Aspects regarding braking process of passenger trains with different braking systems in composition

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Aspects regarding braking process of passenger trains with different braking systems in composition

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Abstract. The paper investigates the influence of different braking systems on the braking process of railway passenger vehicles, considered individual, as well as in train composition. The study focuses on the braking capacity – evaluated by stopping distance – and on the longitudinal dynamic of train, involving vehicles fitted with either disc brakes in fast-action, or cast iron block brakes in high power action mode. The specific dependence of the correspondent friction engenders particular braking characteristics. If associated to the limited braking forces by wheel-rail adhesion, different stopping distances result, while time evolution of braking process generates specific response of in-train forces, depending on vehicles’ combination in the train body. The study is based on MATLAB numerical simulations. For higher accuracy, experimental data of air pressure evolution in brake cylinders were adequately implemented. Simulations were performed for trains consisting of five coaches on four axles hauled by a six axle locomotive. All possible combinations of braking characteristics in the train composition were considered. Compression and tension forces in couplers and stopping distances are presented as results. The current study continues our previous research in this area. To our knowledge, such aspects of braking process in passenger trains were not investigated by other authors.

1. Introduction
It is generally agreed that braking systems are crucial for railway traffic safety and actual tendencies to increased tonnages and higher velocities put more pressure. International and national regulations cover essential aspects regarding brake systems, braking performance and operational problems.

Traditionally, the UIC compressed air brake [1] is the main system in use, given the well-known advantages, such as high reliability and safety in operation. Still, it is to mention that factors like the pneumatic command and action, associated to mechanical transmission of forces through the brake rigging, the dependency on wheel-rail adhesion, etc. makes braking a very complex process.

Trains are composed of individual vehicles coupled by elastic-dampened elements. As a response to the specific constructive and operational particularities, during braking actions, longitudinal forces develop in the train body between neighbouring vehicles. Under certain conditions, these forces may reach quite high levels, jeopardizing the traffic safety and affecting even the passengers’ comfort or the integrity of transported goods. So it is justified the special interest of specialists and researchers regarding the longitudinal dynamic of trains during braking actions and scientific concerns in this area cover more than 70 years.

Theoretical approaches, experiments and complex simulation programs, e.g. [2-9] are essential to increase the level of knowledge and also to respond to constructive, functional and operational issues,
in order to improve the safety of traffic. An interesting and useful overview regarding the development of simulation programs for longitudinal dynamic of trains is performed by Maksym Spiryagin et al. [10], pointing out the main problems and the development direction needed. Based on the comparison of nine simulators presented in a dedicated benchmarking project, Qing Wu et al. ranked them according to the computing speed [11].

It is clear that instantaneous difference in decelerations among the vehicles composing the train generate compression or tensile forces in couplers and the main forces implied in generating such behaviour are individual braking forces developed by each vehicle. In the first stages following a braking action command, the successive actuation of air distributors in the long of the train and the particular increase gradient of the air pressure in brake cylinders, determined by the mandatory filling characteristics [1], are the determinant sources of longitudinal dynamic forces, generating also oscillatory movements of the vehicles in the train body. As the pressures in the brake cylinders of the vehicles equalize, oscillations diminish and vanish and so the forces in couplers.

Such specific pattern is usually valid if the component vehicles of the train are featured with identical braking systems. That is the general case of freight trains, on which most interest was shown and many studies were elaborated, see for instance [3, 4, 7, 12-13].

In the case of passenger vehicles, to deal with higher speeds, the main braking system is more sophisticated [14]. According to the designed top speed, there are in use systems based on brake shoes or disc brakes, the latter being compulsory beginning with 160 km/h.

Different friction properties between cast iron and wheel tread, respectively plastic pads and disc, conduct to different braking forces versus speed evolutions. More than that, in order to increase the braking capacity when cast iron is involved, high power brake becomes mandatory. That system provides two levels of air pressure in the brake cylinders: a higher one until the speed falls under a threshold limit, followed by a lower one until stopping.

Providing different forces at same speed, in trains composed of vehicles equipped with different brake systems, supplementary longitudinal forces shall develop and overlap on the previous discussed ones. Such combinations of vehicles are quite usual in passenger trains operations.

