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

**Part 1:
General algorithms utilizing mean
value calculation**

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

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





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Foreword

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

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

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

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

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

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

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

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

Introduction

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

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

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

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

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

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

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

Part 1: General algorithms utilizing mean value calculation

1 Scope

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

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

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

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

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

2 Normative references

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

ISO 20138-2¹⁾, *Railway applications — Calculation of braking performance (stopping, slowing and stationary braking) — Part 2: General algorithms utilizing step by step calculation*

3 Terms and definitions

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

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

- ISO Online browsing platform: available at <https://www.iso.org/obp>
- IEC Electropedia: available at <https://www.electropedia.org/>

3.1

train

operational formation consisting of one or more units

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

3.2

trainset

fixed formation unit that can operate as a train

3.3

unit

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

3.4

vehicle

individual element of a unit

EXAMPLE Locomotive, coach, wagon, driving coach.

3.5

retention force

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

3.6

brake force

retarding force

forces generated by a brake equipment type or external forces

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

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

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

4 Symbols

For the purposes of this document, the general symbols given in [Table 1](#) apply.

Table 1 — Symbols

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

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

Table 1 (continued)

Symbol	Definition	Unit
$F_{AMg,st}$	Attraction force of one permanent magnet	N
F_B	Contribution of the friction brake	N
F_{Bd}	Blended retarding force	N
F_{BED}	Retarding force of electro-dynamic brake	N
$\bar{F}_{BED}$	Mean retarding force of electro-dynamic brake	N
$F_{BED,max}$	Maximum retarding force of electro-dynamic brake	N
F_{BFR}	Retarding force of fluid retarder	N
$\bar{F}_{BFR}$	Mean retarding force of fluid retarder	N
$F_{BFR,max}$	Maximum retarding force of fluid retarder	N
$\bar{F}_{B,n}$	Mean braking force of brake equipment type n	N
$F_{B,ind}$	Adhesion independent (not related to the wheel to rail contact) force, e.g. force of permanent magnetic track brake	N
$F_{B,ind,z}$	Adhesion independent (not related to the wheel to rail contact) retarding force per type of equipment	N
$F_{B,st}$	Total stationary brake force acting at the rail	N
$F_{B,ax,st}$	Stationary brake force acting on that wheelset	N
$F_{B,\tau,i}$	Adhesion dependent (related to the wheel to rail contact) retarding force generated by applied parking brake (i is an index used for sorting wheelsets)	N
$F_{B,\tau,req}$	Adhesion dependent (related to the wheel to rail contact) retarding force	N
$F_{B,\tau,req,rem}$	Remaining force of the mass to be held	N
F_b	Brake block force	N
$F_{b,st}$	Parking brake force acting on the tread of the wheel from a single parking brake unit	N
$F_{b,ax}$	Single brake block force	N
$F_{b,ax,st}$	Static single brake block force	N
$F_{b,tot}$	Total force acting on all disc faces or total brake block force	N
$F_{b,tot,st}$	Total static force acting on all disc faces or total static brake block force	N
$F_{Cr,H}$	Crank handle or hand wheel force	N
F_c	Internal cylinder force	N
F_{cl}	Clamping force	N
$F_{cl,n}$	Brake calliper clamping force	N
F_D	Downhill force due to gravity	N
$F_{d,ax}$	Proportion of downhill force to be resisted per wheelset with applied parking brakes	N
$\bar{F}_{ext}$	Mean external force	N
F_{ext}	External forces (e.g. wind force)	N
F_G	Output force of parking brake mechanism	N
F_g	Weight	N
$F_{g,ax}$	Static axle load	N
F_H	Retention force	N
$F_{Mg,st}$	Parking brake force of one permanent magnet	N
$F_{Mg,st,tot}$	Total parking brake force of all permanent magnets in a vehicle	N
^a bar or kPa is also allowed; 1 bar = 10 ⁵ Pa.		

