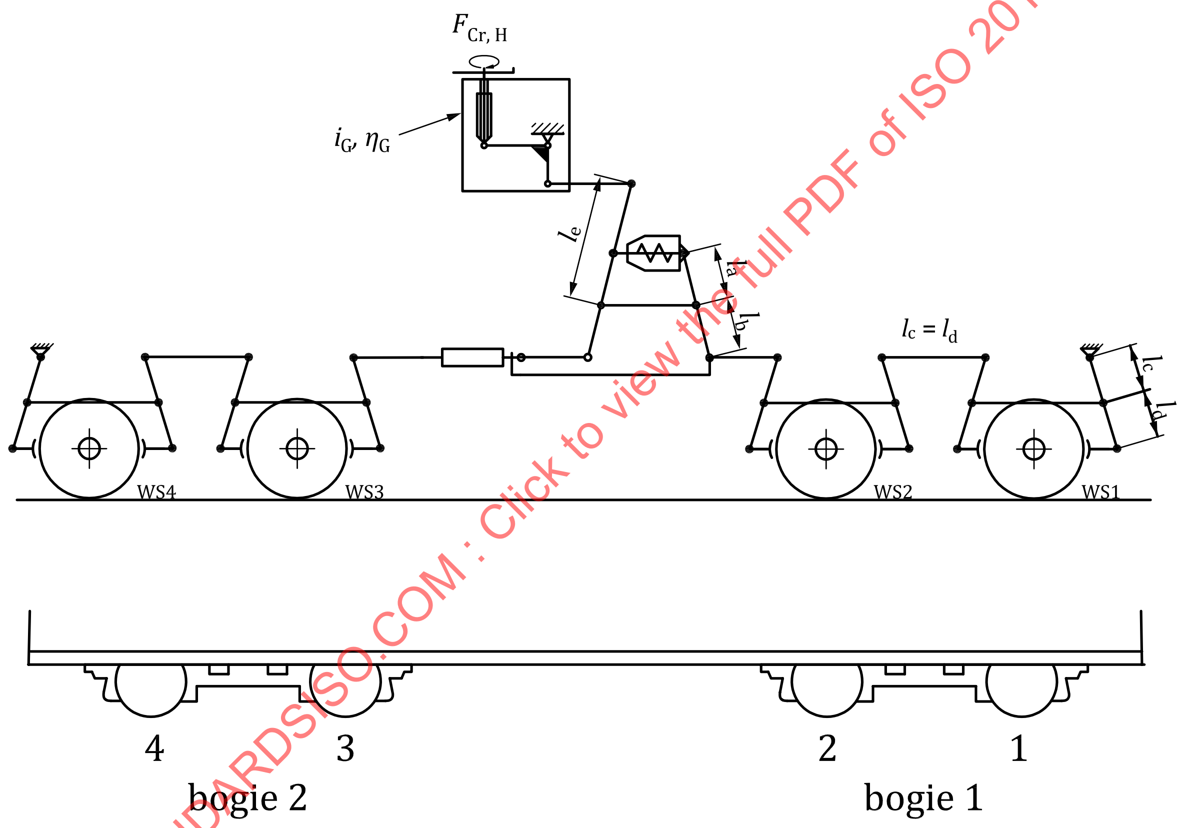

Figure C.1 — Freight wagon with four wheelsets

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Table C.1 — Brake equipment type

Bogie no.	Wheelset	Brake equipment types and clause reference
1	1 and 2	Main brake system: Central brake rigging with composite brake blocks, tread brake (see 5.3.2) Stationary brake: Screw applied hand brake acting on all wheels (see 6.7.1)
2	3 and 4	

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C.1.2 Example for brake calculation of a freight wagon