More than that, passenger vehicles are mandatory featured with wheel slide protection devices to protect wheels against excessive sliding during braking in case of impaired adhesion. The operating principle is based on correspondent reductions of air pressure in the brake cylinders of the vehicles affected. In such situations, as a consequence, important random variations of vehicles’ braking forces occur, in different places in the train, in various moments. This results also in supplementary dynamic reactions, affecting the whole longitudinal dynamic of the train [15].

The arguments given above, even summarily presented, reveal an increased complexity of the braking process in the case of passenger trains and provided reasonable grounds for a thorough research regarding these aspects.

Consistent with our purpose of enhancing understanding the braking process of passenger trains and continuing our previous research [16], the aim of the current paper is to investigate the influence of different braking systems on the braking process of railway passenger vehicles, considered individual, as well as in train composition.

The study is focused on the braking capacity – evaluated by stopping distance – and on the longitudinal train dynamics, involving vehicles fitted with either disc brakes in fast-action, or cast iron block brakes in high power action mode, as is commonly found in operation, at least at Romanian Railway.

The specific dependence of the friction coefficients between composite pads and discs, respectively between cast iron shoes and wheel tread is considered. Associated to the limits of the braking forces determined by wheel-rail adhesion, different stopping distances are obtained, while time evolution of braking process generates specific response of in-train forces, depending on vehicles’ combination in the train body.

The study is based on numerical simulations and, consistent with the intended purpose, we considered necessary two distinct stages to be followed. The first one refers to individual vehicle
submitted to braking action from specific top speed. The aim was to determine the maximum braking force conditioned by wheel-rail adhesion, in accordance with the particularities of each braking system that was considered: high power action and cast iron shoes, respectively fast action and disc brake with composite pads. The subsequent simulation stage refers to the train assembly submitted to braking action.

The main goal of these sets of simulations was to determine the effects in level and evolution of in-train forces when vehicles equipped with different braking systems are combined in the same train composition.

The train was considered a system of rigid mases, connected by couplers defined by specific elastic and damping characteristics. For higher accuracy, experimental data regarding the air pressure evolution in brake cylinders were adequately implemented into the simulation program. Resistance forces were also considered.

Simulations were performed for trains consisting of five weight-identical coaches on four axles hauled by a six axle locomotive. All possible combinations of braking characteristics in the train composition were considered.

The results of the numerical simulations indicate important differences in braking performances both for vehicles considered individually and for various train compositions. Combinations of vehicles equipped with different braking systems generate higher in-train forces and the whole longitudinal dynamic of the train is affected. Compared with trains in uniform composition, maximum levels of buff and draft forces increase and different couplers are more affected, according to the position of vehicles in the train body. Another conclusion regards the influence on time evolution of in-train forces during braking actions.

Based on the results, certain patterns in longitudinal dynamic are suggested and recommendations for optimal disposition of vehicles in passenger trains operation are formulated.

2. Particularities of main brake systems for railway passenger vehicles

The main braking system, compulsory for railway vehicles, has to be able to provide consistent and controllable retardation forces covering the entire speed range between the designed top speed and 0, enabling speed reductions, stopping at a fixed point, maintaining constant velocity in traffic on slopes and preventing vehicles displacements while stationary on gradients. It is also mandatory for the system to provide automatic action, meaning that braking action have to be triggered off whenever train breakage or actuation of the emergency signal occur, independent of the driver’s actions.

Traditionally, rolling stock is equipped with UIC compressed air brake system and, in particular, for the passenger vehicles fast-acting (P) type is required. The electropneumatic brake system is also admitted, provided that correct operation is ensured even by pneumatic control only.

The operational principle of UIC pneumatic brake is simple and, therefore, very reliable. The system is ready-for-use and brakes are released when a working pressure (usually 5 bars) is established; braking actions are commanded by pressure reductions. According to the variations in the brake pipe, the air distributors establish on each vehicle the correspondent pneumatic connections to increase/decrease the air pressure in the brake cylinders.