Table 1 (continued)

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

Table 1 (continued)

Symbol	Definition	Unit
$\eta_{R,st}$	Overall static efficiency of brake rigging	—
η_{rig}	Efficiency of brake rigging/calliper	—
$\eta_{rig,st}$	Static efficiency of calliper	—
I	Current	A
i	Gradient of the track (positive rising/negative falling)	—
i_{Cbl}	Cable mechanical ratio	—
i_c	Internal rigging ratio of brake unit	—
i_G	Gear ratio	—
i_{max}	Maximum achievable gradient	—
$i_{max,roll}$	Maximum achievable gradient for rolling	—
$i_{max,slide}$	Maximum achievable gradient for sliding	—
i_{rig}	Rigging ratio	—
$i_{rig,ax,n}$	Lever ratio per brake beam	—
$i_{rig,C}$	Calliper lever ratio (parking brake)	—
$i_{rig,n}$	Calliper lever ratio	—
$i_{s,rig}$	Rigging ratio for restoring force	—
J	Inertia	kg·m ²
k_0, k_2	Coefficient (provided by the supplier)	—
k_1	Coefficient (provided by the supplier)	s/m
k_{1v}, k_{2v}	Factor describing an active or passive brake cylinder	—
l_a, l_b	Main brake lever length	m
$l_{a,n}, l_{b,n}$	Calliper lever length	m
l_e	Main brake lever length (parking brake)	m
l_c, l_d	Bogie lever length	m
M	Mass to be held of the vehicle/unit/train	kg
m_{dyn}	Dynamic mass	kg
m_{rot}	Equivalent rotating mass	kg
$m_{rot,ax}$	Equivalent rotating mass of the braked wheelset	—
m_{st}	Static mass	kg
$m_{st,ax}$	Static mass per wheelset	kg
μ_{Mg}	Mean friction coefficient of magnet (pole shoe)	—
$\mu_{Mg,st}$	Static friction coefficient of permanent magnet (pole shoe)	—
μ_m	Mean friction coefficient of brake block/brake pad	—
μ_{st}	Static friction coefficient of brake block/brake pad	—
N	Number of brake equipment types	—
$n_{PB,ax}$	Number of wheelsets with applied parking brake	—
n_{Beam}	Number of brake beams	—
n_{BW}	Number of braked wheelsets	—
n_{disc}	Number of brake discs	—
n_{face}	Number of disc faces	—
n_{Mg}	Number of magnets in a vehicle	—
n_{SP}	Number of spring brake units	—
n_1, n_2	Value of power in speed range above v_{cha} normally obtained from supplier	—

^a bar or kPa is also allowed; 1 bar = 10⁵ Pa.

Table 1 (continued)

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

^a bar or kPa is also allowed; 1 bar = 10⁵ Pa.

Table 1 (continued)

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

5 Stopping and slowing distances calculation

5.1 General

A summary of the methodology to establish the braking forces acting on the train is presented in [Figure A.1](#).

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

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

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

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

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

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

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

[Annex C](#) provides examples for brake calculations of different vehicles and units.

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

[Figure 1](#) gives a general overview of brake equipment types.

