BS EN 14531-1:2015

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Railway applications — Methods for calculation of stopping and slowing distances and immobilization braking

Part 1: General algorithms utilizing mean value calculation for train sets or single vehicles

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National foreword

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The UK participation in its preparation was entrusted to Technical Committee RAE/4/-/1, Railway applications - Braking.

A list of organizations represented on this committee can be obtained on request to its secretary.

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Bahnanwendungen - Verfahren zur Berechnung der Anhalte- und Verzögerungsbremswege und der Feststellbremsung - Teil 1: Allgemeine Algorithmen für Einzelfahrzeuge und Fahrzeugverbände unter Berücksichtigung von Durchschnittswerten

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European foreword

This document (EN 14531-1:2015) has been prepared by Technical Committee CEN/TC 256 “Railway applications”, the secretariat of which is held by DIN.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by June 2016, and conflicting national standards shall be withdrawn at the latest by June 2016.

This document supersedes EN 14531-1:2005.

This document has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association, and supports essential requirements of EU Directive 2008/57/EC.

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

This series of European standards EN 14531, *Railway applications — Methods for calculation of stopping and slowing distances and immobilization braking* consists of:

— *Part 1: General algorithms utilizing mean value calculation for train sets or single vehicles*;

— *Part 2: Step-by-step calculations for train sets or single vehicles*.

The two parts are interrelated and should be considered together when conducting the step-by-step calculation of stopping and slowing distances.

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

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

Introduction

This European Standard describes a common calculation method for railway applications. It describes the general algorithms utilizing mean value calculation for use in the design and validation of brake equipment and braking performance for all types of train sets and single vehicles. In addition the algorithms provide a means of comparing the results of other braking performance calculation methods.

EN 14531 was originally planned to have six parts covering the calculation methodology to be used when conducting calculations relating to the braking performance of various types of railway vehicles under the heading EN 14531, Railway applications – Methods for calculation of stopping, slowing distances and immobilization braking. The six parts were as follows:

— Part 1: General algorithms

— Part 2: Application to single freight wagon

— Part 3: Application to mass transit (LRV's and D- and E- MU's)

— Part 4: Application to single passengers coach

— Part 5: Application to locomotive

— Part 6: Application to high speed trains

EN 14531-1 was originally published in 2005 followed by EN 14531-6 which was published in 2009.

Following the above it was decided that a common methodology could be used for Parts 2 to 5 and this should be contained under a revised version of Part 1 with a title of *Railway applications — Methods for calculation of stopping and slowing distances and immobilisation braking — Part 1: General algorithms utilizing mean value calculation for train sets or single vehicles* while revising Part 6 to be Part 2 with the title of *Railway applications - Methods for calculation of stopping and slowing distances and immobilization braking - Part 2: Step by step calculations for train sets or single vehicles*.

EN 14531-1:2005 and EN 14531-6:2009 are referenced in the current technical specifications for interoperability (TSIs) (Freight wagons and locomotive and passenger rolling stock (RST)). The tables of the Annex ZA give the equivalence of the TSI referenced clauses of the original EN 14531 series to the clauses of this issue of EN 14531-1 and EN 14531-2.

1 Scope

This European Standard describes general algorithms for the brake performance calculations to be used for all types of train sets, units or single vehicles, including high speed, locomotive and passenger coaches, conventional vehicles and wagons.

This European Standard does not specify the performance requirements. It enables the estimation and/or comparison by calculation of the various aspects of the performance: stopping or slowing distances, dissipated energy, power, force calculations and immobilization braking.

If it is required to validate, verify or assess braking performance it is recommended that a more detailed calculation is performed in accordance with EN 14531-2, i.e. a step by step calculation.

This European Standard contains generic examples of the calculation of brake forces for individual brake equipment types and calculation of stopping distance and immobilization braking relevant to a train (see Annexes C and D).

2 Normative references

The following referenced documents, in whole or in part, are normatively referenced in this document and are indispensable for its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

EN 14067-4, *Railway applications - Aerodynamics - Part 4: Requirements and test procedures for aerodynamics on open track*

EN 14478, *Railway applications - Braking - Generic vocabulary*

EN 14531-2, *Railway applications – Methods for calculation of stopping and slowing distances and immobilisation braking – Part 2: Step by step calculations for trains or single vehicles*

prEN 15328, *Railway applications - Braking - Brake pads*

EN 16452, *Railway applications – Braking – Brake blocks*

EN 15663, *Railway applications - Definition of vehicle reference masses*

3 Terms, definitions, symbols and indices

3.1 Terms and definitions

For the purpose of this document, the terms and definitions given in EN 14478 and EN 14531-2 and the following apply.

3.1.1
static mass per axle
mass measured by weighing at the wheel-rail interface, or estimated from design evaluation, of each axle in a stationary condition for each operating condition required

3.1.2
static mass of the train
summation of all the static mass values per entity

Note 1 to entry: E.g. per axle, for each operating condition.

3.1.3
equivalent rotating mass
linear conversion of the moment of inertia due to rotating parts coupled to the wheelsets during braking into an equivalent additional static mass

Note 1 to entry: This includes brake discs, gear wheels etc.

3.1.4
brake equipment type
group of equipment that provide braking force

Note 1 to entry: When brake equipment is used on one part of the train under certain conditions and used on another part of the same train under other conditions, two different brake equipment types shall be considered.

3.1.5
tread brake unit/disc brake unit
functional unit from which brake force is delivered, typically consisting of a brake cylinder, slack adjuster portion and all associated component parts

Note 1 to entry: Sometimes referred to as tread/disc brake actuator.

3.1.6
isolated brake equipment
equipment not considered in the calculation due to assumed isolation

Note 1 to entry: E.g. brake equipment of a bogie.

3.1.7
active brake equipment
equipment considered to be operational in the calculation of a specific brake equipment type

3.1.8
mean value calculation
calculation method in which the values used for each active brake equipment type are a mean value based on speed, force or distance as applicable for a particular speed range

3.1.9
decelerating force
force resulting from summation of all forces acting contrary to the direction of movement when considering a train

Note 1 to entry: Each operational brake equipment type produces its own decelerating force which when added to the additional external forces opposing motion results in the total decelerating force of the train.

Note 2 to entry: For the purpose of this standard a decelerating force is considered as a positive value, therefore accelerating force is considered as a negative value.

3.1.10
braking force
force produced by the active brake equipment types to brake the train

Note 1 to entry: It does not include external forces which contribute to the overall decelerating force of the vehicle or train.

3.1.11
external forces
forces typically including rolling resistance, gradient, head wind etc

3.1.12
entity
group or item considered in a calculation

Note 1 to entry: E.g. train, vehicle, bogie, axle, wheel.

3.2 Symbols and indices

For the purposes of this document, the general symbols given in Table 1 and indices given in Table 2 apply.

NOTE Specific symbols and indices are defined in the relevant clauses.

Table 1 — Symbols

Symbol	Definition	Unit
A	area	m^2
a	deceleration	m/s^2
A_s	area of a friction surface swept by the friction elements	m^2
B	braked weight	T
D	wheel diameter	m
F	force	N
F_B	braking force related to the rail	N
F_g	downhill force on the train	N
g_n	standard acceleration of free fall = 9,80665 m/s^2 (see: ISO 80000-3)	m/s^2
i	gradient (rising gradient is positive; e.g. for a gradient of 5 ‰, i = 0,005)	-
i_c	cylinder ratio	-
i_{rig}	rigging ratio	-
i_{tra}	transmission ratio	-
J	inertia due to rotation of masses	$kg \cdot m^2$
m	mass	kg
n	quantity	-
P	power	W
p	pressure	Pa
r	radius	m
s	distance	m

S	safety factor	-
t	time	s
τ	coefficient of adhesion	-
v	speed	m/s
W	energy	J
W_S	energy per unit area	$\frac{J}{m^2}$
λ	braked weight percentage	-
μ	coefficient of friction	-
η	efficiency	-

Table 2 — General indices

Indices	Term
AMG	attraction force for a magnetic track
ap	application point
ax	axle
a	available
B	braking force
BEC	braking force for an eddy current brake
BED	electro-dynamic braking force
BFR	fluid retarder braking force
BMG	braking force for a magnetic track brake
b	block or pad
bog	bogie
C	cylinder
act	unit/actuator
cha	characteristic
Bd	braking force demanded
disc	disc
dyn	dynamic
e	equivalent
ent	entity
ext	external
fin	final state
H	hand brake
i	brake equipment type
j	range or step
im	immobilization, parking, holding
ind	independent of adhesion
nt	internal
inst	instantaneous
max	maximum
min	minimum
mot	motor
MG	magnetic track brake
n	normal direction
R	response
Ra	train resistance to motion
req	required
rig	rigging
rot	rotating
R1	return spring
R2	regulator
S	spring
st	static
T	tangential direction
tot	total
tra	transmission

wind	wind
0	initial state
1, 2, 3, 4 –- etc.	intermediate state

4 Stopping and slowing distances calculation

4.1 General

The principle of the algorithm flow is presented in Annex A, Figure A.1.

In general the formulae contained in this clause are used in the first instance when considering constant brake forces with respect to speed.

In the second instance the formulae may be used as a mean value calculation when considering a non constant speed dependent brake force which is transformed to a mean brake force value. This mean value of brake force is considered as a fully developed force without considering the response time and results in the same braking distance as if calculated using the speed dependent brake force. See Annex E.

The algorithms in this standard use mean values, however if it is necessary to use instantaneous values and algorithms using finite time steps then EN 14531-2 shall be used.

4.2 Accuracy of input values

The accuracy of the calculation described here depends directly on the accuracy of the input data.

The accuracy of the input data values shall be relevant to the purpose of the calculation and shall be traceable as to how these values were established e.g. engineer's estimation, test results, manufacturer’s data etc. Supporting calculations or test reports (or extracts of these documents) should be attached with the performance calculation where applicable.

Representative curves of the performance of a type of brake equipment e.g. electro dynamic brake, can be determined by numerical or practical methods. The values can be given as a table.

4.3 General characteristics

4.3.1 Train formation

The brake system design parameters necessary to conduct the calculation shall be defined at the level of an entity e.g. an axle, bogie or a vehicle.

Calculations shall be performed for each brake equipment type. In so doing, the brake force contributions from each of the brake equipment types (e.g. disc brakes, tread brakes, electrodynamic brakes) shall be taken into consideration. All of the various types of brake equipment applied to one entity shall be identified and accounted for in the calculation.

The parameters which are typically used to define train formation include but are not limited to:

— quantity of motor axles;

— quantity of trailer axles;

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

— quantity of non-adhesion dependent brake equipment type.

NOTE 1 When there are several brake equipment types, it is preferable to identify each type (for example by means of a number: type 1, type 2, etc.).

A train can consist of one or more units or vehicles. For the purpose of this document, no distinction is made between unit and vehicle, therefore the term vehicle is used.

NOTE 2 'Unit' is used in the same sense as described in the technical specification for interoperability (TSI) relating to the rolling stock sub-system – 'Locomotives and Passenger rolling stock' of the Trans-European conventional rail system.

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

4.3.2 Characteristics of a train

4.3.2.1 Train mass

EN 15663 shall be used to provide a common set of reference masses on which the assessment of loads and performance evaluation can be based; it also describes how each is to be derived.

4.3.2.2 Static mass of the train, or axle (m_{st})

The static mass of the train and/or the static mass of the axle (as defined in 3.1) shall be used to establish the brake force required or the adhesion requirements respectively, for each operating condition required e.g. operational mass in working order, as defined in EN 15663.

When there are different 'static masses per axle values' e.g. due to different vehicle arrangements or axle mounted equipment, the static mass shall be calculated for each axle.

It may be required to assess the effect of the position of a vehicle type in a train e.g. with respect to the adhesion required.

4.3.2.3 Equivalent rotating mass (m_{rot})

The equivalent rotating mass shall be calculated using a theoretical approach or established as a result of tests, using test conditions similar to the expected operating conditions.

The wheel size applicable to the rotating mass shall be identified. Any value of equivalent rotating mass, identified by a % of static mass, is normally calculated using the assumed static mass of each vehicle within the train.

When there are different 'rotating masses per axle' e.g. a mix of trailer and motor axles, the rotating mass shall be calculated for each type.

It may be required to assess the effect of the position of a vehicle type in a train e.g. with respect to the adhesion required.

If an inertia value *(J)* due to the rotating masses is known, rather than the equivalent mass of the rotating parts, then the associated wheel diameter shall be defined, this diameter is normally the new wheel diameter. The formula for the calculation of equivalent rotating mass using inertia is shown below:

$$m_{\text{rot}} = \frac{4 \cdot J}{D^2} \tag{1}$$

where:

m_{rot}	equivalent rotating mass, in kg
J	Inertia
D	wheel diameter, in m

4.3.2.4 Dynamic mass (m_{dyn})

Dependent on 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. axle, bogie, vehicle etc.

$$m_{\mathrm{dyn}} = \sum \left(m_{\mathrm{st}} + m_{\mathrm{rot}} \right) \tag{2}$$

where:

m_{dyn}	dynamic mass, in kg
m_{st}	static mass, in kg
m_{rot}	rotating mass, in kg

4.3.2.5 Wheel diameter

The wheel diameter is measured on the nominal rolling circle.

NOTE The wheel diameter used in the emergency brake calculation is usually that which gives the lowest deceleration e.g. in the case of disc brakes, this would normally be the maximum wheel diameter.

When checking the required adhesion τ_{req} the wheel diameter used shall be the size which generates the maximum adhesion demand e.g. in the case of disc brakes, this would normally be the minimum wheel diameter.

If the train is equipped with different sizes of wheels (by design not due to wear) each size of wheel shall be identified in the train composition.

4.3.2.6 Mean train resistance

The train resistance is a component of the train decelerating force provided by the structure of the train, referred to as resistance to motion in EN 14067-4. This however uses instantaneous values. The running resistance formula considers straight and level tracks, zero wind conditions, in open air and at constant speed, The characteristic of train resistance can be 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.

As an approximation or first calculation the following mean mathematical formula derived from the instantaneous formula in EN 14067-4 shall be used:

$$\overline{F}_{\mathrm{Ra}} = A + \tfrac{2}{3} \cdot B \cdot \frac{{v_0}^2 + v_0 \cdot v_{\mathrm{fin}} + {v_{\mathrm{fin}}}^2}{v_0 + v_{\mathrm{fin}}} + \tfrac{1}{2} \cdot C \cdot \left({v_0}^2 + {v_{\mathrm{fin}}}^2 \right) \tag{3}$$

where:

$\overline{F}_{\mathrm{Ra}}$	mean value of the train resistance force, in N
v_0	initial speed of the train, in m/s
v_{fin}	final speed of the train, in m/s
A	characteristic coefficient of the train independent of speed considered as C_1 in EN 14067-4, in N
B	characteristic coefficient of the train proportional to the speed considered as C_2 in

EN 14067-4, in $\frac{\text{N}}{\text{m/s}}$

C characteristic coefficient of aerodynamic train resistance considered as C_3 in EN 14067-4, in $\frac{\text{N}}{(\text{m/s})^2}$

The above mathematical units should be used for the calculations purpose; however the speed can be expressed usually in km/h and the train resistance in N or kN.

NOTE A, B, and C coefficients are function of various parameters, e.g. mass, train length. Values for A, B, and C can be obtained using the test method given in EN 14067-4.

When conducting brake performance calculations as a first estimation, designers may ignore train resistance if they also ignore the rotating masses, until these particular values are available.

For the application of other mathematical units, the coefficients of the formula shall be adapted accordingly.

4.4 Brake equipment type characteristics

4.4.1 General

This part of the standard identifies how to calculate the braking force generated by each brake equipment type related to the rail.

When the same brake equipment is used on one part of the train under certain conditions and used on another part of the same train under other conditions (e.g. different pressure/force conditions) two different brake equipment types shall be assumed and considered in the formulae.

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

The following is a description of brake equipment types currently in use:

a) Adhesion dependent (wheel to rail)

   1) Friction brake

      i) Tread brake equipment type

      ii) Disc brake equipment type

   2) Screw hand brake

      i) Tread brake equipment type

      ii) Disc brake equipment type

   3) Electro-dynamic brake equipment type

   4) Fluid retarder brake equipment type

b) Adhesion independent

   1) Eddy current brake equipment type

2) Magnetic track brake equipment type

When using the following formulae to calculate the brake force for the different brake equipment types (i) and possible arrangements of this equipment e.g. a clasp brake arrangement of a tread brake, the number of arrangements fitted per entity shall be considered. As applicable, factors have been identified in each of the formulae and are shown in the examples given.

For tread and disc brake equipment types typically a pressure applied brake cylinder is used. However, a spring applied brake cylinder may be used. For the calculation of the output force (F_c) the formula shall consider the following with reference to Figure 1:

a) The internal ratio/factor of the cylinder (i_C) is normally one except for a design with an internal cylinder ratio. The sign of the ratio/factor when considered in the calculation formula is dependent on the type of brake equipment i.e. for pressure applied brake equipment it is a **positive value** and for spring applied brake equipment it is **negative value.**

b) The cylinder spring force ($F_{S,C}$) in a pressure applied brake cylinder operates to release the brake force (cylinder piston return) and is therefore a **negative** value. For a spring applied brake $F_{S,C}$ operates as a braking force and is therefore a **positive** value.

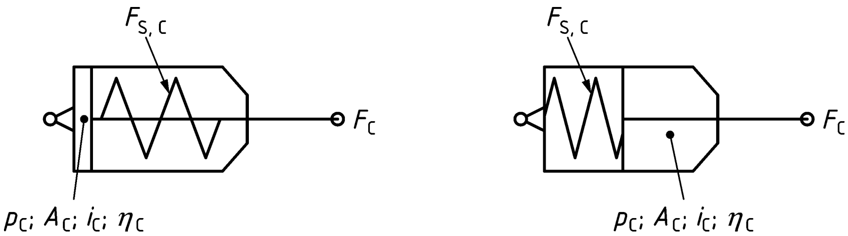


Figure 1 — Basic principle of a pressure applied (left) and spring applied (right) cylinder

Subclauses 4.4.2 and 4.4.3 consider the formulae for a representative tread brake unit and disc brake unit with an arrangement as shown in the corresponding figures. Different designs of tread and disc brake units may have the component parts arranged differently and the formulae may be changed to represent the physical arrangement and its effect on the force calculation.

4.4.2 Tread braking

The braking effect is caused by the friction of the brake blocks to the wheel treads; the resulting heat needs to be dissipated through the brake blocks and the wheels.

The brake cylinder force is transferred to the brake blocks with the relevant degrees of efficiency (cylinder, rigging and brake lever efficiency). Additionally, design-specific spring forces also shall be overcome (e.g. cylinder spring, slack adjuster spring).

Calculation of the braking force is directly derived from the defined mechanical functional chain. It is based on a computation of forces only, taking into consideration rigging ratios and degrees of efficiency from a mathematical point of view.

The braking force is independent of the wheel diameter.

4.4.3 Disc braking

The braking effect is caused by the friction of the brake pad to the brake disc the resultant heat being dissipated by the brake pad and disc.

The brake cylinder force is transferred to the brake pads with the relevant degrees of efficiency (cylinder, rigging and brake caliper efficiency); additionally design-specific spring forces also shall be overcome (e.g. cylinder spring, slack adjuster spring).

Calculation of the braking force is directly derived from the defined mechanical functional chain. It is based on a computation of forces only, taking into consideration rigging (caliper lever) ratios and degrees of efficiency from a mathematical point of view.

The braking force is dependent on the wheel diameter.

4.4.4 Forces of friction brake (tread brake) equipment

4.4.4.1 Tread brake unit

The brake equipment of a tread brake unit acts on one brake block configuration per cylinder as shown in Figure 2, this shows a representative arrangement of a pressure applied tread brake unit. The principle is the same for a spring applied design but the values for internal brake unit spring force and the internal ratio of a cylinder are reversed.