Given the pneumatic braking operational particularities, the length of the train, associated to the air compressibility and to the location of the driver’s brake valve at the head of the brake pipe, the braking and release commands are transmitted with limited speed. Consequently, the air distributors react successively in long of the train and differences in braking forces between the vehicles result in longitudinal dynamic forces.

In order to diminish that phenomenon, particular gradient of air pressure variations in brake cylinders are provided by vehicles distributors. In the case of fast-acting UIC brake, international regulations require a filling time of the brake cylinders for individual vehicles of 3...5 s and a draining time of 15...20 s, while the maximum pressure obtained must be 3.8 ± 0.1 bar [1].

The main brake systems are friction-based and usually featured with cast iron or composite brake blocks acting on the wheel tread, or with composite pads acting on discs. Considering \( \Sigma N_i \) the total
application forces (clamping forces) on brake blocks or pads and $\mu$ the correspondent friction coefficient, the braking force can be described based on Columbus low [14-16]:

$$F_b = k \cdot \mu \cdot \sum N_i$$ (1)

In (1), $k$ is a factor depending on constructive characteristics particularities of each braking system.

Generally, the friction coefficient depends on numerous parameters and some of the most important that can be controlled are the running speed, the clamping force (applied normal force), the specific pressure between block and wheel. It is a well-known fact that the friction materials used in railway vehicles braking have different behaviours that definitely are influential in the braking capabilities (see figure 1) [16].

As wheel-rail adhesion dependent systems, the maximum affordable braking force (considered at wheel-rail contact level) should not exceed the adhesion force $F_a$ depending on the coefficient of adhesion $\tau_a$ and the mass of the vehicle, $g$ representing the gravitational acceleration:

$$F_{b, \text{max}} \leq F_a = m \cdot g \cdot \tau_a$$ (2)

Given the specific friction coefficient – running speed dependency of cast iron shoes (see figure 1), the stopping distances may exceed the limits required by traffic safety conditions. This is the main reason for limiting the constructive speed to less than 160 km/h for vehicles equipped with cast iron brake blocks. If necessary, generally for top speeds higher than 120 km/h, the braking capacity is enhanced by using the high power braking system.

![Figure 1. Friction coefficient dependency on velocity, application force $P_s$, specific pressure $p_s$.](image1)

![Figure 2. Operational principle for high power brake featured with cast iron brake blocks.](image2)

The high power action brakes operate with two stages of applied forces on the brake blocks, depending on the actual velocity of the vehicle. At the beginning of braking, a high pressure in the cylinders is applied and maintained constant as long as the vehicle’s speed is greater than a threshold velocity. As soon as it falls below the predetermined value, a changeover of the air pressure within the brake cylinders becomes effective and the low level pressure is kept afterward steady until the vehicle has stopped. Consequently, the vehicle benefits from a higher average braking force during the process (see figure 2), in the limits of the same wheel/rail adhesion [16].

In the case of UIC high power brake featured with brake blocks, regulations stipulate for trailed vehicles that maximum air pressure in brake cylinders should be $p_{bc, \text{max}} = 3.8 \pm 0.1$ bar (high level), respectively $p_{bc, \text{low}} = 1.9 \pm 0.1$ bar (low level).

Very significant on the evolution of braking process, with crucial impact on each vehicle and, especially, for the longitudinal behaviour of trains, is the filling characteristic of brake cylinders.
During the filling time, it is a first stage characterised by rapid pressure increase up to about 0.4 bars, limit beyond which is generally accepted that effective application forces begin to act between brake blocks or pads and wheel tread or discs. Up to that limit, the force generated by the growing pressure on the brake cylinder piston has to overcome friction, other resistant forces, initial clearances, inertia of the brake rigging kinematic chain etc. In fact, the end of this stage indicates the beginning of the vehicle’s braking effect. It is followed by a continuous increase of the air pressure in the braking cylinders up to the commanded level. The pressure evolution is graduated by the air distributor in respect to the required filling time. Next, the pressure normally remains invariant, unless release command occurs. An exception is for high power brake system: when speed drops below the threshold value, the process continues with steady low pressure in cylinders.