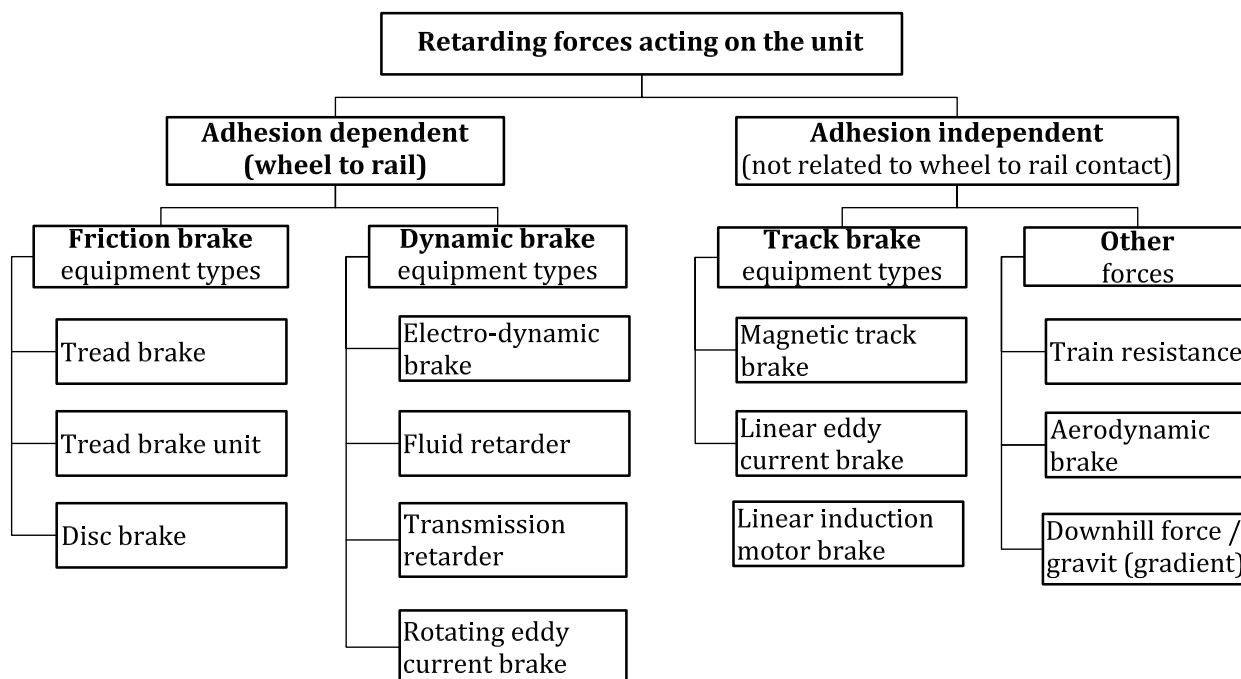


Figure 1 — General overview of retarding forces acting on the unit

5.2 Vehicle characteristics

5.2.1 Static mass, m_{st}

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

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

5.2.2 Equivalent rotating mass, m_{rot}

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

- the rotation of the wheelsets, and
- the rotating parts coupled to the wheelsets during braking.

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

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

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

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

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

where

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

J is the inertia, expressed in $\text{kg}\cdot\text{m}^2$;

D is the wheel diameter, expressed in m.

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

5.2.3 Dynamic mass, m_{dyn}

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

$$m_{\text{dyn}} = m_{\text{st}} + m_{\text{rot}} \quad (2)$$

where

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

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

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

5.2.4 Wheel diameter

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

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

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

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

5.3 Adhesion wheel/rail dependent brake equipment type characteristics

5.3.1 Basic brake cylinder

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

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

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

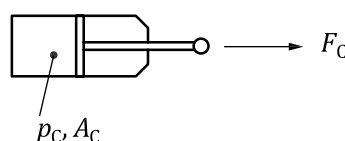


Figure 2 — Basic principle of a pressure applied cylinder

$$F_c = p_c \cdot A_c$$

(3)

where

- p_c

is the brake cylinder pressure, expressed in Pa;
- A_c

is the brake cylinder area, expressed in m²;
- F_c

is the internal cylinder force, expressed in N.

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

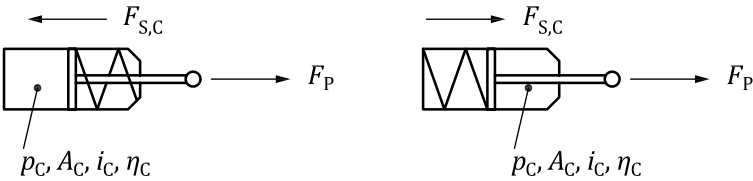


Figure 3 — Basic principle of an active (left) and passive (right) cylinder

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

$$F_p = k1_v \cdot |p_c \cdot A_c \cdot \eta_c \cdot i_c| + k2_v \cdot |F_{S,C}|$$

(4)

where

- F_p

is the piston force, expressed in N;
- p_c

is the brake cylinder pressure, expressed in Pa;
- A_c

is the brake cylinder area, expressed in m²;
- η_c

is the internal efficiency of brake unit;
- i_c

is the internal rigging ratio of brake unit;
- $F_{S,C}$

is the restoring force of brake unit or spring applied force, expressed in N.

Table 2 — Factors $k1_v$ and $k2_v$ for active and passive brake cylinder

	Active cylinder (pressure-applied brake)	Passive cylinder (spring-applied brake)
$k1_v$	1	-1
$k2_v$	-1	1