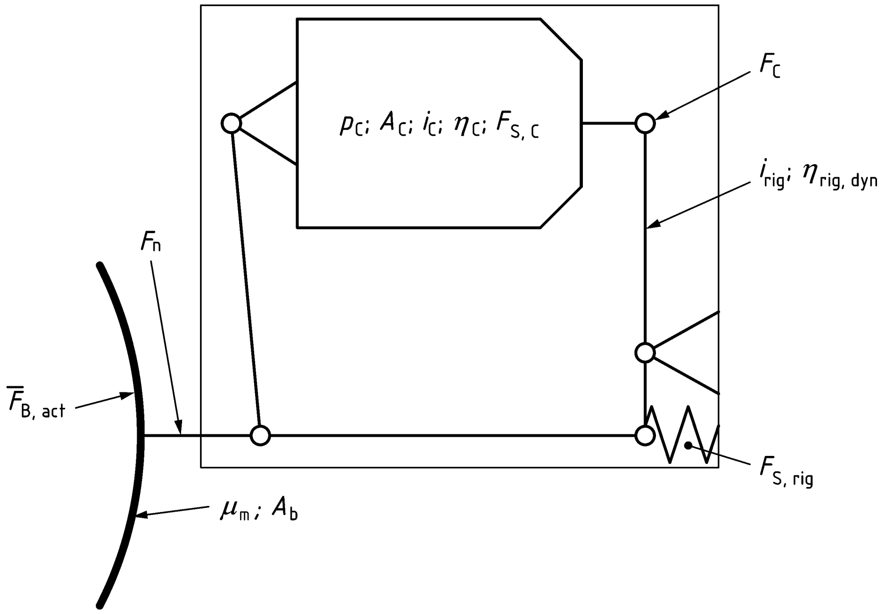


Figure 2 — Representative tread brake unit

The braking force characteristic of a tread brake unit can be expressed by:

Cylinder output force

$$F_C = p_C \cdot A_C \cdot i_C \cdot \eta_C + F_{s,C} \tag{4}$$

Force on the application point

$$F_n = F_C \cdot i_{rig} \cdot \eta_{rig,dyn} + F_{s,rig} \tag{5}$$

Mean braking force per brake unit

$$\overline{F}_{B,act} = F_n \cdot \mu_m \tag{6}$$

Pressure per application point

$$p_{ap} = \frac{F_n}{A_b} \tag{7}$$

where:

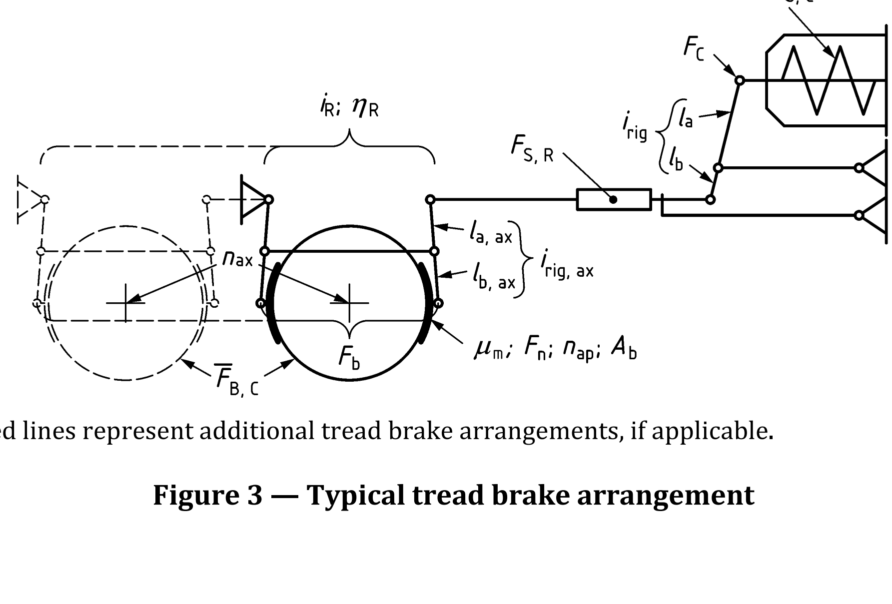
p_C	brake cylinder pressure, in Pa
A_C	brake cylinder piston area, in m²
η_C	internal brake cylinder efficiency
i_C	internal brake cylinder ratio/factor
$F_{S,C}$	sum of internal brake actuator spring forces, in N
$\eta_{rig,dyn}$	rigging efficiency in dynamic condition
i_{rig}	rigging ratio
$F_{S,rig}$	rigging spring force normally used as a rigging return spring, therefore a negative value, in N
μ_m	mean friction coefficient of brake block
A_b	total area of the friction surface per application point, in m²

NOTE With reference to rigging efficiencies, ratios and return springs, for tread brake units these can be wholly contained in the unit itself.

4.4.4.2 Tread brake

The brake equipment of a tread brake acts on an arrangement with several blocks per cylinder as shown in Figure 3 (a typical clasp brake arrangement). However, this is also applicable to a pusher brake arrangement with one application point per wheel.

The force generated by the brake cylinder is typically transmitted to the main brake rods via the central brake linkage and then on to the brake shoes via the slack adjuster and the brake levers.



NOTE Dashed lines represent additional tread brake arrangements, if applicable.

Figure 3 — Typical tread brake arrangement

The braking force characteristic of a tread brake arrangement can be expressed by:

Output cylinder force

$$F_{\mathrm{C}} = p_{\mathrm{C}} \cdot A_{\mathrm{C}} \cdot i_{\mathrm{C}} \cdot \eta_{\mathrm{C}} + F_{\mathrm{S,C}} \tag{8}$$

Total force on the application points

$$F_{\mathrm{b}} = \left(F_{\mathrm{C}} \cdot i_{\mathrm{rig}} + F_{\mathrm{S,R}}\right) \cdot i_{\mathrm{R}} \cdot \eta_{\mathrm{R}} \tag{9}$$

With the ratios

$$i_{\mathrm{rig}} = l_{\mathrm{a}} / l_{\mathrm{b}} \tag{10}$$

and

$$i_{\mathrm{R}} = n_{\mathrm{ax}} \cdot n_{\mathrm{ap}} \cdot i_{\mathrm{rig,ax}} \tag{11}$$

with

$$i_{\mathrm{rig,ax}} = l_{\mathrm{a,ax}} / l_{\mathrm{b,ax}} \tag{12}$$

Mean brake force per complete tread brake arrangement

$$\overline{F}_{\mathrm{B,C}} = F_{\mathrm{b}} \cdot \mu_{\mathrm{m}} \tag{13}$$

Force per application point

$$F_{\mathrm{n}} = \frac{F_{\mathrm{b}}}{2 \cdot n_{\mathrm{ax}} \cdot n_{\mathrm{ap}}} \tag{14}$$

Pressure per application point

$$p_{\mathrm{ap}} = \frac{F_{\mathrm{n}}}{A_{\mathrm{b}}} \tag{15}$$

where:

p_C brake cylinder pressure, in Pa

A_C brake cylinder piston area, in m^2

η_C internal brake unit efficiency

i_C internal brake cylinder ratio/factor

The value of the ratio or factor is normally one; alternatively if internal rigging is present this may vary.

The sign of the ratio or factor is dependent on the type of brake equipment i.e. for pressure applied brake equipment it is **positive**; for spring applied brake equipment it is **negative**.

$F_{S,C}$ sum of internal brake cylinder spring forces, in N

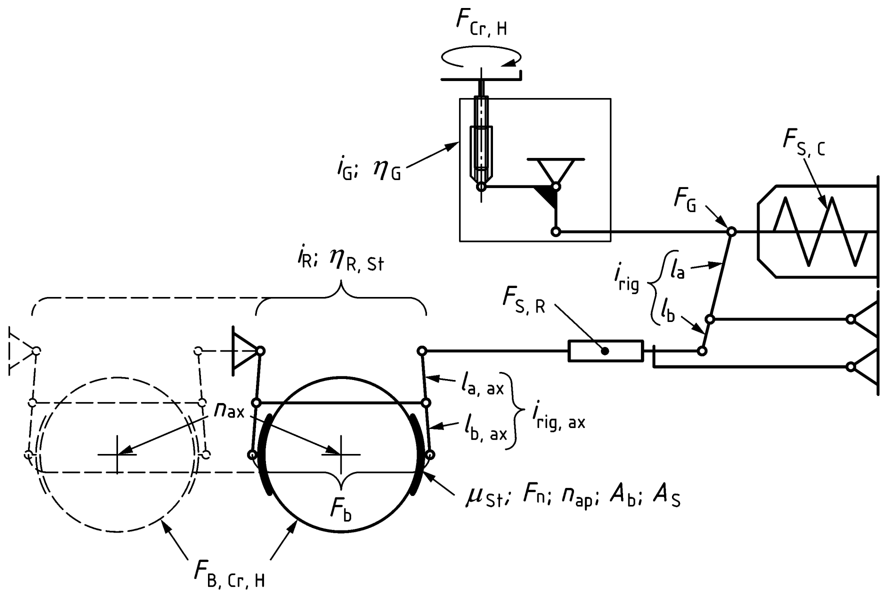
$F_{S,R}$ slack adjuster force normally assumed to be a negative value, in N

i_{rig} central brake rigging ratio (if a changeover load is defined, the ratio shall be calculated to the

	appropriate load)
l_a	lever length of the central rigging – BC connection to pivot, in m
l_b	lever length of the central rigging – pivot to output connection, in m
i_R	overall rigging ratio behind the slack adjuster; e.g. i_R = 4 for a conventional 2-axle freight wagon with 4 application points (refer to n_{ap}) per axle
η_R	overall rigging efficiency
$i_{rig,ax}$	axle rigging ratio per application point; typically the factor is $l_{a,ax}$ / $l_{b,ax}$ = 1
$l_{a,ax}$	lever length of the axle rigging – BC connection to pivot, in m
$l_{b,ax}$	lever length of the axle rigging – pivot to output connection, in m
μ_m	mean block friction coefficient
n_{ap}	number of force application points per wheel in a clasp arrangement n_{ap} = 2 and if there is only 1 application point per wheel (single face) n_{ap} = 1
n_{ax}	number of braked axles
A_b	total area of the friction surface per application point, in m^2

4.4.4.3 Screw hand brake (Tread brake)

The brake equipment of a tread brake typically acts on an arrangement with several brake blocks per cylinder as shown in Figure 4. The force generated by the hand wheel or a crank handle is typically transmitted to the main brake rods via the central brake linkage and then on to the brake blocks via the slack adjuster and the brake levers. This arrangement is typically used as a parking brake.



NOTE Dashed lines represent additional tread brake arrangements, if applicable.

Figure 4 — Typical screw hand brake (tread brake) arrangement

The braking force of a screw hand brake (tread brake) arrangement can be expressed by:

Gear force

$$F_{\mathrm{G}} = F_{\mathrm{Cr,H}} \cdot i_{\mathrm{G}} \cdot \eta_{\mathrm{G}} + F_{\mathrm{S,C}} \tag{16}$$

Total force of all application points

$$F_{\mathrm{b}} = \left(F_{\mathrm{G}} \cdot i_{\mathrm{rig}} + F_{\mathrm{S,R}}\right) \cdot i_{\mathrm{R}} \cdot \eta_{\mathrm{R,st}} \tag{17}$$

With the ratios

$$i_{\mathrm{rig}} = l_{\mathrm{a}} / l_{\mathrm{b}} \tag{18}$$

and

$$i_{\mathrm{R}} = n_{\mathrm{ax}} \cdot n_{\mathrm{ap}} \cdot i_{\mathrm{rig,ax}} \tag{19}$$

with

$$i_{\mathrm{rig,ax}} = l_{\mathrm{a,ax}} / l_{\mathrm{b,ax}} \tag{20}$$

Total braking force per complete tread brake arrangement

$$F_{\mathrm{B,Cr,H}} = F_{\mathrm{b}} \cdot \mu_{\mathrm{st}} \tag{21}$$

Force per application point

$$F_{\mathrm{n}} = \frac{F_{\mathrm{b}}}{2 \cdot n_{\mathrm{ax}} \cdot n_{\mathrm{ap}}} \tag{22}$$

Pressure per application point

$$p_{\mathrm{ap}} = \frac{F_{\mathrm{n}}}{A_{\mathrm{b}}} \tag{23}$$

where:

$F_{Cr,H}$	crank handle force, in N
η_G	gear efficiency
i_G	gear ratio
$F_{S,C}$	sum of internal brake cylinder spring forces, in N
$F_{S,R}$	slack adjuster counterforce normally assumed to be a negative value, in N
i_{rig}	central brake rigging ratio (if a changeover load is defined, the ration shall be calculated to the appropriate load), in N
l_a	lever length of the central rigging, in m
l_b	lever length of the central rigging, in m
i_R	overall rigging ratio behind the slack adjuster; e.g.: $i_{\mathrm{R}} = 4$ for a conventional 2-axle freight wagon with 4 application point (refer to n_{ap}) per axle

$\eta_{R,st}$ static overall rigging efficiency

$i_{rig,ax}$ axle rigging ratio per application point; typically the factor is $l_{a,ax}$ / $l_{b,ax}$ = 1

$l_{a,ax}$ lever length of the axle rigging, in m

$l_{b,ax}$ lever length of the axle rigging, in m

μ_{st} static block friction coefficient

n_{ap} number of force application points per wheel

in clasp arrangement n_{ap} = 2 and if there is only one application point per wheel (single face) n_{ap} = 1

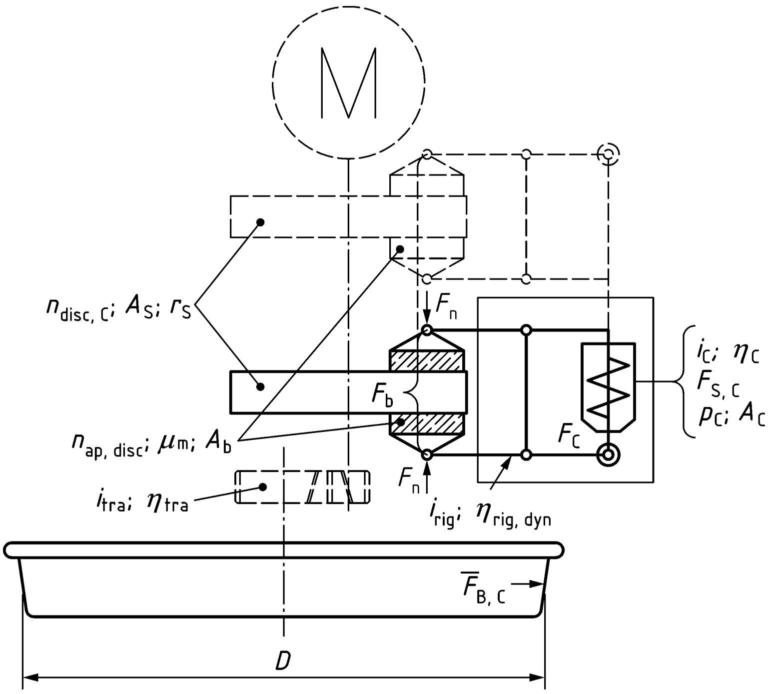
n_{ax} number of braked axles

A_b total area of the friction surface per application point, in m²

4.4.5 Forces of friction brake (disc brake) equipment

4.4.5.1 Disc brake unit arrangement

A disc brake unit typically acts on one caliper arrangement per cylinder as shown in Figure 5. However, it can operate on more than one caliper arrangement.



NOTE Dashed lines represent additional disc caliper arrangements, if applicable.

Figure 5 — Typical pressure applied disc brake unit arrangement

The braking force characteristic is typically expressed by:

Cylinder output force

$$F_C = p_C \cdot A_C \cdot i_C \cdot \eta_C + F_{S,C} \tag{24}$$

NOTE Dependent on the design of the actual brake unit being considered the order of the physical components can be reflected within the formula e.g. the position and effect of spring force.

Total clamp force per disc brake unit arrangement

$$F_{\mathrm{b}} = F_{\mathrm{C}} \cdot i_{\mathrm{rig}} \cdot \eta_{\mathrm{rig,dyn}} \tag{25}$$

Force per application point

$$F_{\mathrm{n}} = \frac{F_{\mathrm{b}}}{n_{\mathrm{disc,C}} \cdot n_{\mathrm{ap,disc}}} \tag{26}$$

Tangential force on each disc

$$F_{\mathrm{t}} = \frac{F_{\mathrm{b}} \cdot \mu_{\mathrm{m}}}{n_{\mathrm{disc,C}}} \tag{27}$$

Pressure per application point

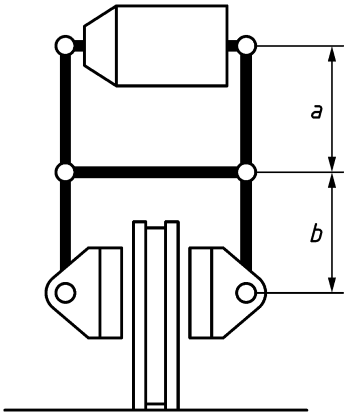
$$p_{\mathrm{ap}} = \frac{F_{\mathrm{n}}}{A_{\mathrm{b}}} \tag{28}$$

Mean braking force related to the wheel per disc brake unit arrangement

$$\overline{F}_{\mathrm{B,C}} = F_{\mathrm{b}} \cdot \mu_{\mathrm{m}} \cdot \frac{r_{\mathrm{s}}}{D/2} \cdot \frac{i_{\mathrm{tra}}}{\eta_{\mathrm{tra}}} \tag{29}$$

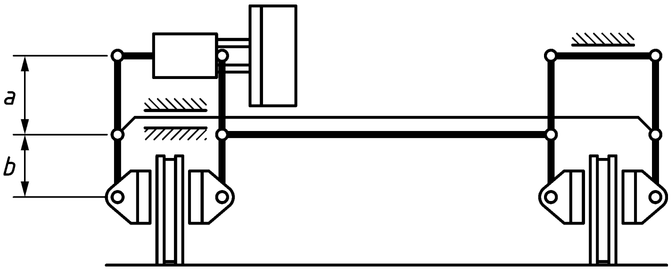
where:

p_C	brake cylinder pressure, in Pa
A_C	brake cylinder piston area, in m²
η_C	brake cylinder efficiency
i_C	internal brake unit ratio/factor
$F_{S,C}$	internal brake cylinder spring forces, in N Value is positive if it supports the braking activity, negative in case of a counter force.
$\eta_{rig,dyn}$	rigging efficiency in dynamic condition
i_{rig}	caliper ratio i.e. the ratio when considering both the caliper levers per disc e.g. single lever ratio (a/b) × 2 for single disc per cylinder arrangement



or

single lever ratio ((a/b) × 2) × 2 for 2 disc per cylinder arrangement



When performing a calculation the total rigging ratio should consider the number of discs operated by a single cylinder if more than one.

$n_{disc,C}$ number of brake discs per brake unit
e.g. normally $n_{\mathrm{disc,C}} = 1$
however in a double caliper arrangement $n_{\mathrm{disc,C}} = 2$.
in a wheel mounted disc arrangement the number of discs per wheel $n_{\mathrm{disc,C}} = 1$.

$n_{ap,disc}$ number of force application points per brake disc
in caliper arrangement $n_{\mathrm{ap,disc}} = 2$

μ_m mean pad friction coefficient

η_{tra} transmission efficiency used when a drive gearbox is positioned between the wheel and the axle on which the disc is mounted. If there is no transmission gearbox $\eta_{\mathrm{tra}} = 1$.

i_{tra} transmission ratio used when a drive gearbox is positioned between the wheel and the axle on which the disc is mounted. If there is no transmission gearbox $i_{\mathrm{tra}} = 1$.

r_s mean swept radius of the brake pad on the disc face, in m

D diameter of the wheel, in m

A_b total area of the friction surface per application point, in m^2

4.4.5.2 Screw hand brake (disc brake)

The brake equipment of a screw hand brake acts on an arrangement with several calipers as shown in Figure 6. The force generated by the hand wheel or a crank handle is typically transmitted by a gearbox and then via flexible cables to the calipers and then on to the brake pads via the caliper levers. This is typically used as a parking brake.