3. Theoretical aspects

3.1. Movement equation of individual vehicle

In the case of a running railway vehicle submitted to a braking force \( F_b \), considering \( x \) the distance covered, \( W \) the total resistance to forward movement and \( \rho \) the higher than one mass factor accounting the inertia of the rotating masses, the equation describing the displacement is:

\[ F_b(t, x) + W = \rho \cdot m_v \cdot \ddot{x} \]  

(3)

The retardation force determined by a friction-based brake can generally be evaluated according to (1). For typical mechanical construction of main brake systems, the application forces \( N \) are generated by the air pressure \( p_{bc} \) introduced in the brake cylinders, amplified and transmitted through the brake rigging. This is an articulated bar mechanism characterised by the increment (multiplication) ratio \( i_c \) and a certain mechanical efficiency \( \eta \). The assembly is usually provided with automatic brake-rigging adjuster device to keep in correct limits the geometry of the mechanism, irrespective of inherent wear of components, including the friction elements. Also, the brake cylinder is featured with a back spring, having the main role to help return of the brake rigging elements to initial positions, when brake is released [14].

Within the kinematic chain of brake rigging, both the back spring in cylinders and the brake-rigging adjuster generates resistances, \( F_r \) respectively \( F_k \), that diminish accordingly the forces transmitted to brake blocks or pads.

Considering the typical construction of the disc brake with individual rigging on each disc and automatic brake-rigging adjuster device incorporated in the brake cylinder piston rod, as currently in use, the braking force \( F_{bd} \) of a vehicle having \( n_{bc} \) cylinders of \( d_{bc} \) diameter is [16]:

\[ F_{bd} = n_{bc} \cdot \mu_D \cdot \left( p_{bc} \cdot \frac{\pi \cdot d_{bc}^2}{4} - F_r - F_R \right) \cdot i_r \cdot 4r_m \cdot \eta_r \]  

(4)

In (4), \( \mu_D \) is the friction coefficient between pad and disc – generally independent of other parameters (see figure 1), for calculation purposes considered to a mean value \( \mu_D = 0.35 \), \( r_m \) the rubbing radius and \( D_b \) the diameter of the vehicle’s wheels.

In the case of cast iron shoes, currently the passenger vehicles have symmetrical brake rigging with brake-rigging adjuster on the main brake bar. Taking into account the current constructive solutions, the total increment ratio depends on the central brake rigging multiplication ratio \( i_c \), on the amplification ratio of vertical levers \( i_v \), and on the number of triangular axes \( n_\Delta \) [14]:

\[ i_{rc} = i_c \cdot i_v \cdot n_\Delta \]  

(5)

The friction coefficient between cast iron shoe and wheel \( \mu_C \) depends mainly on velocity \( V \), clamping force/shoe \( P \), and specific pressure \( p_s \), in contact area with the wheel tread (see figure 1). Empirical relations for these dependencies were established and one of the most used in calculations is [5, 14]:

5
\[ \mu_C(V, P_s) = 0.6 \cdot \frac{V + 100}{5\bar{V} + 100} \cdot 1.631R_b + 100 \cdot \frac{8.155P_s + 100}{\bar{V}} \]  

(6)

The braking force \( F_{bc} \) of the vehicle is:

\[ F_{bc} = n_{bc} \cdot \mu_C \cdot \left[ \left( \frac{p_{bc}}{\frac{\pi \cdot d_{bc}^2}{4}} - \frac{F_c}{F_R} \right) \cdot i_v \cdot \eta_c \right] \cdot i_y \cdot n_\Delta \cdot \eta_r \]  

(7)

Examining (4) and (7), it can be assumed that certain terms depending on brake rigging design are invariant for the same vehicle, i.e. number and dimensions of discs, shoes, pads, brake cylinders etc. So, it can be defined a generic equivalent term \( \xi \) that includes them, according to the constructive and operational characteristics of the main braking system of each vehicle. Taking into account the direct dependency of the force obtained at brake cylinder piston rod on the actual air pressure within the brake cylinders \( p_{bc} \), (4) and (7) can be summarised in a generalized expression:

\[ F_b(t) = \xi \cdot \mu \cdot p_{bc}(t) \]  