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

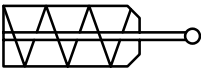


Figure 4 — Illustration of general brake cylinder

5.3.2 Tread brake

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

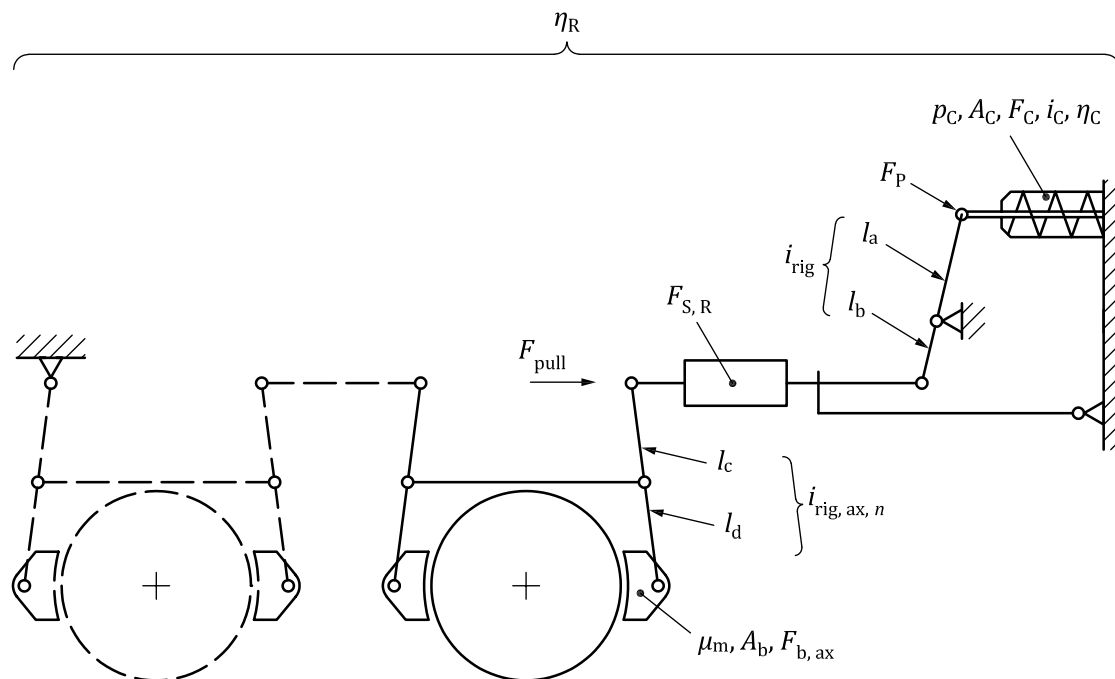


Figure 5 — Typical clasp tread brake arrangement

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

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

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

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

where

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

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

NOTE 1 Due to the brake beam, there are two brake blocks per wheelset. Therefore, the multiplier $\frac{1}{2}$ in [Formula \(6\)](#) is used to calculate the single brake block force, $F_{b,ax}$. For [Figure 5](#), four brake beams per bogie means eight brake blocks per bogie. For [Figure 6](#), two brake beams per bogie means four brake blocks per bogie.

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

$$F_{b,tot} = \sum F_{b,ax} \cdot \eta_R \quad (9)$$

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

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

$$p_{ab} = \frac{F_{b,ax}}{A_b} \quad (10)$$

where

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

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

$$\bar{F}_r = F_{b,tot} \cdot \mu_m \quad (11)$$

where

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

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

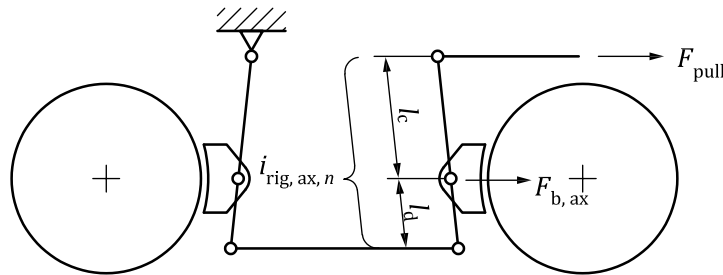
If the internal bogie ratios are equal, the simplifications shown in [Formula \(12\)](#) can be done:

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

[Formula \(9\)](#) can be simplified with

- [Formulae \(6\)](#) and (7) for bogie with single-sided brake block arrangement (see [Figure 6](#)), or
 - [Formulae \(6\)](#) and (8) for bogie with double-sided brake block arrangement (see [Figure 5](#))
- to [Formula \(13\)](#):

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



EXAMPLE Three-piece bogie.

Figure 6 — Typical single-side brake block arrangement

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

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

where

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

5.3.3 Tread brake unit

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

The principle is the same for a spring applied design.