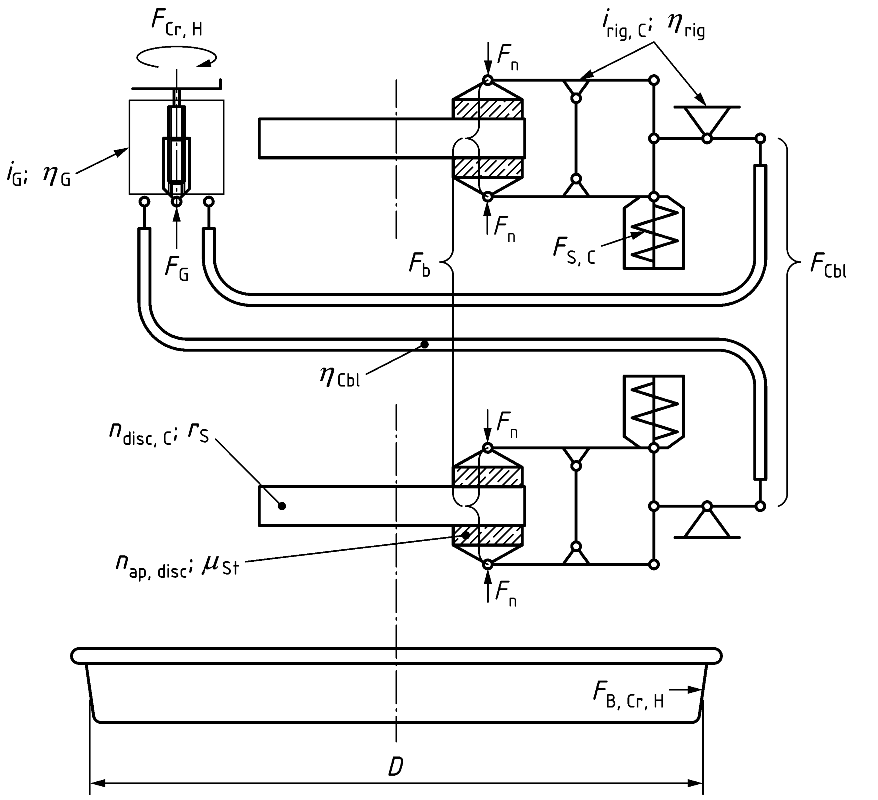


Figure 6 — Typical screw brake arrangement of disc brakes

The braking force of a screw hand brake (disc brake) arrangement is typically expressed by:

Gear force

$$F_G = F_{Cr,H} \cdot i_G \cdot \eta_G \tag{30}$$

Total cable force

$$F_{Cbl} = F_G \cdot \eta_{Cbl} \tag{31}$$

Total force of all the application points

$$F_b = \left(F_{Cbl} + n_{disc,C} \cdot F_{S,C}\right) \cdot i_{rig,C} \cdot \eta_{rig} \tag{32}$$

Application force per application point

$$F_n = \frac{F_b}{n_{disc,C} \cdot n_{ap,disc}} \tag{33}$$

Tangential force per disc

$$F_t = \frac{F_b \cdot \mu_{St}}{n_{disc,C}} \tag{34}$$

Total braking force per screw brake related to the wheel

$$F_{\mathrm{B,Cr,H}} = F_{\mathrm{b}} \cdot \mu_{\mathrm{St}} \cdot \frac{r_{\mathrm{s}}}{D/2} \tag{35}$$

where:

$F_{Cr,H}$	crank handle force, in N
η_G	gear efficiency
η_{Cbl}	cable efficiency
i_G	gear ratio
$F_{S,C}$	sum of internal brake cylinder spring forces, in N Value is positive if it supports the braking activity, negative in case of a counter force.
$i_{rig,C}$	rigging ratio per calliper i.e. the ratio when considering both caliper levers and the rigging between caliper and cable
η_{rig}	rigging efficiency
$n_{disc,C}$	number of brake discs per brake unit e.g. normally $n_{\mathrm{disc,C}} = 1$ However in a double caliper arrangement $n_{\mathrm{disc,C}} = 2$. In a wheel mounted disc arrangement the number of discs per wheel $n_{\mathrm{disc,C}} = 1$.
$n_{ap,disc}$	number of force application points per brake disc In caliper arrangement $n_{\mathrm{ap,disc}} = 2$.
μ_{St}	static brake pad friction coefficient
r_s	mean swept radius of the brake pad on the disc face, in m
D	diameter of the wheel, in m

4.4.6 Mean dynamic coefficient of friction (μ_m) tread and disc brakes

The nominal dynamic values of the coefficient of friction shall be established using the methods set out in prEN 15328 for brake pads or EN 16452 for brake blocks.

Corresponding test reports (or extracts of these documents) should be attached with the performance calculation.

4.4.7 Brake forces of other brake equipment types

4.4.7.1 General

For other brake equipment types, other than friction brakes, in particular where the brake forces are speed dependent following defined curves e.g. as per the following subclauses in 4.4.7, the mean brake force over certain speed ranges can be determined by numerical or practical methods and can be given as a table.

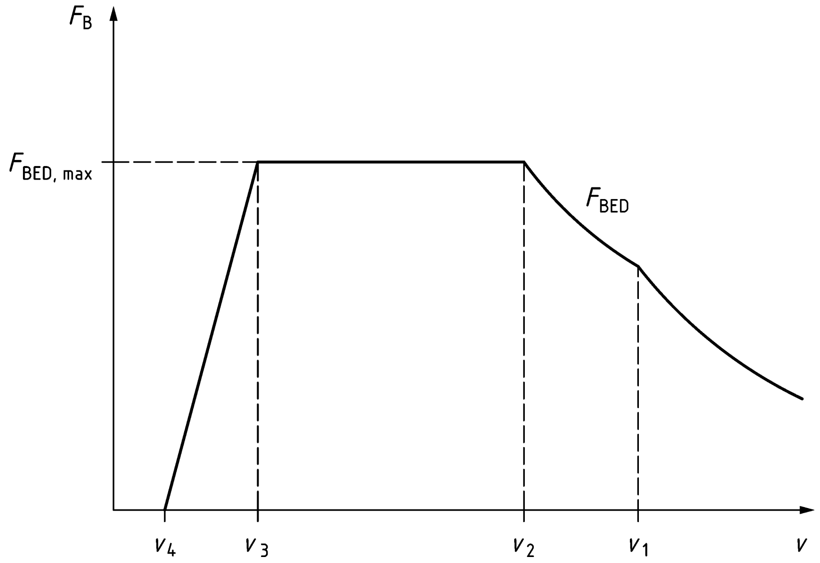
In the absence of mean values of brake force throughout the relevant speed range the following describes a method of conducting calculations to estimate the brake force (see 4.8.1).

If alternative formulae to those used in this clause are known and used the origin of the formulae should be stated.

4.4.7.2 Electrodynamic brake

Generally, the electro dynamic brake force may be represented by an idealized characteristic curve. This principle is shown in Figure 7.

Figure 7 shows a characteristic instantaneous braking effort curve achieved by a typical electric motor when a single brake demand is requested and the vehicle is braked from its initial speed to zero.



NOTE The indices 1, 2, 3, 4 of the speed v, are given in the sense of the braking process, starting with the initial speed.

Figure 7 — Characteristic of the electrodynamic brake force

The section of the curve for speeds higher than v_1 (depending on $\frac{1}{v^2}$) 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 is used with electro dynamic braking when the power is limited due to the limitation of the motor design.

The section of the curve from v_2 *to* v_3 is assumed to be the maximum electro dynamic braking force.

The section of the curve from v_3 to v_4 is the portion of the curve where the electro dynamic force reduces due to speed and is normally replaced by the blended friction brake force to maintain the demanded brake force.

The curve is composed of:

a linear section from v_4 to v_3

$$F_{\text{BED}} = F_{\text{BED,max}} \cdot \frac{v - v_4}{\text{v}_3 - v_4} \tag{36}$$

a constant section from v_3 to v_2

$$F_{\text{BED}} = F_{\text{BED,max}} \tag{37}$$

a hyperbolic section with constant power from v_2 to v_1

$$F_{\mathrm{BED}} = F_{\mathrm{BED,max}} \cdot \frac{v_2}{v} \tag{38}$$

a section depending on 1/ v_2 for speeds higher than v_1

$$F_{\mathrm{BED}} = F_{\mathrm{BED,max}} \cdot \frac{v_2 \cdot v_1}{v^2} \tag{39}$$

where:

F_{BED}	instantaneous electrodynamic braking force, in N
$F_{BED,\max}$	maximum electrodynamic braking force (= value of the force in the constant section), in N
v	instantaneous speed, in m/s
$v_1..v_4$	particular speeds, in m/s

If the deceleration of the train is assumed relatively constant during the braking, the mean value of the electro dynamic braking force from v_0 to 0 is given by the Formulae (40), (41) and (42).

These formulae assume braking to a stop. They may be adapted for slowing calculation using the applicable speed ranges.

Considering the initial speed when in the speed range v_3 to 0 and with reference to the characteristic curve described in Figure 7, it is assumed that a fully compensated brake force is achieved in this speed range, however for simplification this force is assumed to be provided by the electrodynamic brake only. The formulae describe the mean value of braking effort achieved while stopping, when ignoring the electro dynamic brake response time.

$$\text{if } 0 < v_0 \leq v_2 \qquad \overline{F_{\mathrm{BED}}} = F_{\mathrm{BED,max}} \tag{40}$$

$$\text{if } v_2 < v_0 \leq v_1 \qquad \overline{F_{\mathrm{BED}}} = F_{\mathrm{BED,max}} \cdot \left[\frac{3 \cdot v_0^2 \cdot v_2}{2 \cdot v_0^3 + v_2^3} \right] \tag{41}$$

$$\text{if } v_1 < v_0 \qquad \overline{F_{\mathrm{BED}}} = F_{\mathrm{BED,max}} \cdot \left[\frac{6 \cdot v_0^2 \cdot v_1 \cdot v_2}{3 \cdot v_0^4 + v_1^4 + 2 \cdot v_1 \cdot v_2^3} \right] \tag{42}$$

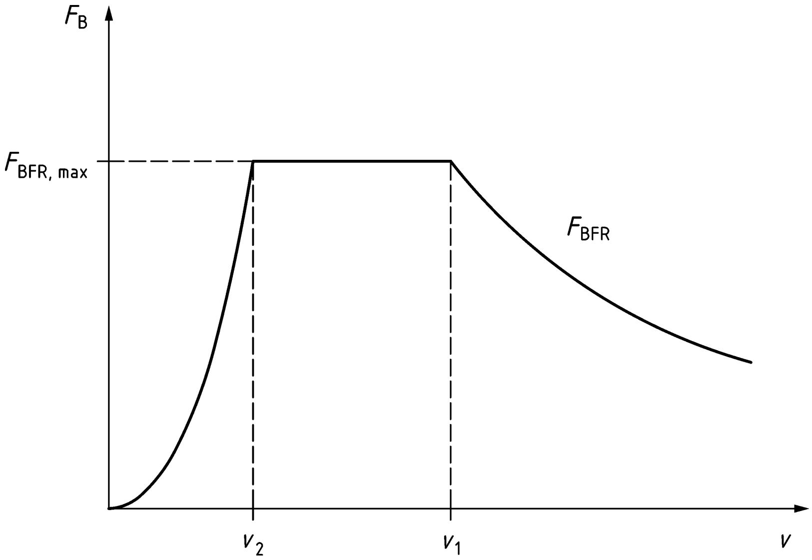
where:

$\overline{F_{BED}}$	mean electrodynamic braking force depending on the initial speed, in N
$F_{BED,\max}$	maximum electro dynamic braking force (equals the value of the force in the constant section), in N
v_0	initial speed, in m/s
$v_1..v_4$	particular speeds, in m/s

4.4.7.3 Fluid retarder

Generally, the fluid retarder brake force may be represented by an idealized characteristic curve. This principle is shown in Figure 8.

The retarder braking characteristic is a function of speed and is assumed to develop in three phases. After a hyperbolic increase of the braking force, the maximum braking force $F_{BFR,\max}$ is reached at a speed of v_1. As from the speed of v_2, the braking force decreases parabolically. The speeds v_1 to v_2 allow the characteristic braking force profile to be adjusted.



NOTE 1 The indices 1, 2 of the speed *v*, are given in the sense of the braking process, starting with the initial speed.

NOTE 2 The fluid retarder brake force can vary as a function of the static mass.

Figure 8 — Brake force characteristic of a fluid retarder

The section of the curve from $v_{\max}$ to v_1 is used with fluid retarder braking when the power is limited. This hyperbolic section of the curve is represented by:

$$\text{if } v_1 < v \le v_{\max} \quad F_{\text{BFR}} = F_{\text{BFR,max}} \cdot \frac{v_1}{v} \tag{43}$$

The section of the curve from v_1 to v_2 is the assumed maximum fluid retarder braking force. It is represented by:

$$\text{if } v_2 < v \le v_1 \quad F_{\text{BFR}} = F_{\text{BFR,max}} \tag{44}$$

The section of the curve from v_2 to 0 is the parabolic portion of the curve where the fluid retarder force reduces due to speed and is normally replaced by the blended friction brake force to maintain the demanded brake force. It is represented by:

$$\text{if } 0 < v \leq v_2 \qquad F_{\mathrm{BFR}} = F_{\mathrm{BFR,max}} \cdot \frac{v^2}{v_2^2} \tag{45}$$

If the deceleration of the train is assumed relatively constant during the braking, the mean value of the fluid retarder braking force from v_0 to 0 is given by Formulae (46) and (47).

These formulae assume braking to a stop. They may be adapted for slowing calculation using the applicable speed ranges.

Considering the initial speed when in the speed range v_2 to 0 and with reference to the characteristic curve described in Figure 8, it is assumed that a fully compensated brake force is achieved in this speed range, however for simplification this force is assumed to be provided by the fluid retarder only. Formulae (46) and (47) describe the mean value of braking effort achieved while stopping, when ignoring the fluid retarder brake response time.

$$\text{If } v_0 \leq v_1 \qquad \overline{F_{\mathrm{BFR}}} = F_{\mathrm{BFR,max}} \tag{46}$$

$$\text{if } v_1 < v_0 \leq v_{\mathrm{max}} \qquad \overline{F_{\mathrm{BFR}}} = F_{\mathrm{BFR,max}} \cdot \left[\frac{3 \cdot v_0^2 \cdot v_1}{2 \cdot v_0^3 + v_1^3} \right] \tag{47}$$

where:

F_{BFR}	instantaneous fluid retarder braking force, in N
$\overline{F_{\mathrm{BFR}}}$	mean fluid retarder braking force depending on the initial speed, in N
$F_{BFR,\mathrm{max}}$	maximum fluid retarder braking force (equals the value of the force in the constant section), in N
v_0	initial speed, in m/s
$v_1..v_2$	particular speeds, in m/s
v_{max}	maximum design speed, in m/s

4.4.7.4 Magnetic track brake

4.4.7.4.1 Characteristic curve

Generally, the magnetic track brake force is represented by a curve that shows the braking force versus the speed. This principle is shown in Figure 9.

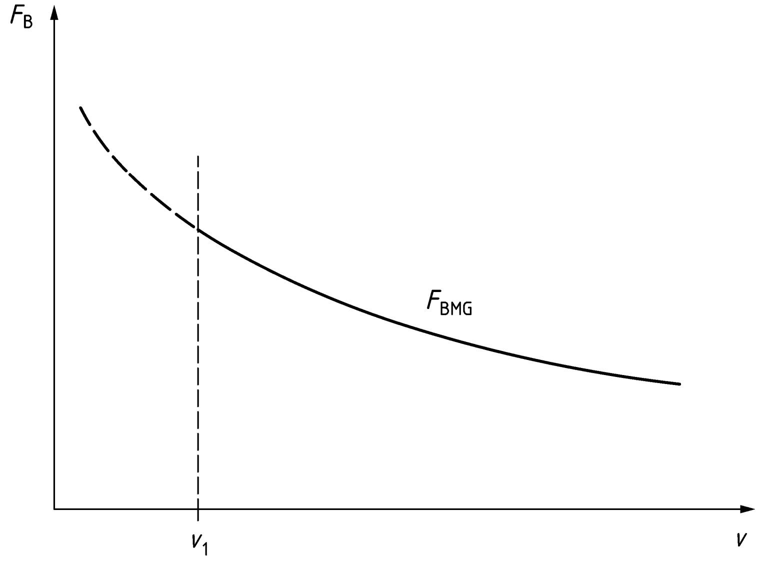


Figure 9 — Characteristics of the magnetic track brake force

4.4.7.4.2 Mean coefficient of friction and the resultant mean braking force

The instantaneous magnetic track brake braking force is:

$$F_{\mathrm{BMG}} = F_{\mathrm{AMG}} \cdot \mu_{\mathrm{MG}} = F_{\mathrm{AMG}} \cdot \frac{1}{k_1 \cdot v + k_0} \tag{48}$$

The mean magnetic track braking force can be developed from Formula (49) as follows:

$$\overline{F_{\mathrm{BMG}}} = F_{\mathrm{AMG}} \cdot \frac{v_0^2 - v_1^2}{\frac{2}{3} \cdot k_1 \cdot \left(v_0^3 - v_1^3\right) + k_0 \cdot \left(v_0^2 - v_1^2\right)} \tag{49}$$

Therefore the mean coefficient of friction between the magnet and the track depending on the initial speed v_0 and cut off speed v_1 is as follows:

$$\overline{\mu_{MG}} = \frac{v_0^2 - v_1^2}{\frac{2}{3} \cdot k_1 \cdot \left(v_0^3 - v_1^3\right) + k_0 \cdot \left(v_0^2 - v_1^2\right)} \tag{50}$$

where:

F_{BMG}	instantaneous magnetic track braking force, in N
$\overline{F_{\mathrm{BMG}}}$	mean magnetic track braking force, in N
F_{AMG}	total magnetic attraction force (≅ constant) (typically the total force of two magnets in a bogie assembly), in N
μ_{MG}	instantaneous coefficient of friction between the magnet and the track
$\overline{\mu}_{MG}$	mean coefficient of friction between the magnet and the track depending on the initial and switch of speed
v	instantaneous speed, in s/m

v_0 initial speed of braking, in m/s

v_1 cut off speed of the magnetic track brake, in m/s

k_0 constant coefficient (normally provided by the supplier)

k_1 constant coefficient (normally provided by the supplier), in m/s

4.4.7.4.3 Distance calculation based on total braking force including magnetic track brake

For an estimation of the total stopping/slowing distance within the range of v_0 to v_{fin}, the calculation is separated into different stages of braking:

— Phase 1: response time;

— Phase 2: braking with magnetic track brake force and the sum of the friction brake force; (v_0 to v_1)

— Phase 3: braking with the sum of friction brake force without magnetic track brake.(v_1 to v_{fin})

This requirement leads to:

$$s = v_0 \cdot t_e (\text{Phase1}) + \frac{m_{\text{dyn}}}{\overline{F_{\text{BMG}}} + \sum \overline{F_{\text{B}}}} \cdot \frac{v_0^2 - v_1^2}{2} (\text{Phase 2}) + \frac{m_{\text{dyn}}}{\sum \overline{F_{\text{B}}}} \cdot \frac{v_1^2 - v_{\text{fin}}^2}{2} (\text{Phase 3}) \quad (51)$$

It is acceptable to use a simplified formula, see (52), if the magnetic track brake cut off speed is nominally 20 % or less of the initial speed and the contribution of the magnetic track brake is 20 % or less of the total train braking force and assuming straight and level track, without considering external forces e.g. resistance. Within these limits this simplification leads to an error of less than 1 % in the stopping distance, i.e. shorter. If this degree of accuracy is not acceptable Formula (51) should be used.

$$s = v_0 \cdot t_e (\text{Phase 1}) + \frac{m_{\text{dyn}}}{\overline{F_{\text{BMG}}} + \sum \overline{F_{\text{B}}}} \cdot \frac{v_0^2 - v_{fin}^2}{2} (\text{Phase 2}) \quad (52)$$

where:

s stopping/slowing distance due to total braking force, in m

t_e equivalent response time, in s

$\overline{F_B}$ mean friction brake force, in N

v_0 initial speed of braking, in m/s

v_{fin} final speed, in m/s

m_{dyn} dynamic mass

$\overline{F_{BMG}}$ mean magnetic track braking force, in N

4.4.7.5 Eddy current brake

4.4.7.5.1 Characteristic curve

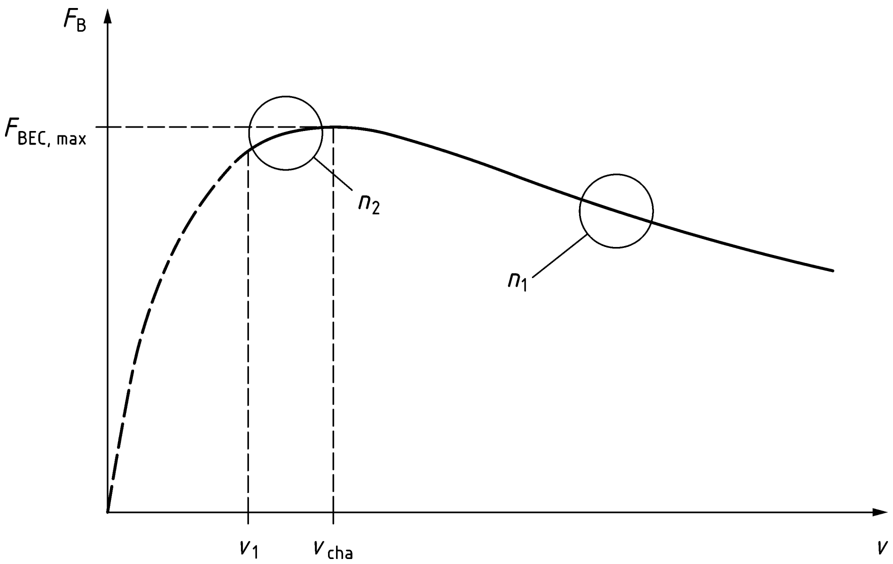


Figure 10 — Characteristics of the eddy current brake force

The eddy current braking force depends on:

— the gap between the shoe and the track;

— the instantaneous speed;

— the intensity of the magnetic field.