(8)

Knowing the air pressure evolution in the cylinders during the braking process and its maximum level \( p_{bc,\ max} \) and keeping in mind the adhesion condition (2), the instantaneous braking force can be described as follows [16]:

- for vehicles equipped with disc brakes and constructive top speed \( V_{\ max} \):

\[ F_{bd}(t, V_{\ max}) = \begin{cases} 0, & \text{if } p_{bc}(t) < 0.4 \text{ bar, else} \\ \frac{p_{bc}(t)}{p_{bc,\ max}} \cdot m_v \cdot g \cdot \tau_a(V_{\ max}) & \end{cases} \]  

(9)

- for vehicles equipped with \( n_b \) cast iron brake blocks admitting (by adhesion conditions) a maximum \( P_{s,\ max} \) application force on each shoe:

\[ F_{bc}(t, V) = \begin{cases} 0, & \text{if } p_{bc}(t) < 0.4 \text{ bar, else} \\ n_b \cdot \frac{p_{bc}(t)}{p_{bc,\ max}} \cdot P_{s,\ max} \cdot \mu C(t, p_{bc}, V) & \end{cases} \]  

(10)

Regarding the resistances, as theoretical approach, it is more practical to consider separately the propulsion (inherent) resistances \( W_p \) acting permanently whenever the vehicle is moving on a straight uninclined track and the incidental resistances \( W_s \). The latter are intermittent opposing forces, acting in certain conditions i.e. when running on grades, curves, during acceleration cycle, when operating in adverse winds. These are overlapping the propulsion resistances, to which are added up algebraically when appropriate [2, 17-21].

Typically, propulsion resistance \( W_p(\bar{x}, \bar{x}^2) \) is described on the basis of Davis’ equation that take into account mainly the flange, journal and rolling frictional resistance and the air resistance in still air. The calculation of rolling stock resistance to forward movement is still dependent on empirical formulae experimentally determined. For computational reasons, sometimes it is preferred to refer to the specific propulsion resistance \( w_p[N/kN] \), defined in relation to the weight of the vehicle:

\[ w_p = \frac{W}{m_v \cdot g} \]  

(11)

3.2. Train model

Given the complexity of the diverse processes and phenomena ongoing in the train body during the braking process, it is difficult to build a reasonable model being very close to the real behavior in operation. In the case of UIC brake, it is a chain of pneumatic processes which triggers a series of mechanical processes which develop in different moments, with varying intensities, at various places
in the length of the train body. Numerous parameters are involved in the development of the braking process and each comes with particular influences that may potentially affect, more or less, the traffic safety: speed, load, track characteristics, composition and length of the train, constructive, technical and operational characteristics of the vehicles, the type of braking system and particularities of component elements, wheel-rail adhesion conditions, etc.

It is generally agreed that in longitudinal dynamic studies of trains submitted to braking actions, both mechanical and pneumatic models of the train must be developed and, than, suitably combined to reflect the correct succession of events. Problems regarding train models have been presented and discussed in numerous publications as, for example, [2-8, 22-26].

In mechanical perception, the usual train model in longitudinal dynamics is defined as an elastodampened system consisting of individual rigid mases, connected through couplings characterized by elastic and damping characteristics, corresponding to the particularities of buffers and traction devices. Based on a multibody approach, for the "i"-th vehicle of the train, keeping previous notations and considering \( P \) the forces acting in couplers, the differential equation describing its longitudinal movement is (see figure 3):

\[
m_{v,i} \ddot{x}_i + F_{b,i} + W_i - P_{i-1,i} = 0
\]

![Figure 3. Forces acting on generalized “i”-th vehicle of the train during braking actions.](image)

Applying (12) to all the \( n \) vehicles and considering \( P_{0,1} = P_{n,n+1} = 0 \) as boundary conditions, results the equation system describing the longitudinal movements of the vehicles of the train. Knowing the masses \( m_{v,i} \), the time evolution of braking \( F_{b,i} \) and resistance forces \( W_i \) and the elastic and damping characteristics of couplers elements for each vehicle, the in-train \( P_i \) forces can be determined by integration.