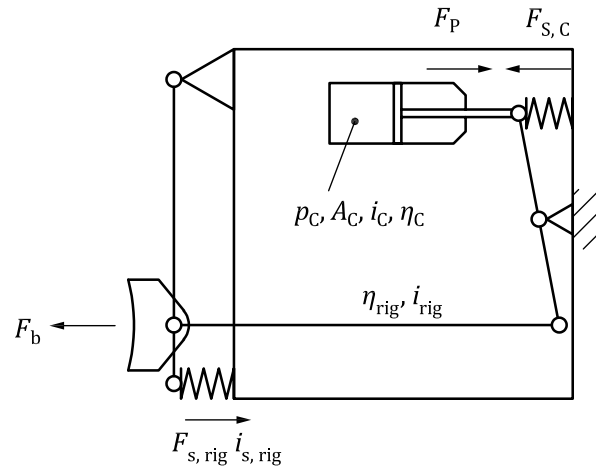


Figure 7 — Representative tread brake unit

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

where

- F_b is the brake block force, expressed in N;
- F_P is the piston force [see [Formula \(4\)](#)], expressed in N;
- i_{rig} is the rigging ratio;
- η_{rig} is the efficiency of brake rigging;
- $F_{s,rig}$ is the restoring force, expressed in N;
- $i_{s,rig}$ is the rigging ratio for restoring force.

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

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

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

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

where

- $\bar{F}_r$ is the mean retarding force acting at the rail generated by a tread brake unit, expressed in N;
- F_b is the brake block force, expressed in N;
- μ_m is the mean friction coefficient of brake block.

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

5.3.4 Disc brake

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

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

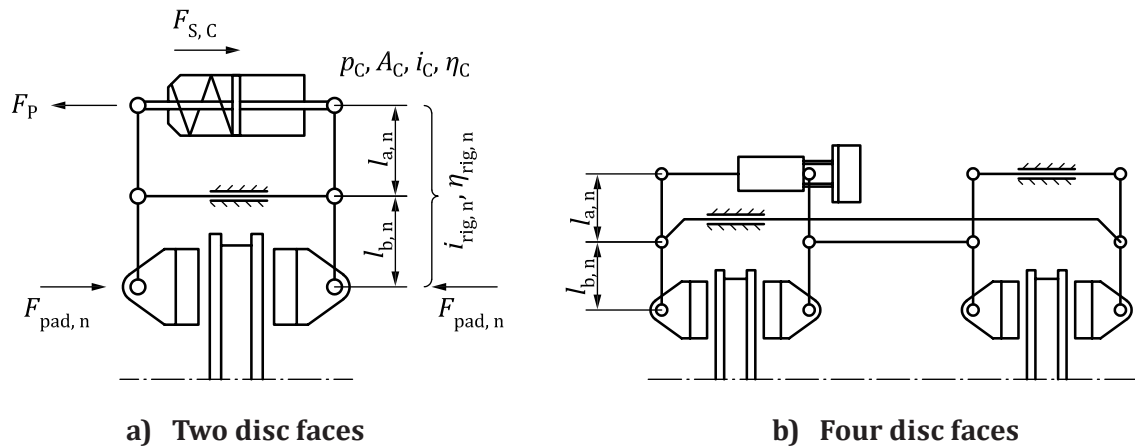


Figure 8 — Typical arrangement of brake disc calliper

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

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

$$F_{pad,n} = F_P \cdot i_{rig,n} \cdot \eta_{rig} \quad (18)$$

With calliper lever ratio as shown in [Formula \(19\)](#):

$$i_{rig,n} = \frac{l_{a,n}}{l_{b,n}} \quad (19)$$

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

$$p_{ap} = \frac{F_{pad}}{A_p} \quad (20)$$

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

$$F_{cl,n} = 2 \cdot F_{pad,n} \quad (21)$$

where

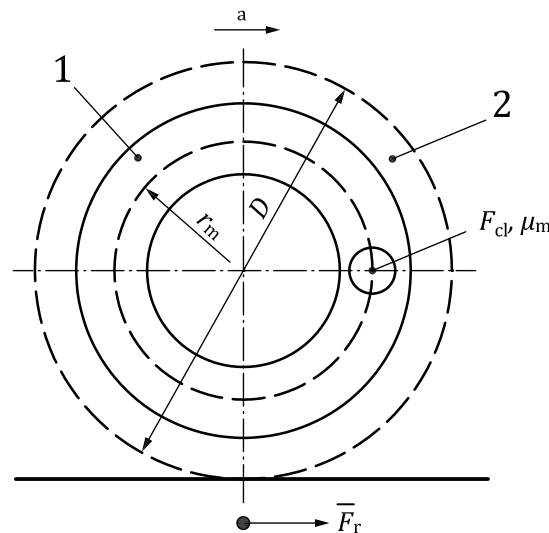
F_{pad} is the force acting on single disc surface, expressed in N;