Table C.2 — Freight wagon — Input data

Bogie no.	Description	Symbol	Example value	Unit
	Main Brake System			
—	Brake cylinder area	A_c	706,9·10^{-4}	m^2
—	Brake cylinder pressure	p_c	0,38	MPa
—	Restoring force of brake unit	$F_{S,C}$	1 500	N
—	Internal efficiency of brake unit	η_c	1	—
—	Internal rigging ratio of brake unit	i_c	1	—
1 and 2	Main brake lever length	l_a	0,515	m
1 and 2	Main brake lever length	l_b	0,325	m
1 and 2	Rigging ratio	i_{rig}	1,584	—
1 and 2	Lever ratio per brake beam, $l_c = l_d$	$i_{rig,ax,n}$	1	—
1 and 2	Overall efficiency of brake rigging	η_R	0,83	—
1 and 2	Restoring force (slack adjuster)	$F_{S,R}$	2 000	N
1 and 2	Mean friction coefficient of brake block	μ_m	0,2	—
1 and 2	Contact area per brake block	A_B	256·10^{-4}	m^2
1 and 2	Dynamic mass of the vehicle (=m_{st} + m_{rot}) with m_{rot} = 0	m_{dyn}	90 000	kg
—	Final speed	v_{fin}	0	m/s
—	Initial speed (100 km/h)	v_0	27,78	m/s
—	Equivalent build-up time	t_e	2	s
	Stationary brake (screw hand brake)			
—	Crank handle force	$F_{Cr,H}$	500	N
—	Gear efficiency	η_G	0,19	—
—	Gear ratio	i_G	236	—
1 and 2	Number of wheelsets with applied parking brake	$n_{PB,ax}$	4	—
1 and 2	Main brake lever length (parking brake)	l_e	0,665	m
1 and 2	Static friction coefficient of brake block	μ_{st}	0,2	—
1 and 2	Overall static efficiency of brake rigging	$\eta_{R,st}$	0,75	—
—	Angle of slope	α	2	°
—	External forces	F_{ext}	0	N
—	Wind force on the train	F_{wind}	0	N
1 and 2	Stationary train resistance force	$F_{Ra,st}$	0	N
1 and 2	Adhesion independent (not related to the wheel to rail contact) force	$F_{B,ind}$	0	N
1 and 2	Mass to be held of the vehicle/unit/train	M	90 000	kg
1 and 2	Maximum permitted wheel/rail adhesion	τ_{max}	0,12	—

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Table C.3 — Freight wagon — Examples of calculation

Formula no.	Title	Formula	Result value
Main brake system (central brake rigging with composite brake block)			
(3)	Internal cylinder force	$F_c = p_c \cdot A_c$ $F_c = 3{,}8 \cdot 10^5\,\text{Pa} \cdot 706{,}9 \cdot 10^{-4}\,\text{m}^2$	26 862 N
(4)	Piston force	$F_P = k1_V \cdot \lvert p_c \cdot A_c \cdot \eta_c \cdot i_c \rvert + k2_V \cdot \lvert F_{S,C} \rvert$ $F_P = k1_V \cdot \lvert F_c \cdot \eta_c \cdot i_c \rvert + k2_V \cdot \lvert F_{S,C} \rvert$ $F_P = \lvert F_c \cdot \eta_c \cdot i_c \rvert - \lvert F_{S,C} \rvert$ $F_P = 26\,862\,\text{N} \cdot 1 \cdot 1 - 1\,500\,\text{N}$	25 362 N
(5)	Force on application point bogie	$F_{pull} = F_P \cdot i_{rig} - F_{S,R}$ $F_{pull} = 25\,362\,\text{N} \cdot 1{,}584 - 2\,000\,\text{N}$	38 173 N
(6)	Single brake block force	$F_{b,ax} = \frac{1}{2} \cdot F_{pull} \cdot i_{rig,ax,n}$ $F_{b,ax} = \frac{1}{2} \cdot 38\,173\,\text{N} \cdot 1$	19 087 N
(9)	Total brake block force	$F_{b,tot} = \sum F_{b,ax} \cdot \eta_R$ $F_{b,tot} = 16 \cdot F_{b,ax} \cdot \eta_R$ $F_{b,tot} = 16 \cdot 19\,087\,\text{N} \cdot 0{,}83$	253 476 N
(11)	Mean retarding force acting at the rail generated by the brake equipment	$\bar{F}_r = F_{b,tot} \cdot \mu_m$ $\bar{F}_r = 253\,476\,\text{N} \cdot 0{,}2$	50 695 N
(10)	Specific pressure per brake block	$p_{ab} = \frac{F_{b,ax}}{A_b}$ $p_{ab} = \frac{19\,087\,\text{N}}{256 \times 10^{-4}\,\text{m}^2}$	745 586 N/m²
(46)	Equivalent deceleration	$a_e = \frac{\sum_{n=1}^{N} \bar{F}_{r,n} + \sum \bar{F}_{ext}}{m_{dyn}}$ $a_e = \frac{50\,695\,\text{N} + 0\,\text{N}}{90\,000\,\text{kg}}$	0,56 m/s²