For the accuracy of simulations, it is very important to adopt an appropriate model for the coupler assembly, so that the mechanical characteristics to be expressed as close as possible to reality.

The couplers, generally comprising the assembly of shock and traction devices plus the coupling gear, have significant influences on the longitudinal dynamics of the train, with running stability implications, besides the role to protect the vehicles structure and laden against shocks. It is well known that, traditionally, in Europe is preferred the solution with a pair of lateral buffers, a traction and a coupling gear at each extremity of the vehicle couplers, while in the rest, the central automatic couplers solution is widely spread. However, whatever the case, the constructive and operational characteristics, such as variable stiffness-damping, hysteretic properties, preloads of elastic elements, draw-gear compliance, clearance between the buffers discs, etc., have remarkable influence. There are various constructive solutions for shock and traction equipment used on railway vehicles, based on metallic, rubber, silicon type elastomers, hydraulic, pneumatic or hydro-pneumatic elastic elements.

The complexity of buffer and draw-gear response in operational conditions raises obvious difficulties in their mechanical modelling. It is a vast literature in this area, with valuable approaches,
see for instance [27-30]. The comprehensive review of Wu, Cole et al. must be highlighted [31]. Anyway, according to each particular goal, an adequate balance between the accuracy of simulated behaviour, on one hand and computational and time resources on the other hand, has to be maintained.

In the present study, typical constructive solutions for buffers and traction devices with metallic elastic rings (RINGFEDER type) based on friction elements to fulfil the required damping effects are considered. The force-displacement characteristics depend mainly on the stroke (the relative displacement between adjoined vehicles) and relative velocity, associated to its sign. In order to avoid intricacies determined by the discontinuities of friction forces between elastic rings, a smoother approach was preferred [26-27]. The forces in couplers were evaluated assuming an algebraic sum of elastic and friction forces.

According to functional specific action of the elastic elements comprising friction elements, considering $\Delta x_{ij}$ the stroke and $\Delta \dot{x}_{ij}$ the relative speed between the $i$ and $j$ adjoined vehicles, there were defined [32]:

$$P_{ij}(\Delta x_{ij}, \Delta \dot{x}_{ij}) = \begin{cases} k_b \cdot \Delta x_{ij} + c_b \cdot |\Delta x_{ij}| \cdot \tanh(u \cdot \Delta \dot{x}_{ij}), & \text{for } \Delta x_{ij} < 0 \\ 0, & \text{if } \Delta x_{ij} = 0 \\ k_t \cdot \Delta x_{ij} + c_t \cdot |\Delta x_{ij}| \cdot \tanh(u \cdot \Delta \dot{x}_{ij}), & \text{for } \Delta x_{ij} > 0 \\ \end{cases} \quad (13)$$

In (13), $k$ and $c$ are equivalent constants depending on the elastic and friction characteristics between the metallic rings inside, indexes “b” and “t” are referring to buffer and traction devices and $u>>1$ is a scaling factor. Theoretical aspects are more in detail presented in [25, 32].

Such model have the advantage of a much simpler simulation, but also certain limitations, for instance the fact that vehicles in contact tend to drift until the resultant force vanishes, which does not exactly correspond to the operational effects [27, 32-33].

As regard the pneumatic perception, fundamentally is that braking commands propagate with limited transmission speed along the train and the braking system of each vehicle specifically interacts with the complete pneumatic plant of the train through the air distributors.

Two kind of events are characteristic and essential for the evolution of in-train forces: the actuation moment of the air distributor on each vehicle of the train, related to moment of braking command through the driver’s valve; the subsequent evolution of the air pressure in the brake cylinders of each vehicle, depending on the filling time and filling characteristic provided by the air distributors. Accordingly, there are basically two pneumatic processes that are usual carefully considered in models for simulations: the propagation of braking signal along the brake pipe and the response of the distributor of each vehicle [3-4, 6-7, 34-35].

In the current study, given that passenger trains are generally short, a constant rate for the brake signal propagation in the long of the train was considered. As regard the air pressure evolution in brake cylinders after the distributors’ actuation, experimentally acquisitioned data were adequately implemented into the simulation program.