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

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

- $i_{rig,n}$ is the calliper lever ratio;
- η_{rig} is the efficiency of calliper;
- $l_{a,n}, l_{b,n}$ are the calliper lever lengths, expressed in m;
- $F_{cl,n}$ is the brake calliper clamping force, expressed in N;
- p_{ap} is the specific pressure per brake pad, expressed in N/m²;
- A_p is the contact area per brake pad, expressed in m².

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



Key

- 1 disc
- 2 wheel
- a Driving direction.

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

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

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

where

- $\bar{F}_r$ is the mean retarding force acting at the rail generated by the brake equipment, expressed in N;
- F_{cl} is the clamping force, expressed in N;
- n_{disc} is the number of brake discs;

- μ_m is the mean friction coefficient of brake pad;
- r_m is the mean swept radius of the brake pad on the disc face, expressed in m;
- D is the wheel diameter, expressed in m.

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

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

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

$$F_t = F_{cl} \cdot \mu_m \quad (23)$$

where

- F_t is the tangential force, expressed in N;
- F_{cl} is the clamping force, expressed in N;
- μ_m is the mean friction coefficient of brake pad.

5.3.5 Electro-dynamic brake

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

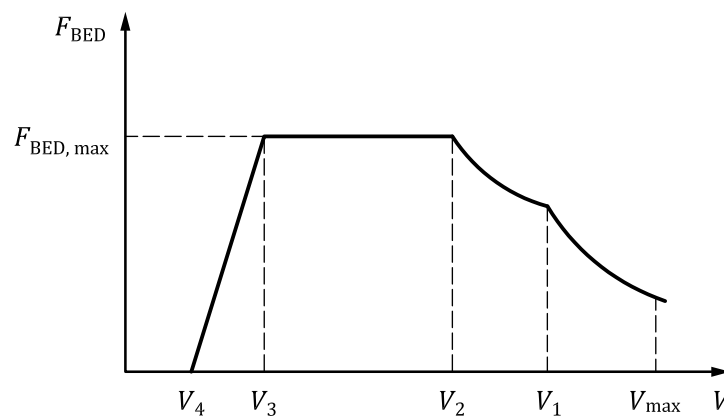


Figure 10 — Indicative characteristic braking force of an electro-dynamic brake

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

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

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

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

An example of electro-dynamic braking force calculation is provided in [D.3](#).

5.3.6 Fluid retarder

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

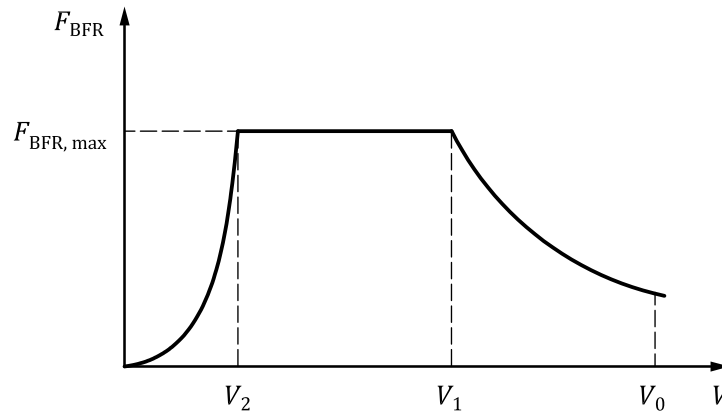


Figure 11 — Indicative characteristic retarding force of the fluid retarder

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

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

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

An example of retarding force of the fluid retarder, $F_{\text{BFR,max}}$, calculation is provided in [D.4](#).

5.3.7 Transmission retarder

Reserved.

5.3.8 Rotating eddy current brake

Reserved.

5.4 Adhesion independent brake equipment type characteristics

5.4.1 Magnetic track brake

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

- the friction coefficient of the magnetic track brake segments in contact with the rail;
- the attraction force, F_{AMg} , which itself depends on the magnetic characteristics of the pole shoe and track materials.