4.4.7.5.2 Mean braking force of the eddy current brake

The instantaneous eddy current braking force is:

$$F_{BEC} = F_{BEC,max} \cdot \frac{2}{\left(\frac{v}{v_{cha}}\right)^n + \left(\frac{v_{cha}}{v}\right)^n} \tag{53}$$

with:

$n = n_1$ for $v \geq v_{cha}$

$n = n_2$ for $v < v_{cha}$

The mean eddy current braking force is developed from Formula (54) as follows:

$$\overline{F_{BEC}} = F_{BEC,max} \cdot \frac{v_0^2 - v_1^2}{\left(\frac{v_{cha}^n}{2-n} \cdot \left(v_0^{2-n} - v_1^{2-n}\right) + \frac{1}{2+n} \cdot \frac{\left(v_0^{2+n} - v_1^{2+n}\right)}{v_{cha}^n}\right)} \tag{54}$$

where:

F_{BEC}	instantaneous eddy current braking force, in N
$\overline{F_{BEC}}$	mean eddy current braking force, in N
$F_{BEC,\max}$	maximum eddy current braking force, in N
v	instantaneous speed, in m/s
v_0	initial speed, in m/s
v_1	speed at eddy current brake switch off, in m/s
v_{cha}	characteristic speed where F_{BEC} = $F_{BEC,ma}$, in m/s
n_1	value of power in speed range above v_{cha} normally obtained from the supplier
n_2	value of power in speed range below v_{cha} normally obtained from the supplier

4.4.7.5.3 Distance calculation based on total braking force including eddy current brake

For an estimation of the total stopping/slowing distance within the range of v_0 to v_{fin}, the calculation is separated into different stages of braking:

— Phase 1 step: response time;

— Phase 2 step: braking with eddy current brake and the sum of the friction brake force; v_0 to v_1

— Phase 3 step: braking with the sum of the friction brake force without eddy current brake. This is because the eddy current brake is permitted to be used from the maximum operating speed down to 50 km/h with notable effect on the braking distance. v_1 to v_{fin}

This requirement leads to following formula:

$$s = v_0 \cdot t_e (\text{Phase } 1) + \frac{m_{dyn}}{\overline{F_{BEC}} + \sum \overline{F_B}} \cdot \frac{v_0^2 - v_1^2}{2} (\text{Phase } 2) + \frac{m_{dyn}}{\sum \overline{F_B}} \cdot \frac{v_1^2 - v_{fin}^2}{2} (\text{Phase } 3) \quad (55)$$

Where:

s	stopping/slowing distance due to total braking force, in m
v_{fin}	final speed, in m/s
v_0	initial speed, in m/s
v_1	speed at eddy current brake switch off, in m/s
t_e	equivalent response time, in s
$\overline{F_B}$	mean friction brake force, in N
$\overline{F_{BEC}}$	mean eddy current braking force, in N

Formula (55) represents simultaneous braking with eddy current and friction brakes, often the eddy current brake is used on its own. If the eddy current brake is used only, Formula (55) should be adapted accordingly.

4.4.8 Time characteristics

4.4.8.1 Derivation of brake equipment time characteristics

The time characteristic of a brake equipment type can be simulated by numerical methods or determined by practical methods or by estimations. The values can be given as a table.

For the purpose of this standard typically, the time characteristic is considered for each brake equipment type when the brake 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.

4.4.8.2 Creation of input data

4.4.8.2.1 Delay time (t_a)

Period of time commencing when a change, (positive or negative), in brake demand is initiated and ending when achieving *a* % of the established braking force deceleration or pressure of the brake equipment (see Figure 11).

4.4.8.2.2 Build-up time (t_{ab})

Period of time commencing at the end of the delay time and ending when achieving *b* % of the established braking force deceleration or pressure of the brake equipment (see Figure 11).

4.4.8.2.3 Response time (t_b)

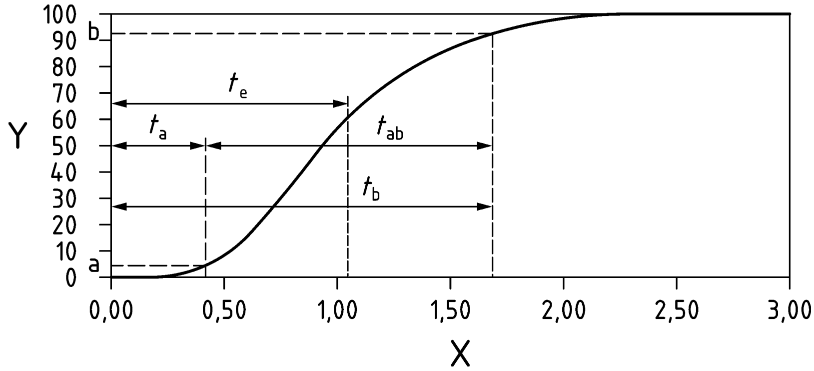
Period of time commencing when a change (positive or negative), in brake demand is initiated and ending when achieving *b* % of the established braking force, deceleration or pressure, of the brake equipment.

It can be calculated using the following formulae (see Figure 11):

For brake application:

$$t_{\mathrm{b}} = t_{\mathrm{a}} + t_{\mathrm{ab}} \tag{56}$$

NOTE Formula (56) can be used for each brake equipment type (i) using $t_{a,i}$ and $t_{\mathrm{ab,i}}$ to calculate a mean response time $\bar{t}_{\mathrm{b,i}}$ related to the whole train when considering the rate of propagation of the command signals for the active brake equipment type (i).



Key

Y	factor of nominal braking force, deceleration or pressure (in the case of pressure systems), in %
X	time, in s
t_a	delay time, in s
t_{ab}	brake build-up time, in s
t_b	response time, in s
a	is employed for the commencement of braking
b	point when the brake force, deceleration or pressure has been substantially achieved
t_e	equivalent response time, in s

Figure 11 — Delay and build-up time for brake application

Usually, the time characteristic is considered for each brake equipment type (i) when the brake force of this equipment becomes greater than zero.

For a brake application after reaching t_b no time characteristic should be considered for any subsequent change in brake force demand.

4.4.8.3 Equivalent response time (t_e)

This is the theoretical response time used to calculate stopping and slowing distance.

It is normally assumed to correspond to a model in which:

— the first part of the time is a 'free run' time with theoretical force equal to zero;

— the second part of the time is a full force run time.

Under typical circumstances (normally the vehicle stops with fully applied brakes after the response time t_b) Formula (57) is used to calculate the equivalent response time ($t_{e,i}$) of each brake equipment type.

$$t_{\mathrm{e,i}} = t_{\mathrm{a,i}} + \frac{t_{\mathrm{ab,i}}}{2} \tag{57}$$

When considering a linear function for the build-up phase Formula (58) may be used particularly on level track with low initial speeds and also if the response time $t_{b,i}$ for the brake equipment type is greater than 20 % of $\frac{v_0 - v_{fin}}{a_e}$

$$t_{\mathrm{e,i}} = t_{\mathrm{a,i}} + \frac{t_{\mathrm{ab,i}}}{2}\left(1 - \frac{a_{\mathrm{e,i}} \cdot t_{\mathrm{ab,i}}}{12 \cdot v_0}\right) \tag{58}$$

where

$t_{\mathrm{a,i}}$	delay time of brake equipment type i, in s
$t_{\mathrm{ab,i}}$	brake build-up time of brake equipment type i, in s
$t_{\mathrm{e,i}}$	equivalent response time of brake equipment type i, in s
$a_{\mathrm{e,i}}$	equivalent deceleration (see 4.8.3), in m/s^2
v_0	initial speed, in m/s

For units or train sets with a combination of different brake equipment types it is necessary to consider system build-up times which are not pressure systems, e.g. dynamic brake, magnetic track brake, eddy current brake. The assumption here is the build-up time to *b* % (refer to Figure 11).

It is assumed that the calculated equivalent response time t_e is a collective value accounting for different response times and/or different braking forces applicable to the whole train, considering that the brake demand on each brake equipment type is simultaneous.

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

NOTE The equivalent train response time t_e can also be determined on the basis of the train deceleration curve issued from dynamic braking tests of the train.

The equivalent response time of the train t_e, is calculated using Formula (59) and considering the total number of each brake equipment types at the entity level being considered e.g. total number of bogies fitted with each brake equipment type.

$$t_\mathrm{e} = \frac{\sum (t_{\mathrm{e,i}} \cdot \overline{F_{\mathrm{B,i}}})}{\sum \overline{F_{\mathrm{B,i}}}} \tag{59}$$

where:

$\overline{F_{\mathrm{B,i}}}$	mean braking force of brake equipment type i, in N (see 4.8.2)
$t_{\mathrm{e,i}}$	brake equipment type i specific equivalent response time, in s

In cases of a long train formation operating in the 'G', mode it may be necessary to consider an equivalent response time which cannot be simplified to a step function as assumed in Figure 11. In this case a 'ramp' build up instead of a step should be considered. Annex F shows the formulae used by French Railways in accordance with French National Rules for these circumstances.

4.4.9 Blending concept

A blending concept is required when it is intended that the braking force achieved by one brake equipment type is to replace or supplement the braking force provided by another brake equipment type, in order to achieve the desired performance. Normally the concept is to maximize the use of those brakes that do not wear (e.g. the dynamic brakes) and minimize the use of the friction brake (which is subject to wear). The application of the blending concept is specific to the requirements of the vehicle or train type.

A typical example of two blended brake equipment types is the electrodynamic brake and the friction brake. Figure 12 shows a typical braking force versus speed characteristic for a friction brake and an electrodynamic brake, the characteristic of which achieves the full brake demand by a combination of the instantaneous friction and dynamic brake forces.

Individual calculations shall be conducted for each brake equipment type which is to be blended with another brake equipment type, and it should be ensured that in all cases the total brake force achievable from the blended brake equipment types, meets the demanded brake force.

There is not a general blending concept. The concept in normal mode and degraded modes should be designed specifically for each project. The blending formula should be applied, depending on the project, to the applicable entity, an example of a typical formula is shown in Figure 12:

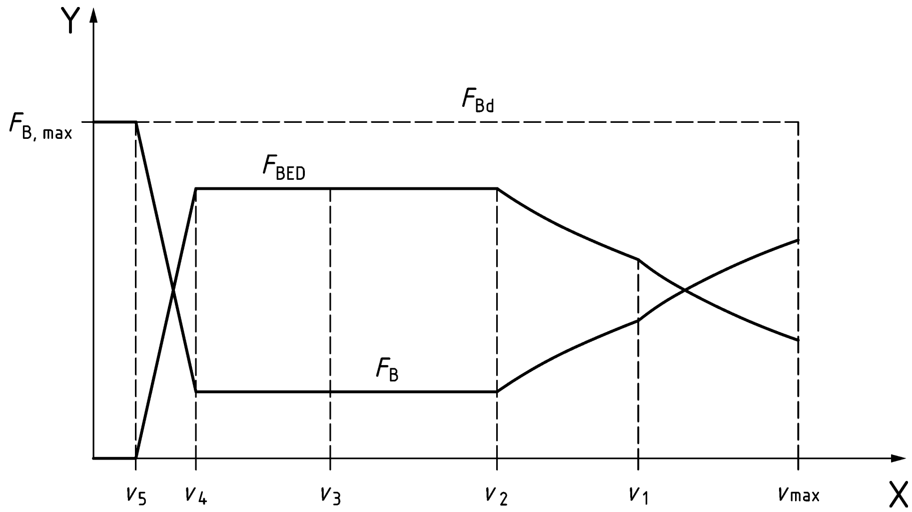


Figure 12 — Example of a blending concept between the electrodynamic brake and the friction brake versus speed

The basic calculation to achieve this concept is given by Formula (60) with reference to the above:

$$F_{\mathrm{Bd}} = F_{\mathrm{BED}} + F_{\mathrm{B}} \tag{60}$$

where:

F_{B}	instantaneous friction brake force, in N
F_{Bd}	brake force demand, in N
F_{BED}	instantaneous electrodynamic braking force, in N
v	instantaneous speed, in m/s

$v_1 \dots v_5$	particular speeds, in m/s
v_{max}	maximum operational speed, in m/s
X	axis of speed, in m/s
Y	axis of braking force, in N

$$F_B = \min\left(F_{B,max}; \left(F_{Bd} - F_{BED}\right)\right) \quad (61)$$

where:

F_B	instantaneous friction brake force, in N
$F_{B,max}$	maximum friction brake force available, in N
F_{Bd}	brake force demand, in N
F_{BED}	instantaneous electrodynamic braking force, in N

4.4.10 Sharing, proportioning of the brake forces - achieved forces

Where brake blending is not required but different brake equipment types are used, the achieved forces should be considered separately and in combination to achieve the overall brake performance throughout the speed range.

4.5 Initial and operating characteristics

4.5.1 Gradient of the track

In general, brake performance calculations are based on the assumption of straight and level track.

If it is required to consider the effect of a gradient this is normally assumed to be a constant value throughout the stopping or slowing distance.

The gradient is defined by:

$$i = \tan\alpha \quad (62)$$

with α angle of gradient

$$\sin\alpha = \frac{i}{\sqrt{i^2 + 1}} \quad (63)$$

and

$$\cos\alpha = \frac{1}{\sqrt{i^2 + 1}} \quad (64)$$

The effect of the gradient is:

$$F_g = \frac{m_{st} \cdot g_n \cdot i}{\sqrt{i^2 + 1}} \quad (65)$$

For calculation of external forces that result from gradients in railway applications the following simplification is commonly used:

$$\sin\alpha \approx \tan\alpha \text{ and} \quad (66)$$

$$\cos\alpha \approx 1$$

The gradient force is therefore:

$$F_{\mathrm{g}} = m_{\mathrm{st}} \cdot g_n \cdot i \tag{67}$$

where:

F_{g}	downhill force on the train, in N
m_{st}	static mass of the train, in kg
g_n	standard acceleration of free fall, in m/s^2
i	gradient (rising gradient is positive)

This simplification creates an error of nominally 1 % at a gradient i = 0,08. For steeper gradients it is recommended that Formula (66) should be used.

NOTE The value of i is dimensionless and may be expressed for example as $^0/_{00}$ or 1 in 'x'.

4.5.2 Initial speed

For design the calculations should be performed with different initial speeds throughout the speed range up to and including the maximum speed.

4.5.3 Coefficient of adhesion

4.5.3.1 Maximum transmittable braking force

If the demanded coefficient of adhesion exceeds the available adhesion, it can lead to an increase of the stopping distance as a consequence of a sliding wheelset or regulation by the wheel slide protection device (if fitted).

In general when a specific value of the coefficient of available adhesion is indicated, the braking force is limited to the maximum braking force that can be transmitted without sliding. This is calculated as follows:

$$F_{\mathrm{B,max,ax}} = m_{\mathrm{dyn,ax}} \cdot \tau_{\mathrm{a}} \cdot g_n \tag{68}$$

where

$F_{\mathrm{B,max,ax}}$	maximum braking force per axle, in N
$m_{\mathrm{dyn,ax}}$	dynamic mass per axle, in kg
τ_{a}	coefficient of available adhesion
g_n	standard acceleration of free fall, in m/s^2

The required coefficient of adhesion of each axle calculated in accordance with 4.5.3.2 shall be lower than any specified coefficient of available adhesion. The coefficient of available adhesion is dependent on the conditions of the wheel/track interface; this for example is affected by the use of sanding, train length, speed, environmental conditions, number of axles, etc.

4.5.3.2 Required mean adhesion value for each braked axle ($\tau_{req,ax}$)

The required mean coefficient of adhesion for each braked axle shall be calculated as follows:

$$\overline{\tau}_{\text{req,ax}} = \frac{\left(\sum_{\text{ax}} \overline{F_{\text{B,i}}} - m_{\text{rot,ax}} \cdot a_{\text{e}}\right)}{m_{\text{st,ax}} \cdot g_n} \cdot \sqrt{1+i^2} \tag{69}$$

where:

$\overline{\tau}_{\text{req,ax}}$	required adhesion value for the axle
Σ_{ax}	sum of all involved adhesion requiring brake equipment types per axle
$\overline{F_{\text{B,i}}}$	mean braking force of brake equipment type i (see 4.8.2), in N
a_{e}	total equivalent deceleration (see 4.8.3), in m/s^2
g_n	standard acceleration of free fall, in m/s^2
i	gradient
$m_{\text{rot,ax}}$	rotational mass of the axle, in kg
$m_{\text{st,ax}}$	static mass acting on the axle, in kg

NOTE 1 For low gradients (see 4.5.1) the influence of the gradient can be neglected.

NOTE 2 The value of coefficient of adhesion calculated here is that required for a mean brake force calculation and is not the maximum required. If the maximum adhesion demand is required refer to the instantaneous calculation in EN 14531-2.

4.5.4 Level of the brake demand

Generally, only the emergency brake demand is considered during calculations (unless otherwise specified).

Other brake demand levels, e.g. full service braking, may be considered when establishing the design of each brake equipment type.

4.5.5 Quantity of each brake equipment type available

Calculations shall be performed with all the brake equipment in working order and with a specified quantity/location of isolated brakes as applicable.

4.5.6 Calculation in degraded conditions

Brake calculations shall be performed with nominal parameters of the brake equipment in working order. As required calculations shall be conducted considering degraded mode conditions affecting the performances of the brakes, like friction coefficient, available coefficient of adhesion, or isolated equipment.

4.6 Total decelerating force at train level

This is the summation of the acting forces on the train provided by:

— each brake equipment type;

— the rotational inertia forces;

— the external forces.

External decelerating forces are considered as being the forces due to rolling resistance, rising gradient, head wind, etc. It should be noted that external forces can also provide accelerating effect in certain instances e.g. falling gradient, following wind, etc.

4.7 External forces

4.7.1 Gradient

To obtain the downhill force the formulae as given in 4.5.1 are used.

4.7.2 Wind force on the train

This force can be approximated using the term '*C*' of the train resistance formula given in 4.3.2.6 (see Formula (3)).

In this case, the wind force on the train is corrected to take into account the direction of the wind that gives the maximum force.

Generally, the wind force is directly specified if the designer has to take it into account:

$$F_{\text{wind}} = D \cdot C \cdot v_{\text{wind}}{}^2 \tag{70}$$

where

F_{wind} wind force on the train, in N

D characteristic coefficient of the train aerodynamic due to the direction of the wind

C characteristic coefficient of the aerodynamic train resistance, in $\frac{\text{N}}{(\text{m/s})^2}$

v_{wind} speed of the wind, in m/s

4.7.3 Train resistance

This force corresponds approximately to the term '*A*' of the train resistance formula (see 4.3.2.6), considering the train at standstill. It may be used when the effect of wind force on the train is taken into consideration.

$$F_{\text{Ra}} = A \tag{71}$$

where

F_{Ra} train resistance, in N

A characteristic coefficient of the train independent of speed, in N

4.8 Stopping and slowing distance calculation based on mean values

4.8.1 General

This calculation considers the stopping and slowing distance which is achieved when mean values of braking force are considered. In addition the equivalent deceleration due to the mean braking forces and mean external forces is considered.