4. Numerical simulation

The simulation program destined to individual vehicle braking process basically integrates (3). The main input data refer to: top speed, mass, adhesion coefficient, particularities of braking system, propulsion resistances. The main outcomes are materialized in numerical values of stopping distance, braking force, maximum instantaneous and mean decelerations and also diagrams depicting time or speed evolutions of braking forces, brake distances and decelerations.

The train longitudinal dynamic simulation program basically integrates the system associated to equation (12). The main input data refer to: train composition (number of vehicles, positions, combination of vehicles in train body), vehicles’ constructive, technical and operational characteristics (length, top speed, tare, laden, type and specific features of buffers and traction devices, propulsion resistances), braking system of each vehicle (fast or high power action, number of brake blocks, number of discs, maximum designed clamping force/brake shoe or braking force), pneumatic data
(maximum air pressure in brake cylinders, filling time, filling characteristics), operational data (speed at the beginning of brake action, type of braking action, track characteristics). The main results consist of numerical values of maximum compression and tensile forces in all couplers and diagrams presenting the time evolution of in-train and braking forces, relative displacements between vehicles. As already accustomed, in all results, the buff (compression) forces are positives, while the draft (tensile) forces are negatives.

4.1. Main assumptions and input data

Consistent to the aim of the study and given the multiple factors influencing the braking process, certain constraints and simplifying hypotheses were assumed in order to highlight the effects of analysed systems and to facilitate a target-oriented analyse of the results, by eliminating, as much as possible, the potentially interferences with the goals enunciated.

Some of the most relevant are outlined as follow:

- coaches are four axles passenger vehicles, identical in terms of geometrical dimensions (length 25 m), total weight (50 t) and inherent resistances;
- traction vehicles are six axles electric locomotives (length 25 m, weight 120 t);
- vehicles have couplers consisting of RINGFEDER type lateral buffers and traction devices;
- vehicles are equipped with UIC brake system;
- disc brakes featured vehicles are designed for 160 km/h and equipped with KE-1 type air distributor operating in fast-acting (P) mode;
- cast iron brake block featured vehicles are designed for 140 km/h and equipped with KEs type air distributor operating in high power (R) mode;
- in the case of high power mode, the threshold velocity actuating the high/low pressure level changeover is 50 km/h;
- trains are considered in classical composition: a locomotive pulling five coaches;
- emergency braking action is performed on straight uninclined track at top designed speed for individual vehicles, respectively 140 km/h for trains;
- vehicles are considered to have identical filling characteristics and to reach the same maximum pressure in brake cylinders;
- for trains, screw couplings are tightened up, with no clearance between the buffer discs.

The basic data referring to the braking systems featuring the vehicles are synthetized in table 1.

Table 1. Main braking characteristics and input data.

| Brake type           | Operating mode | Distributor | Discs /axle | Brake shoes /axle | Filling time [s] | Max. pressure in brake cylinders [bar] |
|----------------------|----------------|-------------|-------------|-------------------|-----------------|----------------------------------------|
| disc brake           | fast-action (P)| KE-1        | 2           | -                 | 3.34            | 3.896/3.896                            |
| cast iron shoes      | high power (R) | KEs         | -           | 8                 | 3.34            | 3.896/1.803                            |

In both simulation programs, the braking forces are determined using equation (9) for disc brakes, respectively equation (10) and friction coefficient given by (6) for cast iron equipment. The wheel-rail adhesion coefficient in dry, normal conditions is considered by the dependence of speed $V$ [km/h]:

$$\tau_a = \frac{0.33}{1 + 0.011 \cdot V}$$

(14)

For couplers, the specific constants in (13) are: $k_b = 2.8 \times 10^6$ N/m and $c_b = 1.4 \times 10^6$ N/m (for buffers); $k_t = 5.46 \times 10^6$ N/m and $c_t = 2.43 \times 10^6$ N/m (for traction devices); the scaling factor $u = 10^4$.