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

$$F_{\text{rMg}} = F_{\text{AMg}} \cdot \mu_{\text{Mg}} \quad (24)$$

where

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

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

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

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

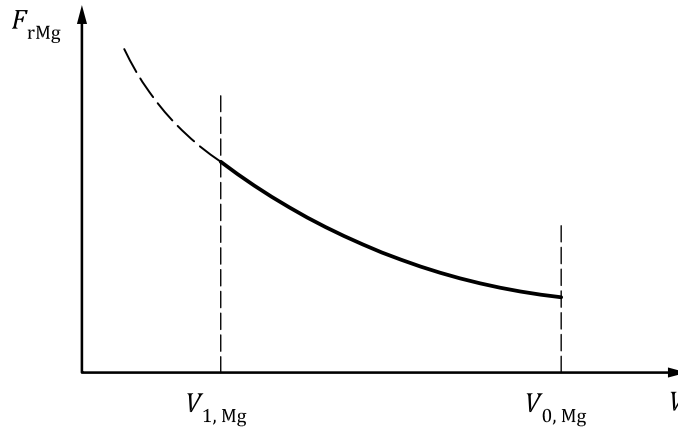


Figure 12 — Indicative retarding force of one magnet of a magnetic track brake vs. speed

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

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

An example calculation for the retarding force of a magnetic track brake is provided in [D.1](#).

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

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

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

where

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

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

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

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

5.4.2 Linear eddy current brake

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

- the gap between the magnets of the eddy current brake and the head of the rail;
- intensity of the magnetic field (magnet coil and excitation current);
- operating speed range including fade in/out ramps.

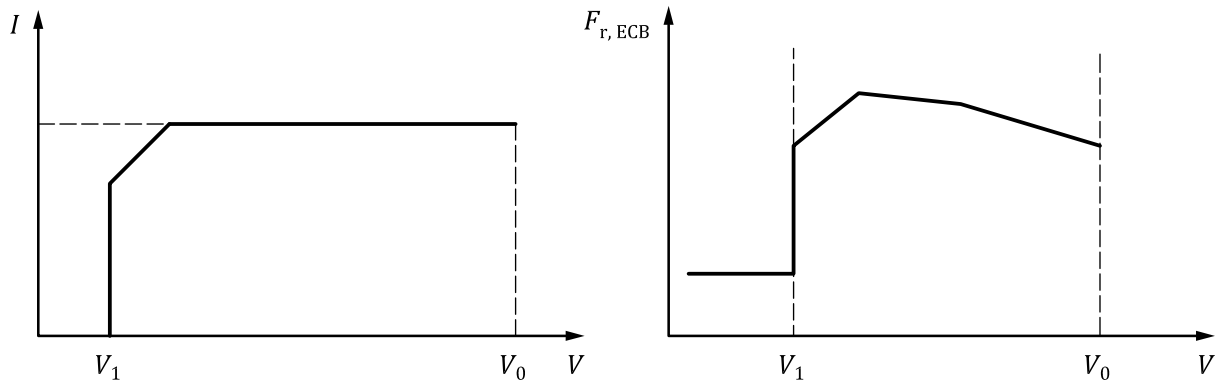


Figure 13 — Typical operating current and retarding force of linear eddy current brake vs. speed

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

An example for retarding force of linear eddy current brake calculation is provided in [D.2](#).

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

5.4.3 Mean retarding force by train resistance

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

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

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

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

where

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

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

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

where

- $\bar{F}_{Ra}$ is the mean retarding force by train resistance, expressed in N;
- C_1 is the characteristic coefficient of the train independent of speed, expressed in N;
- C_2 is the characteristic coefficient of the train proportional to the speed, expressed in $\frac{\text{N}}{\text{m/s}}$;
- C_3 is the characteristic coefficient of aerodynamic resistance due to pressure drag and skin friction drag, expressed in $\frac{\text{N}}{(\text{m/s})^2}$;
- v_0 is the initial speed, expressed in m/s;
- v_{fin} is the final speed, expressed in m/s.

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

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

5.4.4 Aerodynamic brake

Reserved.

5.4.5 Electro dynamic brake generated by linear induction motor

Reserved.

5.5 Time characteristics

5.5.1 Derivation of brake equipment time characteristics

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

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

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

5.5.2 Equivalent response time, t_e

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

- a period when the brake equipment is not applied, and
- a period when the brake equipment is fully applied.