4.8.2 Mean braking force with respect to the distance

The braking forces used for the calculation of braking distance shall be the mean braking forces.

The mean braking force of the brake equipment is calculated using Formula (72) as applied to the brake equipment types described previously in this standard:

$$\overline{F_{\mathrm{B,i}}} = \frac{v_0^2 - v_{fin}^2}{2} \cdot 1 \Bigg/ \int_{v_{\mathrm{fin}}}^{v_0} \frac{v}{F_{\mathrm{B,i}}} dv \tag{72}$$

where

$\overline{F_{\mathrm{B,i}}}$	mean braking force of brake type i, in N
v_0	initial speed, in m/s
v_{fin}	final speed (= 0 in the case of a stopping distance), in m/s
v	speed, in m/s
$F_{\mathrm{B,i}}$	instantaneous friction brake force for brake equipment type i, in N

NOTE This calculation is conducted for each brake equipment type if there is more than one (refer to Annex E).

If the mathematical description of the instantaneous braking force is separately defined in different speed ranges, the integral shall be calculated for each range of speed and finally cumulated. This results in the mean braking force being obtained for the speed range from the specific initial to the specific final speed. If the instantaneous braking force is defined as a constant force the mean braking force is equal to this and does not depend on the speed range.

4.8.3 Equivalent deceleration (a_e) based on mean forces

The equivalent deceleration is equal to a mean deceleration with respect to the distance during braking in a specific speed range and is applicable to the whole train. Therefore it is valid for the speed range of the mean braking force calculation.

The equivalent deceleration a_e is based on a calculation with fully applied brake forces for all functioning brake equipment types.

$$a_{\mathrm{e}} = \frac{\sum \overline{F_{\mathrm{B,i}}} + \sum \overline{F_{\mathrm{ext}}}}{m_{\mathrm{dyn}}} \tag{73}$$

where:

a_e equivalent deceleration, in m/s²

$\overline{F_{\mathrm{B,i}}}$ mean braking force of brake type i, in N

$\overline{F_{ext}}$ mean external force, in N

m_{dyn} dynamic mass of the train $(= m_{st} + m_{rot})$, in kg

This calculation sums the forces from each functioning brake equipment type plus external forces and is subject to the blending rules.

4.8.4 Mean decelerations supplied by each braking force ($\overline{a_i}$)

The mean deceleration $\overline{a_i}$ supplied by each brake type i is calculated from Formula (74):

$$\overline{a_i} = \frac{\overline{F_{\mathrm{B,i}}}}{m_{\mathrm{dyn}}} \tag{74}$$

where:

$\overline{a_i}$ mean deceleration supplied by the brake type i, in m/s²

$\overline{F_{B,i}}$ mean braking force of brake type i, in N

m_{dyn} dynamic mass of the train (= m_{st} + m_{rot}), in kg

This calculation considers the proportion of the train deceleration provided by the specific brake equipment type for a defined entity, e.g. train, vehicle, bogie or axle while that brake equipment type is functioning. The mass considered is the dynamic mass of the train as applicable to the entity being considered.

Initially it is assumed that the required adhesion is available when adhesion dependent brake equipment types are being considered. The required mean adhesion value should be considered as set out in 4.5.3.2.

4.8.5 Equivalent free run distance (s_0)

The equivalent free run distance s_0 is a theoretical distance without deceleration or acceleration. It is calculated using Formula (75) assuming level track:

$$s_0 = v_0 \cdot t_{\mathrm{e}} \tag{75}$$

where:

s_0 equivalent free run distance, in m

t_e equivalent response time, in s

v_0 initial speed, in m/s

4.8.6 Stopping and slowing distance on level track (s)

The stopping or slowing distance is defined as the distance run between the initial brake demand and achieving the final speed.

The following formula can be used:

$$s = v_0 \cdot t_\mathrm{e} + \frac{v_0^2 - v_\mathrm{fin}^2}{2 \cdot a_\mathrm{e}} \tag{76}$$

where:

s	stopping/slowing distance, in m
t_e	total equivalent response time, in s
v_0	initial speed, in m/s
v_{fin}	final speed (= 0 in the case of a stopping distance), in m/s
a_e	equivalent deceleration, in m/s²

In some circumstances there might be a requirement to change the brake force due to speed entering a different speed range e.g. above 250 km/h. In this case the stopping or slowing distance calculation should account for this change as follows:

$$s = v_0 \cdot t_\mathrm{e} + \frac{{v_0}^2 - {v_\mathrm{j}}^2}{2 \cdot a_\mathrm{e,j}} + \frac{{v_\mathrm{j}}^2 - {v_\mathrm{j+1}}^2}{2 \cdot a_\mathrm{e,j+1}} + \ldots + \frac{{v_{n-1}}^2 - {v_n}^2}{2 \cdot a_\mathrm{e,n}} \tag{77}$$

where:

$a_\mathrm{e,j}$	equivalent deceleration supplied within speed range *i*, in m/s²
j	number of speed range step

4.8.7 Stopping and slowing distance on a gradient (s_{grad})

4.8.7.1 Direct calculation of s_{grad} based on constant deceleration - approximation

Formula (76) may be used as an approximation for the calculation of the stopping distance on a gradient.

$$s_{grad} = v_0 \cdot t_\mathrm{e} + \frac{v_0^2 - v_\mathrm{fin}^2}{2 \cdot a_\mathrm{e}} \tag{78}$$

where:

s_{grad}	stopping/slowing distance, in m
t_e	total equivalent response time (see Formula (59)), in s
v_0	initial speed, in m/s
v_{fin}	final speed (= 0 in the case of a stopping distance), in m/s

a_e equivalent deceleration including the effect of gradient, in m/s²

It is recommended that Formula (78) is only used when considering a maximum gradient of not more than 0,01 or the initial speed is not less than 50 km/h or the equivalent response time t_e of the train brake is not more than 3 s. With these limits the expected error from the use of this formula is < 5 %.

4.8.7.2 Direct calculation of s_{grad} based on constant deceleration

If it is required to be more accurate, or to use different values for the gradient or for the speed, it is necessary to consider during the response time, the effect of the gradient on the speed. Then Formula (79) should be applied.

$$s_{\mathrm{grad}} = v_0 \cdot t_\mathrm{e} - \frac{1}{2}\frac{m_{\mathrm{st}}}{m_{\mathrm{dyn}}} \cdot g_n \cdot i \cdot t_\mathrm{e}^2 + \frac{\left(v_0 - \frac{m_{\mathrm{st}}}{m_{\mathrm{dyn}}} \cdot g_n \cdot i \cdot t_\mathrm{e}\right)^2 - v_{fin}{}^2}{2 \cdot a_e} \tag{79}$$

where:

s_{grad} stopping distance on a gradient, in m

s stopping distance without gradient, in m

t_e total equivalent response time, in s

v_0 initial speed, in m/s

m_{st} static mass, in kg

m_{dyn} dynamic mass, in kg

g_n standard acceleration of free fall, in m/s²

i gradient of the track

a_e equivalent deceleration including the effect of gradient, in m/s²

NOTE Both Formula (78) and (79) assume sin α approximately tan α = *i* (refer to Formula (66))

4.8.8 Other specific formulae for the calculation of stopping distance

In the case of long trains operating in the brake mode 'G' other specific methods of calculation can be used. Refer to Annex F which shows the formulae used by French Railways in accordance with French National Rules; this also includes braking on a gradient.

4.9 Supplementary dynamic calculations

4.9.1 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.

4.9.2 Mass to be braked (m_B)

The mass to be braked can be related to the specific brake type or to individual entities i.e. mass to be braked per axle, per bogie, etc. It is the ratio of its mean braking force to the equivalent deceleration of the train. The mass to be braked is calculated using the following formula:

For a brake equipment type:

$$m_{\mathrm{B,i}} = \frac{\overline{F_{\mathrm{B,i}}}}{a_{\mathrm{e}}} \tag{80}$$

For an entity

$$m_{\mathrm{ent}} = \frac{\overline{F_{\mathrm{ent}}}}{a_{\mathrm{e}}} \tag{81}$$

where:

$\overline{F_{\mathrm{B,i}}}$	mean braking force of brake type i, in N
$\overline{F_{\mathrm{ent}}}$	mean braking force of entity (ent), in N
a_{e}	equivalent deceleration, in m/s^2

The mass to be braked can be used, for example, in the consideration of train brake energy dissipation.

4.9.3 Braking energy

4.9.3.1 Total energy (W_{tot})

The 'total energy' is the sum of the dissipated energy of all applied brake equipment types and train resistance as set out in 4.3.2.6, which is equal to the related difference of kinetic and potential energy. It is given by:

$$W_{\mathrm{tot}} = \frac{m_{\mathrm{dyn}} \cdot (v_0{}^2 - v_{\mathrm{fin}}{}^2)}{2} - m_{\mathrm{st}} \cdot g_n \cdot s \cdot \frac{i}{\sqrt{i^2 + 1}} = W_{\mathrm{B}} + W_{\mathrm{Ra}} \tag{82}$$

where:

W_{tot}	total energy, in J
W_{B}	energy dissipated by the brake systems, in J
W_{Ra}	energy dissipated by the train resistance, in J
g_n	standard acceleration of free fall, in m/s^2
i	gradient
m_{dyn}	dynamic mass of the train, in kg
m_{st}	static mass of the train, in kg
v_0	initial speed, in m/s
v_{fin}	final speed, in m/s
s	distance between v_0 and v_{fin}, in m

NOTE 1 In Formula (82) the potential energy is represented by $s \cdot \frac{i}{\sqrt{i^2+1}}$ i.e. the height difference between the start of the brake control and the end of the stopping or slowing distance.

NOTE 2 Typically this formula is used for a stopping calculation where v_{fin} is equal to 0.

4.9.3.2 Energy dissipated by each brake equipment type

The energy dissipated by each brake equipment type is calculated for mean (constant) braking forces as follows:

$$W_{\mathrm{B,i}} = \overline{F_{\mathrm{B,i}}} \cdot (s - s_0) \tag{83}$$

For energy dissipated using instantaneous braking forces see EN 14531-2:

where:

$W_{\mathrm{B,i}}$	energy dissipated by brake equipment type i, in J
$\overline{F_{\mathrm{B,i}}}$	mean braking force for brake equipment type i, in N
s_0	equivalent free run distance, in m
s	stopping/slowing distance, in m

4.9.3.3 Specific energy dissipated by each type of friction brake

The dissipated brake energy per friction area of each type of friction brake may be calculated as follows:

$$W_{\mathrm{S,i}} = \frac{\overline{W_{\mathrm{B,i}}}}{n \cdot A_{\mathrm{S,i}}} \tag{84}$$

where:

$W_{\mathrm{S,i}}$	energy dissipated by friction brake type i per unit area, in J/m^2

$\overline{W_{\mathrm{B,i}}}$	energy dissipated by friction brake type i, in J
$A_{\mathrm{S,i}}$	swept area of friction surface of the brake system type i, in m² (see Figures 3 and 5 for examples of a tread and disc brake.)
n	number of friction surfaces

4.9.4 Maximum braking power of each brake equipment type

The maximum braking power of each brake equipment type i is calculated as follows when using the mean value of brake force and initial speed:

$$P_{\mathrm{max,i}} = \overline{F}_{\mathrm{B,i}} \cdot v_0 \tag{85}$$

where:

$P_{\mathrm{max,i}}$	maximum power of brake equipment type i, in W
$\overline{F}_{\mathrm{B,i}}$	mean braking force of brake equipment type i, in N
v_0	initial speed, in m/s

4.9.5 Maximum specific power flux for each type of friction brake

The maximum specific power flux for each type of friction brake shall be calculated as follows:

$$P_{\mathrm{S,max,i}} = \frac{P_{\mathrm{max,i}}}{A_{\mathrm{s,i}}} \tag{86}$$

where:

$P_{\mathrm{S,max,i}}$	maximum power flux of friction brake system i, in W/m²
$P_{\mathrm{max,i}}$	maximum power of friction brake system i, in W
$A_{\mathrm{s,i}}$	swept area of the friction surface of the disc i or the wheel tread i, in m²

4.10 Specific expressions of braking performance

4.10.1 General

The following expressions are indicators of braking performances other than deceleration. They are commonly used within European railway undertakings, authorities or organizations. The derivation of the units used for these expressions might not be consistent with their physical properties.

4.10.2 Braked weight percentage (λ)

Lambda (λ) is usually the symbol for braked weight percentage when working in accordance with UIC requirements and considering vehicle or train performance. Refer to the bibliography for relevant documents.

Lambda is dependent on the stopping distance achieved during track test; however an estimation may be obtained using the calculated distance at the relevant speeds.

4.10.3 Braked weight

The braked weight is a common type of expression for the braking performance when working in accordance with UIC requirements and considering vehicle/train performance. Refer to the bibliography for relevant documents.

NOTE The braked weight is calculated by multiplying the brake weight percentage by the static mass of the train or vehicle.

4.10.4 Braking ratio

The braking ratio is normally only used for German railways and referred to as 'Abbremsung'. This shows the ratio of the force on the application points of the friction brake to the static weight of a wheel-set or of the vehicle or train. The static weight is calculated by the multiplication of its static mass with the standard acceleration of free fall without any slope influence.

4.10.5 Equivalent brake force

The 'equivalent' brake force data for a rail vehicle is only required for GB railways. The-total 'equivalent' brake force available in a freight train is used to determine the maximum permissible speed.

The 'equivalent' brake force is the value of brake force that needs to be exerted on an 'equivalent tread brake arrangement' with a standard coefficient of friction, to produce the same value of brake decelerating force as that given by the actual combination of brake force and coefficient of friction on the vehicle. The methodology to determine the equivalent brake force data are set out in the GB notified national technical rules.

5 Immobilization brake calculation

5.1 General

The immobilization brake is used to prevent a stationary train from moving i.e. by the use of a holding or parking brake.

The parking brake and holding brake are generally provided by different brake equipment types.

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

5.2 General characteristics

The parameters to define immobilization configuration are:

— quantity of axles;

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

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

— quantity of non-adhesion dependent brake equipment types;

— static mass per axle;

— static coefficient of friction (block or pad).

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 axle shall be identified and considered.

5.3 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 immobilization brake performance.

The mean static values of the coefficient of friction shall be established using the methods as set out in prEN 15328 for brake pads or EN 16452 for brake blocks.

NOTE Typically, the immobilization (holding) brake application follows a stopping brake application, in this case the value of the static coefficient of friction can be affected.

5.4 Train and operating characteristics

Immobilization performance shall be calculated considering the following specific conditions:

— gradient;

— vehicle load;

— wheel diameter;

— wind;

— rolling resistance;

— isolated holding/parking brake equipment;

— available adhesion conditions.

NOTE Requirements for maximum gradient, vehicle load and number of isolated equipment types to be considered can be specified.

5.5 Immobilization force provided by equipment type

5.5.1 General

The immobilization requirements are typically provided by the parking brake when immobilization is required to be for an unlimited time. If immobilization is required for a limited time then an application of, for example, the service brake may be used.

Sub-clauses 5.5.2 and 5.5.3 identify the sub-clauses in Clause 4 containing the relevant formulae when considering different brake equipment types which may be used for immobilization. See Annex C for example calculations (parking brake).

5.5.2 Force of a screw hand brake (Tread brake)

In this case use the formulae as given in 4.4.4.3 to obtain the brake force used for immobilization.

5.5.3 Force of a screw hand brake (Disc brake)

In this case use the formulae as given in 4.4.5.2 to obtain the brake force used for immobilization.

5.5.4 Force of a tread brake unit

When assessing the spring parking brake, a minimum guaranteed fixed value of parking brake block force may be used. This is typically provided by the supplier. See Figure 13.

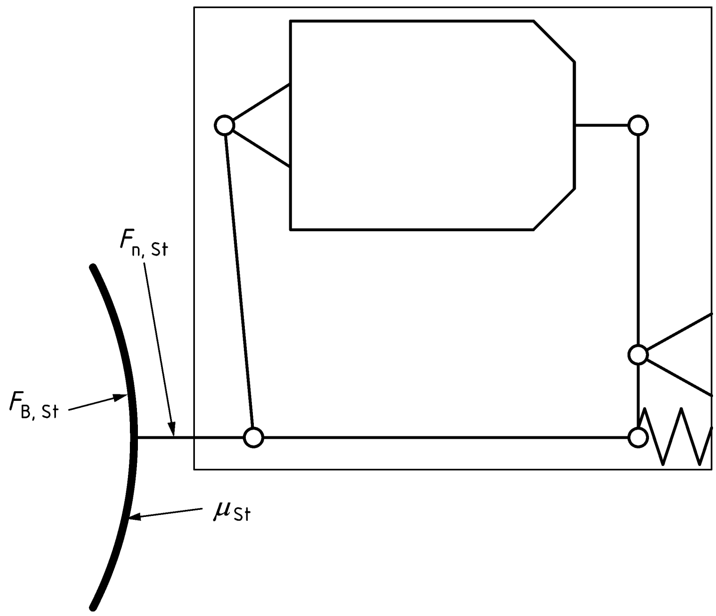


Figure 13 — Basic tread brake unit arrangement showing parking brake block force

Formula (87) is used to calculate the parking brake force related to the wheel per tread brake unit arrangement:

$$F_{B,st} = F_{n,st} \cdot \mu_{st} \quad (87)$$

where:

$F_{B,st}$	parking brake force related to the wheel, in N
$F_{n,st}$	parking brake block force, in N
μ_{ST}	brake block static coefficient of friction

5.5.5 Force of a disc brake unit arrangement

When assessing a spring parking brake, a minimum guaranteed fixed value of the resultant parking brake clamp force for the particular arrangement is used. This is typically provided by the supplier. See Figure 14.

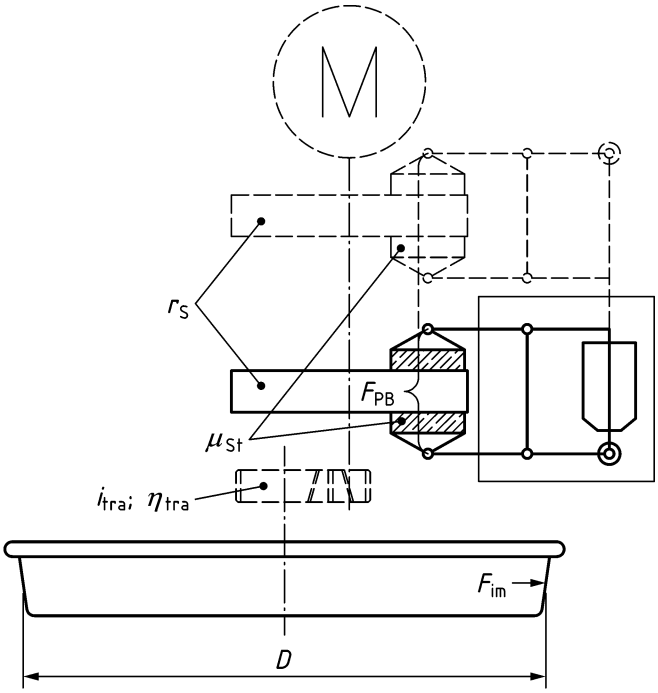


Figure 14 — Basic disc brake unit arrangement showing parking brake block force

Formula (88) is used to calculate the parking brake force related to the wheel per disc brake unit arrangement

$$F_{\text{im}} = F_{\text{PB}} \cdot \mu_{\text{ST}} \cdot \frac{r_{\text{s}}}{D/2} \cdot \frac{i_{\text{tra}}}{\eta_{\text{tra}}} \qquad (88)$$

where:

F_{PB}	parking brake clamp force, in N
μ_{ST}	brake pad static coefficient of friction
r_{s}	mean swept radius of the brake pad on the disc face, in m
D	wheel diameter, in m
i_{tra}	transmission ratio
η_{tra}	transmission efficiency

5.5.6 Force of a permanent magnetic track brake

In this case the formulae given in 4.4.7.4 are used to obtain the brake force used for immobilization.