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Table C.3 *(continued)*

Formula no.	Title	Formula	Result value
(48)	Stopping distance	$s = v_0 \cdot t_e + \frac{v_0^2 - v_{fin}^2}{2 \cdot a_e}$ $s = 27{,}78\frac{m}{s} \cdot 2\,s + \frac{\left(27{,}78\frac{m}{s}\right)^2}{2 \cdot 0{,}56\frac{m}{s^2}}$	745 m
Stationary (screw hand brake)			
(57)	Output force of parking brake mechanism	$F_G = F_{Cr,H} \cdot i_G \cdot \eta_G$ $F_G = 500\,N \cdot 236 \cdot 0{,}19$	22 420 N
(58)	Static force on application point bogie	$F_{pull,st} = F_G \cdot \frac{l_e}{l_b} - F_{S,C} \cdot i_{rig} - F_{S,R}$ $F_{pull,st} = 22\,420\,N \cdot \frac{0{,}665}{0{,}325} - 1\,500\,N \cdot 1{,}584 - 2\,000\,N$	41 499 N
(59)	Static single brake block force	$F_{b,ax,st} = \frac{1}{2} \cdot F_{pull,st} \cdot i_{rig.ax,i}$ $F_{b,ax,st} = \frac{1}{2} \cdot 41\,499\,N \cdot 1$	20 750 N
(62)	Static total brake block force	$F_{b,tot,st} = \sum F_{b,ax,st} \cdot \eta_{R,st}$ $F_{b,tot,st} = 16 \cdot 20\,750\,N \cdot 0{,}75$	249 000 N
(63)	Total stationary brake force acting at the rail	$F_{B,st} = F_{b,tot,st} \cdot \mu_{st}$ $F_{B,st} = 249\,000\,N \cdot 0{,}2$	49 800 N
(91)	Downhill force due to gravity	$F_D = m_{st} \cdot g \cdot \sin(\alpha)$ $F_D = 90\,000\,kg \cdot 9{,}81\frac{m}{s^2} \cdot \sin(2°)$	30 812 N
(92)	Safety against rolling	$S_R = \frac{F_{PB} + F_{B,ind} + F_{Ra,st}}{F_D + F_{ext}}$ $S_R = \frac{49\,800\,N + 0\,N + 0\,N}{30\,812\,N + 0\,N}$	1,6
(100)	Proportion of downhill force to be resisted by the wheelsets with applied parking brakes	$F_{d,ax} = \frac{M \cdot g \cdot \sin(\alpha)}{n_{PB,ax}}$ $F_{d,ax} = \frac{90\,000\,kg \cdot 9{,}81\frac{m}{s^2} \cdot \sin(2°)}{4}$	7 703 N
(88)	Static axle load	$F_{g,ax} = m_{st,ax} \cdot g$ $F_{g,ax} = 22\,500\,kg \cdot 9{,}81\frac{m}{s^2}$	220 725 N

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Table C.3 *(continued)*

Formula no.	Title	Formula	Result value
(87)	Static axle load perpendicular to the rail per wheelset with applied parking brake	$F_{N,ax} = F_{g,ax} \cdot \cos(\alpha)$ $F_{N,ax} = 220\,725\ \text{N} \cdot \cos(2°)$	220 590 N
(102)	Coefficient of adhesion required to resist the downhill force by each braked wheelset	$\tau_{D,req,ax} = \frac{F_{d,ax}}{F_{N,ax}}$ $\tau_{D,req,ax} = \frac{7\,703\ \text{N}}{220\,590\ \text{N}}$	0,035
(103)	Safety against sliding	$S_{\tau,slide} = \frac{\tau_{max}}{\tau_{D,req,ax}}$ $S_{\tau,slide} = \frac{0,12}{0,035}$	3,5
(84)	Maximum achievable gradient for sliding	$i_{max,slide} = \frac{1}{\sqrt{\left(\frac{m_{st} \cdot g}{F_{PB} - F_{wind}}\right)^2 - 1}}$ $i_{max,slide} = \frac{1}{\sqrt{\left(\frac{90\,000\ \text{kg} \cdot 9,81\frac{\text{m}}{\text{s}^2}}{49\,800\ \text{N} - 0\ \text{N}}\right)^2 - 1}}$	0,05
(85)	Maximum achievable gradient for rolling	$i_{max,roll} = \frac{1}{\sqrt{\left(\frac{m_{st} \cdot g}{F_{PB} + F_{Ra,st} - F_{wind}}\right)^2 - 1}}$ $i_{max,roll} = \frac{1}{\sqrt{\left(\frac{90\,000\ \text{kg} \cdot 9,81\frac{\text{m}}{\text{s}^2}}{49\,800\ \text{N} + 0\ \text{N} - 0\ \text{N}}\right)^2 - 1}}$	0,05

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C.2 Coach with two bogies/four wheelsets

C.2.1 General

See Figure C.2.

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