5.6 Immobilization force for each axle

The immobilization force for each axle is the summation of the immobilization forces of the adhesion dependant brake equipment types acting on that axle.

$$F_{\mathrm{B,im,ax}} = \left(\sum_{i} F_{\mathrm{im,i}} \right)_{\mathrm{ax}} \tag{89}$$

where:

$F_{\mathrm{B,im,ax}}$ immobilization brake force on that axle, in N

$F_{\mathrm{im,i}}$ immobilization force of holding/parking brake i, in N

$\sum_{i}$ summation of all immobilization forces of holding/parking brake types i

These individual immobilization forces may possibly be limited by the available adhesion, therefore the minimum transmittable force shall be established, i.e. either the adhesion transmittable force or the immobilization brake applied force.

$$F_{\mathrm{im,ax}} = \min\left(F_{\mathrm{B,im,ax}} ; \tau_{\mathrm{a}} \cdot m_{\mathrm{st,ax}} \cdot g_{n} \cdot i\right) \tag{90}$$

where

$F_{\mathrm{im,ax}}$ transmittable immobilization force on that axle, in N

$F_{\mathrm{B,im,ax}}$ immobilization brake force on that axle, in N

g_{n} standard acceleration of free fall, in m/s^2

$m_{\mathrm{st,ax}}$ static mass per axle, in kg

τ_{a} available adhesion

i gradient

5.7 Total immobilization force per train

The immobilization force of the train is the summation of all immobilization forces, both adhesion dependent and adhesion independent brake equipment types.

$$F_{\mathrm{im}} = \sum F_{\mathrm{B,ind}} + \sum_{\mathrm{ax}} F_{\mathrm{im,ax}} \tag{91}$$

where

$\sum_{ax}$ summation of the adhesion dependant immobilization forces of all immobilization braked axles ($^{\mathrm{ax}}$) of the train

F_{im} immobilization force of the train, in N

$F_{\mathrm{im,ax}}$ adhesion dependent immobilization force per axle ($^{\mathrm{ax}}$), in N

$F_{B,ind}$ immobilization force independent of adhesion, in N

5.8 Immobilization safety factor

The ratio of the immobilization force on the complete train to the forces that would accelerate the train shall be greater than one:

$$S_{im} = \frac{F_{im} + F_{Ra}}{F_g + F_{wind}} \tag{92}$$

where

S_{im} immobilization safety factor

F_{im} immobilization force of the train, in N

F_g downhill force on the train, in N

F_{wind} wind force on the train, in N

F_{Ra} train resistance, in N

For downhill and/or wind force and train resistance forces refer to 4.7.

5.9 Coefficient of adhesion required by each braked axle

The coefficient of adhesion required by each axle is the ratio of the immobilization force of the axle to the axle load depending on the gradient. This calculation is conducted to check if the installed immobilization brake force distribution is sufficient, taking account of the available adhesion to ensure the wheelset will not slip (slipping limits).

$$\tau_{req,ax} = \frac{F_{B,im,ax}}{m_{st,ax} \cdot g_n \Big/ \sqrt{i^2 + 1}} \tag{93}$$

For simplification as set out in 4.5.1, the following formula may be used:

$$\tau_{req,ax} = \frac{F_{B,im,ax}}{m_{st,ax} \cdot g_n} \tag{94}$$

where

$\tau_{req,ax}$ coefficient of adhesion required by each immobilization braked axle

$F_{B,im,ax}$ total immobilization brake force per axle applied, in N

$m_{st,ax}$ static mass per axle, in kg

g_n standard acceleration of free fall, in m/s^2

i Gradient

NOTE This adhesion calculation is commonly used as an approximation.

5.10 Maximum achievable gradient

The maximum achievable gradient is a balanced result of the immobilization force, resistance force, wind force, and downhill force. This calculation is needed to check if the installed immobilization brake force is sufficient regarding the project specific requirements.

$$i_{\max} = \frac{1}{\sqrt{\left(\frac{m_{\mathrm{st}} \cdot g_n}{F_{\mathrm{im}} + F_{\mathrm{Ra,im}} - F_{\mathrm{wind}}}\right)^2 - 1}} \tag{95}$$

For simplification as set out in 4.5.1, the following formula may be used:

$$i_{\max} = \frac{F_{\mathrm{im}} - F_{\mathrm{wind}} + F_{\mathrm{Ra,im}}}{m_{\mathrm{st}} \cdot g_n} \tag{96}$$

where:

$i_{\max}$	maximum achievable gradient
F_{im}	immobilization force of the train, in N
$F_{\mathrm{Ra,im}}$	train resistance, in N
F_{wind}	wind force on the train, in N
m_{st}	static mass, in kg
g_n	standard acceleration of free fall, in m/s^2

NOTE The maximum gradient permitted is normally calculated without taking into account the effects of the wind and the train resistance.

Annex A
(informative)

Workflow of stopping and slowing distance calculation method

The calculation procedure is summarized in Figure A.1 and A.2 and assumes that the entity considered for the friction brake is an axle:

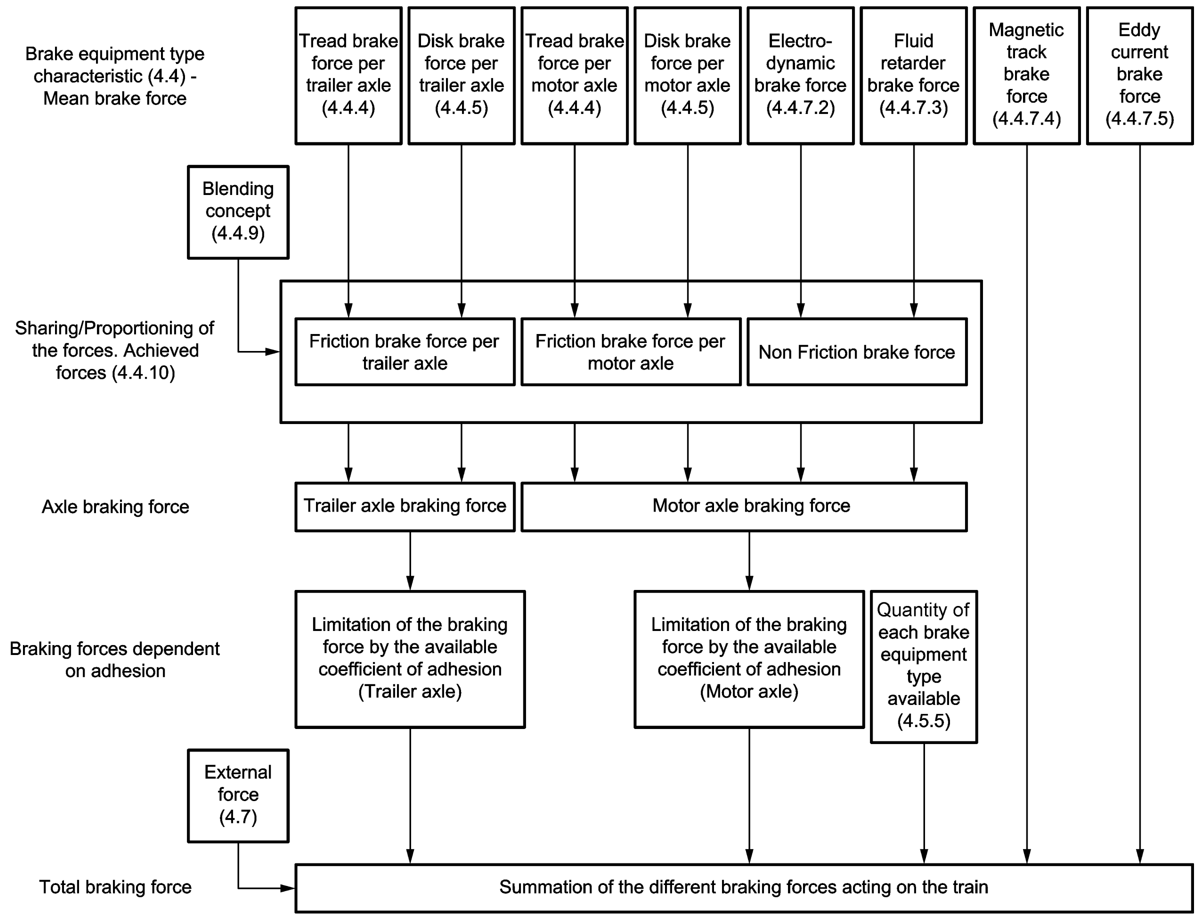


Figure A.1 — Summary of workflow to establish the brake forces acting on the train

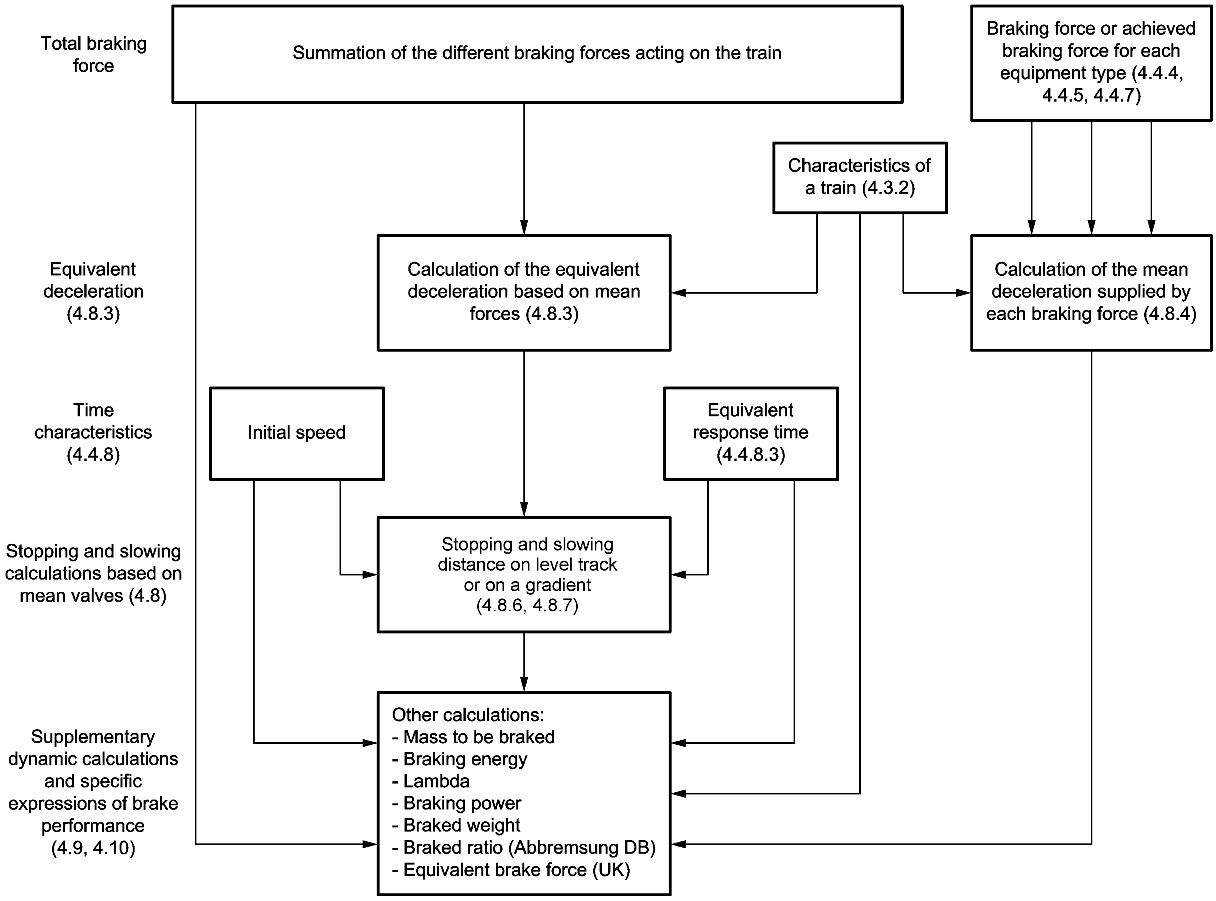


Figure A.2 — Summary of workflow to calculate the stopping or slowing distance from the brake forces

Annex B
(informative)

Workflow of immobilization calculations

The calculation procedure is summarized in Figure B.1:

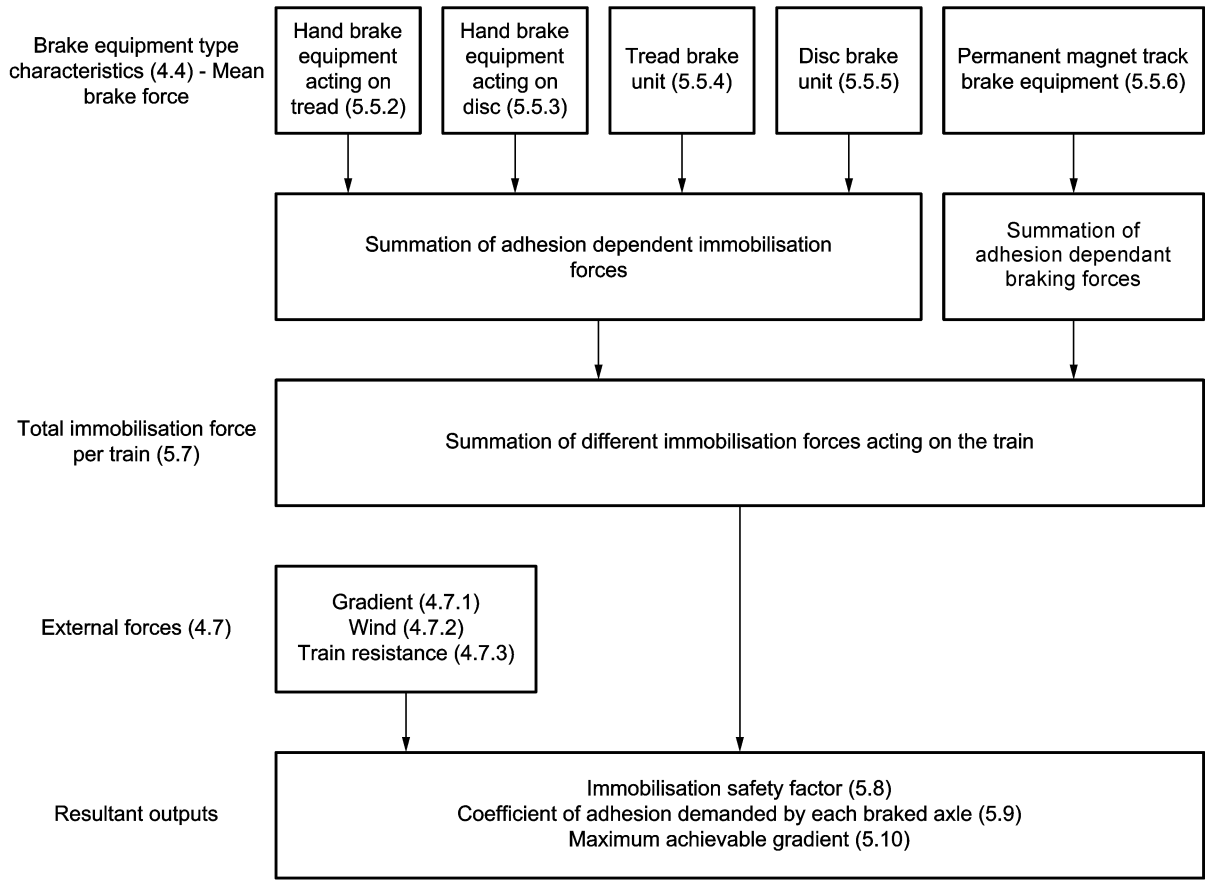
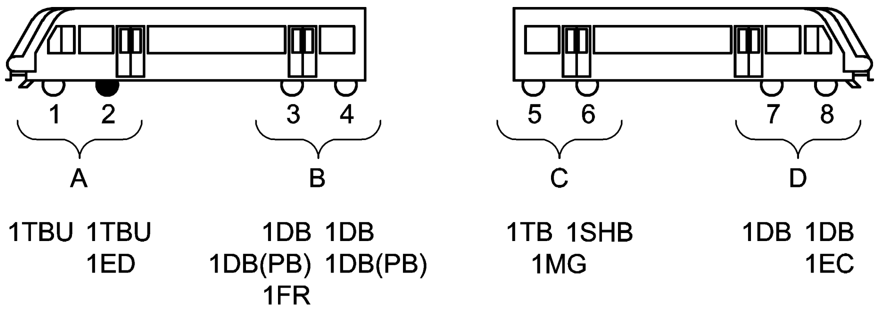


Figure B.1 — Summary of immobilization braking calculation method

Annex C
(informative)

Brake equipment type example calculations

The following are example calculations assuming different brake equipment types are used on a bogie arrangement. The brake equipment types assumed are stated for each bogie A, B, C and D and axle of a fictitious two car train as shown in Figure C.1 and described in Table C.1. The example calculations use the formulae contained in the applicable clauses in the standard and are cross referenced accordingly in Table C.1.



Key

TBU	tread brake unit	TB	tread brake rigging
ED	electrodynamic brake, motored axle	MG	magnetic track brake
DB	disc brake unit	SHB	screw hand brake
DB(PB)	disc brake unit (spring parking brake)	EC	eddy current brake
FR	fluid retarder		

Figure C.1 — Brake equipment types assumed at the entity level of a bogie

Table C.1 — Brake equipment types assumed

Bogie	Axle	Brake equipment types and clause reference
A	1 and 2	Tread brake unit see 4.4.4.1 (axle 1, 2) and electrodynamic brake see 4.4.7.2 (axle 2)
B	3 and 4	Discs brake unit (one per wheel) see 4.4.5.1 and fluid retarder see 4.4.7.3
		Disc brake unit (spring parking brake, one per wheel on one axle) see 4.4.5.1
C	5 and 6	Tread brake (rigging) see 4.4.4.2 and magnetic track brake see 4.4.7.4
		Screw hand brake see 4.4.4.3
D	7 and 8	Discs brake unit (2 discs per cylinder) see 4.4.5.1 and eddy current brake see 4.4.7.5

The values stated in Table C.2 to C.9 for each bogie are not necessarily common but are used for simplification of the worked example. All values chosen for specific applications should be verified by the user.

Example values not given as SI-unit in the following table are converted in the calculation using the factors of ISO 80000-3 and ISO 80000-4 e.g. conversion of 'bar' to 'Pa' with the factor 10^5.

EXAMPLE CALCULATION FOR BOGIE A

- WITH TREAD BRAKE UNITS AND ELECTRODYNAMIC BRAKES

Table C.2 — Table of parameters and typical values

Bogie No.	Characteristic description	Symbol	Unit	Example value
A	Pressure at brake cylinder (p_C = 3,8 bar = 3,8 *10^5 Pa)	p_C	bar	3,8
A	Cylinder area (210 mm - cylinder) (A_C = 346 cm^2 = 346 *10^{-4} m^2)	A_C	cm^2	346
A	Internal brake cylinder ratio/factor	I_C	-	1
A	Internal brake unit efficiency	η_C	-	0,95
A	Brake cylinder spring force (Return spring)	$F_{S,C}$	N	−300
A	Rigging ratio	i_{rig}	-	5
A	Dynamic rigging efficiency	$\eta_{rig,dyn}$	-	0,95
A	Rigging spring force	$F_{S,rig}$	N	−500
A	Mean friction coefficient of brake block (axle 1, 2)	μ_m	-	0,28
A	Maximum electrodynamic braking force (equals the value of the force in the constant section normally from the supplier, see Figure 7)	$F_{BED,\max}$	N	25 000
A	Initial speed	v_0	m/s	45
A	Intermediate speed (relevant to electrodynamic force, see Figure 7)	v_2	m/s	25
A	Delay time (tread brake)	t_a	s	0,6
A	Build–up time (tread brake)	t_{ab}	s	3,0
A	Delay time (electrodynamic brake)	t_a	s	0,3
A	Build–up time (electrodynamic brake)	t_{ab}	s	1,0

Table C.3 — Bogie A – Example calculations

Formula No.	Title	Formula	Value
Tread brake unit for bogie A axles 1 and 2			
4	Cylinder output force	$F_C = p_C \cdot A_C \cdot i_C \cdot \eta_C + F_{S,C}$ $F_C = 3{,}8 \cdot 10^5 \cdot 346 \cdot 10^{-4} \cdot 1 \cdot 0{,}95 - 300$	12 191 N
5	Force on the application point	$F_n = F_C \cdot i_{rig} \cdot \eta_{rig,dyn} + F_{S,rig}$ $F_n = F_C \cdot 5 \cdot 0{,}95 - 500$	57 405 N
6	Mean braking force per brake unit	$\overline{F}_{B,C} = F_n \cdot \mu_m$ $\overline{F}_{B,C} = F_n \cdot 0{,}28$	16 074 N
Electrodynamic brake for bogie A			
41	Mean electrodynamic braking force depending on the initial speed (see 4.4.7.2, NOTE)	$\overline{F_{BED}} = F_{BED,max} \cdot \left[\frac{3 \cdot v_0^2 \cdot v_2}{2 \cdot v_0^3 + v_2^3} \right]$ $\overline{F_{BED}} = 25\,000 \cdot \left[\frac{3 \cdot 45^2 \cdot 25}{2 \cdot 45^3 + 25^3} \right]$	19 188 N
Equivalent response time, Tread brake for bogie A			
57	Equivalent response time Tread brake	$t_{e,i} = t_a + \frac{t_{ab}}{2}$ $t_{e,i} = 0{,}6 + \frac{3}{2}$	2,1 s
Equivalent response time, Electrodynamic brake for bogie A			
57	Equivalent response time Electrodynamic brake	$t_{e,i} = t_a + \frac{t_{ab}}{2}$ $t_{e,i} = 0{,}3 + \frac{1}{2}$	0,8 s

EXAMPLE CALCULATION FOR BOGIE B

WITH DISC BRAKE UNIT (2 x SERVICE AND 2 x PARKING) PLUS FLUID RETARDER

Table C.4 — Table of parameters and typical values

Bogie No.	Characteristic description	Symbol	Unit	Typical value
B	Pressure at brake cylinder (p_C = 3,0 bar = 3,0 * 10^5 Pa)	p_C	bar	3,0
B	Cylinder area (203 mm - cylinder) (A_C = 324 cm^2 = 324 * 10^{-4} m^2)	A_C	cm^2	324
B	Internal brake cylinder ratio/factor	i_C	-	1
B	Internal brake unit efficiency	η_C	-	0,98
B	Brake cylinder spring force (Return spring)	$F_{S,C}$	N	−500
B	Total rigging ratio of the disc brake unit arrangement	i_{rig}	-	5
B	Dynamic rigging efficiency	$\eta_{rig,dyn}$	-	0,95
B	Mean friction coefficient of brake pad	μ_m	-	0,35
B	Mean swept radius of the brake pad on the disc face	r_S	m	0,27
B	Wheel diameter	D	m	0,89
B	Transmission ratio	i_{tra}	-	1
B	Transmission efficiency	η_{tra}	-	1
B	Maximum fluid retarder braking force (Equals the value of the force in the constant section)	$F_{BFR,\max}$	N	11 000
B	Initial speed	v_0	m/s	45
B	Intermediate speed (relevant to fluid retarder force, see Figure 8)	v_1	m/s	20
B	Delay time (disc brake)	t_a	s	0,3

Bogie No.	Characteristic description	*Symbol*	Unit	Typical value
B	Build–up time (disc brake)	t_{ab}	*s*	2,0
B	Delay time (fluid retarder)	t_a	*s*	0,6
B	Build–up time (fluid retarder)	t_{ab}	*s*	4,0
B	Mean static friction coefficient of brake pad	μ_{ST}	-	0,3
B	Number of brake disc arrangements (one disc per cylinder) per bogie	n_{disc}	-	4
B	Number of brake disc arrangements with parking brake (for bogie B)	$n_{disc,im}$	-	2

Table C.5 — Bogie B – Example calculations

Formula No.	Title	Formula	Value
Disc brake unit on each wheel of axles 3 and 4			
24	Cylinder output force	$F_C = p_C \cdot A_C \cdot i_C \cdot \eta_C + F_{S,C}$ $F_C = 3{,}0 \cdot 10^5 \cdot 324 \cdot 10^{-4} \cdot 1 \cdot 0{,}98 - 500$	9 026 N
25	Total clamp force per disc brake unit arrangement	$F_b = F_C \cdot i_{rig} \cdot \eta_{rig,dyn}$ $F_b = F_C \cdot 5 \cdot 0{,}95$	42 872 N
29	Mean braking force per disc brake unit arrangement	$\overline{F}_{B,C} = F_b \cdot \mu_m \cdot \frac{r_s}{D/2} \cdot \frac{i_{tra}}{\eta_{tra}}$ $\overline{F}_{B,C} = F_b \cdot 0{,}35 \cdot \frac{0{,}27}{0{,}89/2} \cdot \frac{1}{1}$	9 104 N
Fluid retarder for bogie B (normally not used during emergency braking)			
47	Mean fluid retarder braking force depending on the initial speed ($v_1 < v_0 \le v_{max}$)	$\overline{F_{BFR}} = F_{BFR,max} \cdot \left[\frac{3 \cdot v_0^2 \cdot v_1}{2 \cdot v_0^3 + v_1^3}\right]$ $\overline{F_{BFR}} = 11\,000 \cdot \left[\frac{3 \cdot 45^2 \cdot 20}{2 \cdot 45^3 + 20^3}\right]$	7 025 N
Equivalent response time, Disc brake for bogie B			
57	Equivalent response time Disc brake	$t_{e,i} = t_a + \frac{t_{ab}}{2}$ $t_{e,i} = 0{,}3 + \frac{2}{2}$	1,3 s
Equivalent response time, Fluid retarder for bogie B			

57	Equivalent response time Fluid retarder	$t_{e,i} = t_a + \frac{t_{ab}}{2}$ $t_{e,i} = 0{,}6 + \frac{4}{2}$	2,6 s
Parking brake force			
	Parking brake clamp force – fixed value provided by supplier		40 572 N
88	Parking brake force related to the wheel per disc brake unit arrangement	$F_{im} = F_{PB} \cdot \mu_{ST} \cdot \frac{r_s}{D/2} \cdot \frac{i_{tra}}{\eta_{tra}}$ $F_{im} = 40\,572 \cdot 0{,}3 \cdot \frac{0{,}27}{0{,}89/2} \cdot \frac{1}{1}$	7 385 N
	Total parking brake force per bogie	$F_{im,bog} = F_{im,disc} \cdot n_{disc,im}$ $F_{im,bog} = 7\,385 \cdot 2$	14 770 N

EXAMPLE CALCULATION FOR BOGIE C

WITH TREAD BRAKE (RIGGING) PLUS MAGNETIC TRACK BRAKE PLUS SCREW HAND BRAKE

Table C.6 — Table of parameters and typical values

Bogie No.	Characteristic description	Symbol	Unit	Typical value
C	Pressure at brake cylinder (p_C = 3,0 bar = 3,0 * 10^5 Pa)	p_C	bar	3,0
C	Cylinder area (203 mm - cylinder) (A_C = 324 cm^2 = 324 * 10^{-4} m^2)	A_C	cm^2	324
C	Internal brake cylinder ratio/factor	i_C	-	1
C	Internal brake unit efficiency	η_C	-	0,98
C	Brake cylinder spring force (Return spring)	$F_{S,C}$	N	−200
C	Lever length *a* of the central rigging	l_a	m	0,515
C	Lever length *b* of the central rigging	l_b	m	0,325
C	Lever length *a* of the axle rigging	$l_{a,ax}$	m	0,25
C	Lever length *b* of the axle rigging	$l_{b,ax}$	m	0,2
C	Number of braked axles	n_{ax}	-	2
C	Number of force application points per wheel	n_{ap}	-	2
C	Slack adjuster force (external rigging)	$F_{S,R}$	N	−2 000
C	Overall rigging efficiency	η_R	-	0,83
C	Mean friction coefficient of brake block	μ_m	-	0,25
C	Total magnetic attraction force (typically from two magnets)	F_{AMG}	N	90 000
C	Constant coefficient	k_0	-	2
C	Constant coefficient	k_1	s/m	0,18
C	Initial speed	v_0	m/s	45
C	Cut off speed of the magnetic track brake	v_1	m/s	6
C	Delay time (tread brake (rigging))	t_a	s	1,0
C	Build–up time (tread brake (rigging))	t_{ab}	s	5,0
C	Delay time (magnetic track brake)	t_a	s	1,5

C	Build–up time (magnetic track brake)	t_{ab}	s	0,4
C	Dynamic rigging efficiency	$\eta_{rig,dyn}$	-	0,95
C	Rigging return spring force	$F_{S,rig}$	N	–500
C	Crank handle force	$F_{Cr,H}$	N	500
C	Gear efficiency	η_G	-	0,19
C	Gear ratio	i_G	-	1 000
C	Mean static friction coefficient of brake block	μ_{st}	-	0,2

Table C.7 — Bogie C – Example calculations

Formula No.	Title	Formula	Value
Tread brake rigging for bogie C axles 5 and 6			
8	Output cylinder force	$F_C = p_C \cdot A_C \cdot i_C \cdot \eta_C + F_{S,C}$ $F_C = 3 \cdot 10^5 \cdot 324 \cdot 10^{-4} \cdot 1 \cdot 0{,}98 - 200$	9 326 N
10	Central brake rigging ratio	$i_{rig} = l_a / l_b$ $i_{rig} = 0{,}515 / 0{,}325$	1,58
12	Axle rigging ratio per application point	$i_{rig,ax} = l_{a,ax} / l_{b,ax}$ $i_{rig,ax} = 0{,}25 / 0{,}2$	1,25
11	Overall axle rigging ratio	$i_R = n_{ax} \cdot n_{ap} \cdot i_{rig,ax}$ $i_R = 2 \cdot 2 \cdot 1{,}25$	5
9	Total force on the application points	$F_b = \left(F_C \cdot i_{rig} + F_{S,R}\right) \cdot i_R \cdot \eta_R$ $F_b = (9\,326 \cdot 1{,}58 - 2\,000) \cdot 5 \cdot 0{,}83$	52 851 N
13	Mean braking force per brake equipment type	$\overline{F}_{B,C} = F_b \cdot \mu_m$ $\overline{F}_{B,C} = 52\,851 \cdot 0{,}25$	13 213 N
Screw parking brake calculation			
16	Gear force	$F_G = F_{Cr,H} \cdot i_G \cdot \eta_G + F_{S,C}$ $F_G = 500 \cdot 1\,000 \cdot 0{,}19 - 200$	94 800 N
17	Total force of all application points	$F_b = \left(F_G \cdot i_{rig} + F_{S,R}\right) \cdot i_R \cdot \eta_R$ $F_b = (94\,800 \cdot 1{,}58 - 2\,000) \cdot 5 \cdot 0{,}83$	613 304 N
18	Central brake rigging ratio	$i_{rig} = l_a / l_b$	1,58

		$i_{\mathrm{rig}} = 0{,}515 / 0{,}325$	
19	Overall axle rigging ratio	$i_{\mathrm{R}} = n_{\mathrm{ax}} \cdot n_{\mathrm{ap}} \cdot i_{\mathrm{rig,ax}}$ $i_{\mathrm{R}} = 2 \cdot 2 \cdot 1{,}25$	5
20	Axle rigging ratio per application point	$i_{\mathrm{rig,ax}} = l_{\mathrm{a,ax}} / l_{\mathrm{b,ax}}$ $i_{\mathrm{rig,ax}} = 0{,}25 / 0{,}2$	1,25
22	Force per application point	$F_n = \frac{F_{\mathrm{b}}}{2 \cdot n_{\mathrm{ax}} \cdot n_{\mathrm{ap}}}$ $F_n = \frac{613\,304}{2 \cdot 2 \cdot 2}$	76 663 N
21	Total braking force per equipment	$F_{\mathrm{B,Cr,H}} = F_{\mathrm{b}} \cdot \mu_{\mathrm{st}}$ $F_{\mathrm{B,Cr,H}} = 613\,304 \cdot 0{,}2$	122 661 N
Magnetic track brake for bogie C			
49	Mean magnetic track braking force	$\overline{F_{\mathrm{BMG}}} = F_{\mathrm{AMG}} \cdot \frac{v_0^2 - v_1^2}{\frac{2}{3} \cdot k_1 \cdot (v_0^3 - v_1^3) + k_0 \cdot (v_0^2 - v_1^2)}$ $\overline{F_{BMG}} =$ $90\,000 \cdot \frac{45^2 - 6^2}{\frac{2}{3} \cdot 0{,}18 \cdot (45^3 - 6^3) + 2 \cdot (45^2 - 6^2)}$	12 024 N
Equivalent response time, Tread brake rigging for bogie C			
57	Equivalent response time Tread brake rigging	$t_{\mathrm{e,i}} = t_{\mathrm{a}} + \frac{t_{\mathrm{ab}}}{2}$ $t_{\mathrm{e,i}} = 1 + \frac{5}{2}$	3,5 s
Equivalent response time, Magnetic track brake for bogie C			
57	Equivalent response time Magnetic track brake	$t_{\mathrm{e,i}} = t_{\mathrm{a}} + \frac{t_{\mathrm{ab}}}{2}$ $t_{\mathrm{e,i}} = 1{,}5 + \frac{0{,}4}{2}$	1,7 s

EXAMPLE CALCULATION FOR BOGIE D

WITH DISC BRAKE UNIT (2 x DISCS PER CYLINDER) AND EDDY CURRENT BRAKE

Table C.8 — Table of parameters and typical values

Bogie No.	Characteristic description	Symbol	Unit	Typical value
D	Pressure at brake cylinder (p_C = 3,0 bar = 3,0 * 10^5 Pa)	p_C	bar	3,0
D	Cylinder area (203 mm - cylinder) (A_C = 324 cm^2 = 324 * 10^{-4} m^2)	A_C	cm^2	324
D	Internal brake cylinder ratio/factor	i_C	-	1
D	Internal brake unit efficiency	η_C	-	0,98
D	Brake cylinder spring force (Return spring)	$F_{S,C}$	N	−200
D	Total rigging ratio per disc brake unit arrangement for a double disc	i_{rig}	-	10
D	Total dynamic rigging efficiency per disc brake unit arrangement	$\eta_{rig,dyn}$	-	0,90
D	Mean friction coefficient of brake pad	μ_m	-	0,35
D	Mean swept radius	r_S	m	0,2
D	Wheel diameter	D	m	0,89
D	Transmission ratio used when a drive gearbox is used	i_{tra}	-	2
D	Transmission efficiency used when a drive gearbox is used (η_{tra} = 90 % = 0,9)	η_{tra}	%	90
D	Maximum eddy current braking force	$F_{BEC,\max}$	N	18 000
D	Characteristic speed where $F_{BEC} = F_{BEC,max}$	v_{cha}	m/s	40
D	Value of power in speed range above v_{cha}	n_1	-	0,6
D	Value of power in speed range below v_{cha}	n_2	-	1,0
D	Initial speed	v_0	m/s	45

Bogie No.	**Characteristic description**	**Symbol**	**Unit**	**Typical value**
D	Speed at eddy current brake switch off	v_1	m/s	40
D	Delay time (Disc brake unit)	t_a	s	0,3
D	Build–up time (Disc brake unit)	t_{ab}	s	3,0
D	Delay time (Eddy current brake)	t_a	s	2,0
D	Build–up time (Eddy current brake)	t_{ab}	s	1,0
D	Rigging return spring force	$F_{S,rig}$	N	−500
D	Characteristic coefficient of the vehicle	A	N	3 000
D	Characteristic coefficient of the vehicle	B	$\frac{N}{m/s}$	110
D	Characteristic coefficient of the vehicle	C	$\frac{N}{(m/s)^2}$	7

Table C.9 — Bogie D – Example calculations

Formula No.	Title	Formula	Value
Disc brake unit on axles for bogie D axles 7 and 8			
24	Cylinder output force	$F_C = p_C \cdot A_C \cdot i_C \cdot \eta_C + F_{S,C}$ $F_C = 3{,}0 \cdot 10^5 \cdot 324 \cdot 10^{-4} \cdot 1 \cdot 0{,}98 - 200$	9 326 N
25	Total clamp force per disc brake unit arrangement	$F_b = F_C \cdot i_{rig} \cdot \eta_{rig,dyn}$ $F_b = F_C \cdot 10 \cdot 0{,}90$	83 934 N
29	Mean braking force per disc brake unit arrangement	$\overline{F}_{B,C} = F_b \cdot \mu_m \cdot \frac{r_s}{D/2} \cdot \frac{i_{tra}}{\eta_{tra}}$ $\overline{F}_{B,C} = F_b \cdot 0{,}35 \cdot \frac{0{,}2}{0{,}89/2} \cdot \frac{2}{0{,}9}$	30 969 N
Eddy current brake for bogie D			
54	Mean eddy current braking force	$\overline{F_{BEC}} = F_{BEC,max} \cdot \frac{v_0^2 - v_1^2}{\left(\frac{v_{cha}^n}{2-n} \cdot \left(v_0^{2-n} - v_1^{2-n}\right) + \frac{1}{2+n} \cdot \frac{\left(v_0^{2+n} - v_1^{2+n}\right)}{v_{cha}^n}\right)}$ $\overline{F_{BEC}} =$ $18000 \cdot \frac{45^2 - 40^2}{\left(\frac{40^{0,6}}{2-0,6} \cdot \left(45^{2-0,6} - 40^{2-0,6}\right) + \frac{1}{2+0,6} \cdot \frac{\left(45^{2+0,6} - 40^{2+0,6}\right)}{40^{0,6}}\right)}$	17 984 N
Equivalent response time, Disc brake unit for bogie D			
57	Equivalent response time Disc brake unit	$t_{e,i} = t_a + \frac{t_{ab}}{2}$ $t_{e,i} = 0{,}3 + \frac{3}{2}$	1,8 s
Equivalent response time, Eddy current brake for bogie D			
57	Equivalent response time Eddy current brake	$t_{e,i} = t_a + \frac{t_{ab}}{2}$ $t_{e,i} = 2 + \frac{1}{2}$	2,5 s

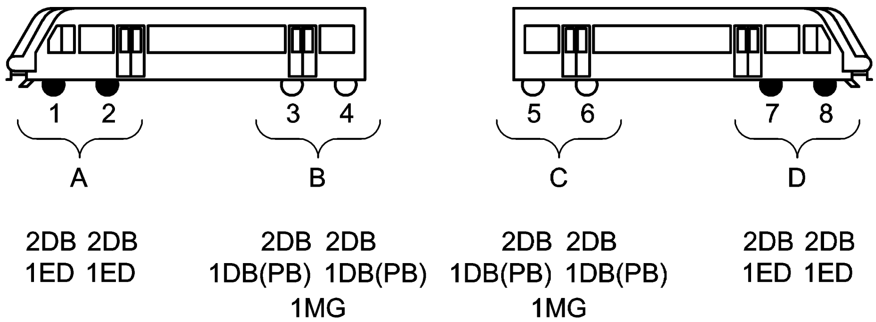
Annex D
(informative)

Train stopping distance and immobilization brake calculation example

D.1 General

For the purpose of demonstrating the use of the formulae derived in the requirements section of this standard the following train calculations are conducted considering a train consisting of two vehicles. The train calculations assume that the brake equipment types on each of the bogies are as shown in Figure D.1 and Tables D.1, D.2, D.3, D.4 and D.5.

The train calculation assumes the existence of a blending concept which itself assumes that when electrodynamic brake force is available then any friction brake force on that bogie is not considered. The magnetic track brake shown as being fitted to bogies two and three is not used in service braking, being only used in emergency braking.



Key

ED Electrodynamic brake

DB Disc brake unit

DB(PB) Disc brake unit (spring parking brake)

MG Magnetic track brake

Figure D.1 — Entity and brake equipment types assumed per bogie/axle

Table D.1 — Entity and brake equipment type description

Bogie	Axle	Brake equipment
A	1 and 2	Discs brake unit (2 per axle) electrodynamic brake
B	3 and 4	Discs brake unit (2 per axle) Disc brake unit (spring parking brake 1 per axle) Magnetic track brake (two magnets)
C	5 and 6	Discs brake unit (2 per axle) Disc brake unit (spring parking brake 1 per axle) Magnetic track brake (two magnets)
D	7 and 8	Discs brake unit (2 per axle) electrodynamic brake

Table D.2 — Bogie and train characteristics and typical values

Bogie No.	Characteristic description	Symbol	Unit	Typical value
A	Operational mass in working order	m_{st}	kg	30 000
A	Rotational mass	m_{rot}	kg	3 000
B	Operational mass in working order	m_{st}	kg	29 000
B	Rotational mass	m_{rot}	kg	1 400
C	Operational mass in working order	m_{st}	kg	29 000
C	Rotational mass	m_{rot}	kg	1 400
D	Operational mass in working order	m_{st}	kg	30 000
D	Rotational mass	m_{rot}	kg	3 000
Train characteristics				
	External forces (train resistance, gradient etc.)	F_{ext}	not considered	
	Rotating mass	m_{rot}	not considered	
	Initial speed	v_0	m/s	45
	Final speed	v_{fin}	m/s	0

D.2 Train stopping calculations

These train calculations assume an initial speed v_0 = 45 m/s and a final speed v_{fin} = 0 m/s using a full service brake on level track. The values used below are taken from the brake equipment type calculations conducted in Annex C.

Vehicle 1: Mean brake force and equivalent response time per brake equipment type

Bogie A

Disc brake (2 disc per axle): $t_{e,i}$ = 1,3 s; $\overline{F_{\mathrm{B,i}}} = F_{\mathrm{B}}$ = 9 104 N per disc (not considered due to blending concept assumption)

Electrodynamic brake (1 per axle): $t_{e,i}$ = 0,8 s; $\overline{F_{\mathrm{B,i}}} = \overline{F_{\mathrm{BED}}} =$ 19 188 N

Bogie B

Disc brake (2 disc per axle): $t_{e,i}$ = 1,3 s; $\overline{F_{\mathrm{B,i}}} = F_{\mathrm{B,C}}$ = 9 104 N per disc

Magnetic track brake (per bogie): $t_{e,i}$ = 1,7 s; $\overline{F_{\mathrm{B,i}}} = \overline{F_{\mathrm{BMG}}} =$ 12 113 N (not considered due to full service braking)

Vehicle 2: Mean brake force and equivalent response time per brake equipment type

Bogie C

Disc brake (2 disc per axle): $t_{e,i}$ = 1,3 s; $\overline{F_{\mathrm{B,i}}} = F_{\mathrm{B}}$ = 9 104 N per disc

Magnetic track brake (1 per bogie): $t_{e,i}$ = 1,7 s; $\overline{F_{\mathrm{B,i}}} = \overline{F_{\mathrm{BMG}}} =$ 12 113 N (not considered due to full service braking)

Bogie D

Disc brake (2 disc per axle): $t_{e,i}$ = 1,3 s; $\overline{F_{\mathrm{B,i}}} = F_{\mathrm{B,C}}$ = 9 104 N per disc (not considered due to blending concept assumption)

Electrodynamic brake (1 per axle): $t_{e,i}$ = 0,8 s; $\overline{F_{\mathrm{B,i}}} = \overline{F_{\mathrm{BED}}} =$ 19 188 N

Table D.3 — Train example calculations

Formula No.	Title	Formula	Value
2	Dynamic mass (train) (m_{rot} ignored as first estimation calculation assumed)	$m_{dyn} = \sum(m_{st} + m_{rot})$ $m_{dyn} = 2 \cdot (30000 + 29000)$	118 000 kg
3	Mean train resistance ($F_{\mathrm{Ra,m}}$ ignored as first estimation calculation assumed)	$\overline{F_{Ra}} = A + \frac{2}{3} \cdot B \cdot \frac{v_0{}^2 + v_0 \cdot v_{fin} + v_{fin}{}^2}{v_0 + v_{fin}} + \frac{1}{2} \cdot C \cdot (v_0{}^2 + v_{fin}{}^2)$ $F_{Ra,m} = 0$	
59	Equivalent response time of the train	$t_e = \frac{\sum(t_{e,i} \cdot \overline{F_{B,i}})}{\sum\overline{F_{B,i}} + \sum\overline{F_{ext}}}$ $t_e = \begin{pmatrix} 8 \cdot 1.3 \cdot 9104 + \\ 4 \cdot 0.8 \cdot 19188 \end{pmatrix} \Big/ \begin{pmatrix} 8 \cdot 9104 + \\ 4 \cdot 19188 \end{pmatrix}$	1,04 s
73	Equivalent deceleration	$a_e = \frac{\sum\overline{F_{B,i}} + \sum\overline{F_{ext}}}{m_{dyn}}$ $a_e = \begin{pmatrix} 8 \cdot 9104 + \\ 4 \cdot 19188 \end{pmatrix} \Big/ (118000)$	1,27 m/s²
76	Stopping distance	$s = v_0 \cdot t_e + \frac{v_0^2 - v_{fin}^2}{2 \cdot a_e}$ $s = 45 \cdot 0.96 + \frac{45^2}{2 \cdot 1.27}$	840 m
69	Required adhesion value for each type of axle. This example uses the characteristics of Bogie A Axle 1 ED brake only.	$\tau_{req,ax} = \frac{(\sum_{ax}\overline{F_{B,i}} - m_{rot,ax} \cdot a_e)}{m_{st,ax} \cdot g_n} \cdot \sqrt{1 + i^2}$	0,13

		$\tau_{req,ax} = \frac{(19188 - 0 \cdot 1.27)}{30000/2 \cdot 9.81} \cdot 1$	

D.3 Train stopping calculations on a gradient

This train calculation assumes a falling gradient of i = −0,02

NOTE Considering Formula (78), in 4.8.6.1, this train configuration meets the criteria because the initial speed is greater than 50 km/h and the equivalent response time of the train is not more than 3 s. Therefore it is acceptable to use Formula (78).

Table D.4 — Train stopping distance on a falling gradient example calculation

Formula No.	Title	Formula	Value
78	Stopping distance on a gradient	$s_{\text{grad}} = v_0 \cdot t_e + \frac{v_0^2 \cdot (s - v_0 \cdot t_e)}{v_0^2 + 2 \cdot \frac{m_{\text{st}}}{m_{\text{dyn}}} \cdot g_n \cdot i \cdot (s - v_0 \cdot t_e)}$ $s_{\text{grad}} = 45 \cdot 1{,}04 + 45^2 \cdot (845 - 45 \cdot 1{,}04) / \begin{pmatrix} 45^2 + 2 \cdot 118\,000/118\,000 \cdot \\ 9{,}81 \cdot (-0{,}02) \cdot (845 - 45 \cdot 1{,}04) \end{pmatrix}$	991 m

D.4 Train immobilization (parking) brake calculations

This train calculation assumes no effect of train resistance or wind, and a falling gradient of *i* = −0,02.

Bogie B

Parking brake force per axle $F_{\text{im,disc}}$ = 7 385 N

Bogie C

Parking brake force per axle $F_{\text{im,disc}}$ = 7 385 N

Table D.5 — Train Immobilization (parking) brake example calculation

Formula No.	Title	Formula	Value
91	Total immobilization (parking) force per train	$F_{\text{im}} = \sum F_{\text{B,ind}} + \sum_{\text{ax}} F_{\text{im,ax}}$ $F_{\text{im}} = 0 + 4 \cdot 7\,385$	29 540 N
67	Downhill force on the train	$F_g = m_{\text{st}} \cdot g_n \cdot i$ $F_g = (30\,000 + 29\,000 + 29\,000 + 30\,000) \cdot 9{,}81 \cdot 0{,}02$	23 151 N
92	Immobilization safety factor	$S_{\text{im}} = \frac{F_{\text{im}} + F_{\text{Ra}}}{F_g + F_{\text{wind}}}$	1,27

		$S_{\text{im}} = \dfrac{29\,540 + 0}{23\,151 + 0}$	
96	Maximum achievable gradient	$i_{\max} = \dfrac{F_{\text{im}} - F_{\text{wind}} + F_{\text{Ra,im}}}{m_{\text{st}} \cdot g_n}$ $i_{\max} = \dfrac{29\,540 - 0}{118\,000 \cdot 9{,}81}$	0,025

Annex E
(informative)

Development of the formula for the mean brake force with respect to the braking distance

Basically an instantaneous characteristic of brake force might be a character of current train speed.

$$F_{\mathrm{B}} = f(v) \tag{E.1}$$

When using a mean value calculation the non-constant speed dependent brake force is transformed to a mean brake force value. This mean value of brake force is considered a fully developed force without considering the response time and results in the same braking distance as in the analytical solution shown in Formula (E.2). The general analytical solution for the braking distance is

$$s = m_{\mathrm{dyn}} \cdot \int_{\mathrm{v_{fin}}}^{\mathrm{v_0}} \frac{v}{F_{\mathrm{B}}(v)} \mathrm{dv} \tag{E.2}$$

where:

$F_B(v)$	instantaneous brake force dependent on the current speed, in N
m_{dyn}	dynamic mass, in kg
v	current speed, in m/s
v_0	initial speed, in m/s
v_{fin}	final speed, in m/s
s	braking distance, in m

In the case of a constant brake force, e.g. mean brake force, the Formula (E.2) leads to the well-known formula:

$$s = \frac{m_{\mathrm{dyn}}}{\overline{F_{\mathrm{B}}}} \cdot \frac{v_0^2 - v_{\mathrm{fin}}^2}{2} \tag{E.3}$$

Equalizing Formula (E.2) and Formula (E.3) on the distance base creates the formula

$$\overline{F_{\mathrm{B}}} = \frac{v_0^2 - v_{\mathrm{fin}}^2}{2} \Bigg/ \int_{\mathrm{v_{fin}}}^{\mathrm{v_0}} \frac{v}{F_{\mathrm{B}}(v)} \mathrm{dv} \tag{E.4}$$

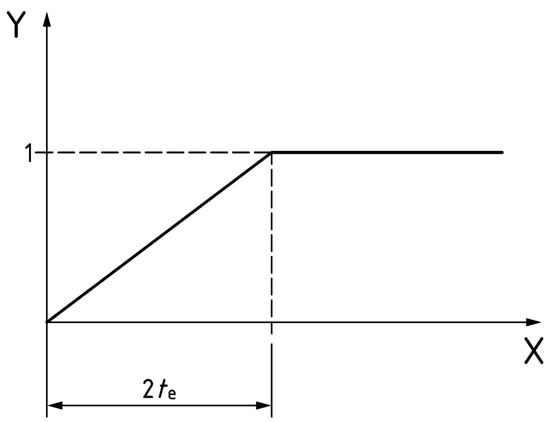
Formula (E.4) is the general rule when determining a mean brake force for brake equipment types which are speed dependent e.g. magnetic track brake, eddy current brake etc.

If the speed dependent brake force characteristic of the brake equipment type demands different sections of the speed range, different brake force formulae are used. The integral of Formula (E.4) is then split into the speed ranges each with a homogeneous characteristic e.g. refer to 4.4.7.1, Figure 7 (electrodynamic brake).

Annex F
(informative)

Slowing or stopping distance calculation using alternative method of equivalent response time calculation as French Railway requirements in particular for trains operating in 'G' position

For this French model of slowing or stopping distance calculation, Figure F.1 may be used for trains operating in 'G' mode with friction braking only. This model has been developed in response to practical experience on French Railways.



Key

Y factor of nominal braking force, deceleration or pressure (in the case of pressure systems), in %

X time, in s

1 the point when the brake force, deceleration or pressure has been substantially achieved, typically 95 %

$2te$ equivalent response time multiplied by 2, in s

Figure F.1 — Graphic representation of equivalent response time

For calculation on a train basis the equivalent response time is calculated using:

$$t_e = t_a + \frac{t_{ab}}{2}. \quad \text{(F.1)}$$

t_a and t_{ab} in accordance with 4.4.8.3.

The stopping (v_n = 0) or slowing distance calculation is as follows:

$$s_{grad} = v_0 \cdot t_e . \frac{a_e}{a_e + g_n.i} + \frac{v_0^2 - v_{fin}^2}{2 \cdot (a_e + g_n.i)} - \frac{a_e.t_e{}^2.(a_e + 4g_n.i)}{6.(a_e + g_n.i)} \quad \text{(F.2)}$$

The formula is valid for stopping or slowing distance calculation when the braking force is fully established.

NOTE The distances calculated with this model are shorter than those when using the model of equivalent response time described in 4.4.8.3.

$$v_0 - v_{\mathrm{fin}} \geq (a_{\mathrm{e}} + 2.g_n.i).t_{\mathrm{e}} \tag{F.3}$$

where:

s_{grad}	stopping distance on a gradient, in m
a_e	equivalent deceleration, in m/s^2
t_e	equivalent response time, in s
v_0	initial speed, in m/s
v_{fin}	final speed, in m/s
g_n	standard acceleration of free fall, in m/s^2
i	gradient of the track

Annex ZA
(informative)

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

This European Standard has been prepared under mandates given to CEN/CENELEC/ETSI by the European Commission and the European Free Trade Association to provide a means of conforming to Essential Requirements of the Directive 2008/57/EC[1)]

Once this standard is cited in the Official Journal of the European Communities under that Directive and has been implemented as a national standard in at least one Member State, compliance with the clauses of this standard given in Table ZA.1 for freight wagons and Table ZA.2 for locomotive and passenger RST, confers within the limits of the scope of this standard, a presumption of conformity with the corresponding Essential Requirements of that Directive and associated EFTA regulations.

Table ZA.1 — Correspondence between this European Standard, the Commission Regulation concerning the technical specification for interoperability relating to the subsystem 'rolling stock – freight wagons' of the rail system in the European Union and repealing Decision 2006/861/EC (published in the Official Journal L 104, 12.4.2013, p.1) and Directive 2008/57/EC

Clause/subclauses of this European Standard	Chapter/§/annexes of the Technical Specification for Interoperability (TSI)	Corresponding text, articles/§/annexes of the Directive 2008/57/EC	Comments
The whole standard is applicable	4.Characterization of the subsystem 4.2. Functional and technical specifications of the subsystem 4.2.4. Brake 4.2.4.3.Functional and technical requirements 4.2.4.3.2 Brake performance 4.2.4.3.2.1 Service brake 4.2.4.3.2.2 Parking brake Appendix C: Additional optional conditions 9, Table C.3 Minimum braking performance for brake modes G and P	Annex III, Essential requirements 1 General requirements 1.1 Safety Clauses 1.1.1, 1.1.2 1.5 Technical compatibility 2 Requirements specific to each subsystem 2.4 Rolling stock 2.4.1 Safety §3 2.4.2 Reliability and availability 2.4.3 Technical compatibility §3	The previous issue of this document is referenced in the TSI, and its use is mandatory. Nevertheless, the use of the two new parts of the EN 14531 series is equivalent to the use of the referenced issues. The equivalences of the new EN 14531 series to the referred subclauses of the original EN 14531 series in the TSI are shown in the following table:

1) This Directive 2008/57/EC adopted on 17th June 2008 is a recast of the previous Directives 96/48/EC 'Interoperability of the trans-European high-speed rail system' and 2001/16/EC 'Interoperability of the trans-European conventional rail system' and revisions thereof by 2004/50/EC 'Corrigendum to Directive 2004/50/EC of the European Parliament and of the Council of 29 April 2004 amending Council Directive 96/48/EC on the interoperability of the trans-European high-speed rail system and Directive 2001/16/EC of the European Parliament and of the Council on the interoperability of the trans-European conventional rail system'.

	TSI		Normative document			
Index N°	**Characteristics to be assessed**	**Clause**	**Current issue of** EN 14531	**Mandatory clauses**	**Revised issue of** EN 14531	**Proposal**
	Brake performance – Service Brake	4.2.4.3.2.1	EN 14531–6:2009	relevant clauses	EN 14531–1 EN 14531–2	4 5
	Braking performance – Parking Brake	4.2.4.3.2.2	EN 14531–6:2009	Clause 6	EN 14531–1 EN 14531–2	4 5
	Brake	C.9	EN 14531–1:2005	5.11	EN 14531–1	4

Table ZA.2 — Correspondence between this European Standard, the TSI Locomotive and Passenger Rolling Stock (approved by the RISC68 on 23 October 2013), and Directive 2008/57/EC

Clause/ subclauses of this European Standard	**Chapter/§/annexes of the technical specification for interoperability (TSI)**	**Corresponding text, articles/§/annexes of the Directive 2008/57/EC**	**Comments**
The whole standard is applicable	4.Characterization of the rolling stock subsystem 4.2 Functional and technical specifications of the subsystem 4.2.4 Braking 4.2.4.5 Braking performance 4.2.4.5.1 General requirements 4.2.4.5.2 Emergency braking 4.2.4.5.3 Service braking 4.2.4.5.5 Parking brake 6. Assessment of conformity or suitability for use and 'ec' verification 6.2 Rolling stock subsystem 6.2.3 Particular assessment procedures for subsystems 6.2. 3.8 Emergency braking 6.2. 3.9 Service braking	Annex III, Essential requirements 1 General requirements 1.1 Safety 1.1.1, 1.1.2 2 Requirements specific to each subsystem 2.4 Rolling stock 2.4.2 Reliability and availability 2.4.3 Technical compatibility §3	The previous issue of this document is referenced in the TSI and its use would be mandatory. Nevertheless, the use of the two new parts of the EN 14531 series is equivalent to the use of the referenced issues The equivalences of the new EN 14531 series to the referenced subclauses of the original EN 14531 series in the TSI are shown in the following table:

	TSI		Normative document			
Index N°	**Characteristics to be assessed**	**Clause**	**Current issue of** EN 14531	**Mandatory clauses**	**Revised issue of** EN 14531	**Proposal**
23	Braking performance – calculation – general	4.2.4.5.1	EN 14531-1:2005 or EN 14531-6:2009	relevant clauses	EN 14531-1 EN 14531-2	4 5
24	Braking performance – friction coefficient	4.2.4.5.1	EN 14531-1:2005	5.3.1.4	EN 14531-1	4.4.6
25	Emergency braking performance – response time / delay time brake weight percentage	4.2.4.5.2	EN 14531–1:2005	5.3.3 5.12	EN 14531–1	4.4.8.2.3 and 4.4.8.3 4.9 and 4.10
26	Emergency braking performance – calculation	4.2.4.5.2	EN 14531–1:2005 or EN 14531–6:2009	relevant clauses	EN 14531–1 EN 14531–2	4 5
27	Emergency braking performance – friction coefficient	4.2.4.5.2	EN 14531–1:2005	5.3.1.4	EN 14531–1	4.4.6
28	Service braking performance – calculation	4.2.4.5.3	EN 14531–1:2005 or EN 14531–6:2009	relevant clauses	EN 14531–1 EN 14531–2	4 5
29	Parking brake performance – calculation	4.2.4.5.5	EN 14531–1:2005 or EN 14531–6:2009	relevant clauses	EN 14531–1 EN 14531–2	4 5
91	Emergency braking performance	6.2.3.8	EN 14531–1:2005	5.11.3	EN 14531–1 EN 14531–2	4 5
92	Service braking performance	6.2.3.9	EN 14531–1:2005	5.11.3	EN 14531–1 EN 14531–2	4 5

WARNING — Other requirements and other EC directives may be applicable to the product(s) falling within the scope of this standard.

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[1] EN 14067-1, *Railway applications - Aerodynamics - Part 1: Symbols and units*

[2] EN 13452-1:2003, *Railway applications - Braking - Mass transit brake systems - Part 1: Performance requirements*

[3] EN 13452-2:2003, *Railway applications - Braking - Mass transit brake systems - Part 2: Methods of test*

[4] EN 15355, *Railway applications – Braking – Distributor valves and distributor isolating devices*

[5] EN 15734-1, *Railway applications - Braking systems of high speed trains - Part 1: Requirements and definitions*

[6] EN 15734-2, *Railway applications - Braking systems of high speed trains - Part 2: Test methods*

[7] Railway Group Standard GM/RT 2040, *Calculation of brake force data for rolling stock library*

[8] ISO 80000-3, *Quantities and units — Part 3: Space and time*

[9] ISO 80000-4, *Quantities and units — Part 4: Mechanics*

[10] UIC 544-1, *Brakes – Braking power*

[11] EN 16185 (all parts), *Railway applications - Braking systems of multiple unit trains*

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