Regarding the air pressure evolution in cylinders during emergency braking action, experimental acquisitioned data were adequately implemented in the simulation programs. Measurements were performed in the Laboratories of Faculty of Transport in University POLITEHNICA of Bucharest for KE 1b and KEs air brake distributors (in current use on Romanian rail vehicles). Samples of recorded evolutions are presented in figure 4.
The propulsion resistance of the locomotive \( W_{\text{loc}} \) [N] is considered by the empirical relation fitted for the majority of six axle electric locomotive in Romanian fleet:

\[
W_{\text{loc}} = 2903.76 + 69.337 \cdot V^2
\]

(15)

The specific propulsion resistance of the passenger vehicles \( w_v \) [N/kN] is considered:

\[
w_v = 1.65 + 2.5 \cdot 10^{-4} \cdot V^2
\]

(16)

4.2. Results and comments on first stage of numerical simulations

Considered the main goal of this stage of simulations, the maximum braking force \( F_{bD,\text{max}} \) for vehicles with disc brakes and the maximum clamping force/shoe \( P_{s,\text{max}} \), were determined, in respect to the wheel-rail adhesion condition (2) and according to the designed top speed \( V_{\text{max}} \). As outputs, the main emphasis was put on the stopping distances \( S_b \), the maximum \( d_{\text{max}} \) and mean \( d_{\text{m}} \) decelerations, as useful in braking capacity evaluations. Numerical results are synthetized in table 2.

Table 2. Numerical results for individual emergency braking action.

| Vehicle | Brake type | Operating mode | \( V_{\text{max}} \) [km/h] | \( F_{bD,\text{max}} \) [kN] | \( P_{s,\text{max}} \) [kN] | \( S_b \) [m] | Deceleration [m/s²] |
|---------|------------|----------------|--------------------------|--------------------------|--------------------------|----------------|----------------------|
| locomotive | disc | P | 160 | 140.75 | - | 877 | 1.24 | 1.16 |
| | shoes | R | 140 | - | 21.5 | 1000 | 1.295 | 0.835 |
| coach | disc | P | 160 | 58.65 | - | 913 | 1.175 | 1.148 |
| | cast | R | 140 | - | 12.5 | 836 | 1.42 | 0.951 |

It is interesting to point out that results highlight two aspects: typical friction particularities of cast iron shoes lead to a lower braking capacity; higher maximum decelerations associated to smaller mean effective decelerations indicate a decreased comfort for the passengers.

Differences in braking capacities can be better underlined by analysing the stopping distances corresponding to the same initial velocity, 140 km/h (see table 3). Shorter stopping distances in the case of disc brake featured vehicles, for identic conditions, shows that in mixed composition trains, higher in-train forces should be expected.

Table 3. Stopping distances in emergency braking from same initial speed.

| Vehicle | Brake type | Mode | Initial speed [km/h] | \( F_{bD} \) [kN] | \( P_s \) [kN] | \( S_b \) [m] |
|---------|------------|------|---------------------|----------------|----------------|----------------|
| locomotive | disc | P | 140 | 140.75 | - | 688 |
| | shoes | R | 140 | - | 21.5 | 1000 |
| coach | disc | P | 140 | 58.65 | - | 711 |
| | cast | R | 140 | - | 12.5 | 836 |
4.3. Results and comments on second stage of numerical simulations

The second stage of research is dedicated to the longitudinal dynamic of six vehicles classical passenger trains, considering all possible combinations of disc/cast iron brake system featured vehicles in the train composition. This results in a number of 64 cases, referred to by a combination of six capital letters, according to the arrangement in the train body of the vehicles equipped with disc brakes (D), respectively cast iron shoes and high power brake (R).

The main results cover, as numerical values, the maximum buff and draft forces in couplers, diagrams presenting the in-train and braking forces evolutions and the stopping distances for the train in different braking systems configurations.

The maximum in-train forces in all the studied cases are condensed in the histogram presented in figure 5, showing an increase of forces in couplers as a result of having vehicles with different braking systems in use in the same train. Their number and, especially, position in the train body, are crucial. Maximum buff forces are less affected by different braking systems combinations and only few configurations conduct to certain pics of tensile forces in the train body. On the other hand, different braking characteristics of the vehicles are more influential on draft forces.

![Figure 5. Maximum in-train forces in all braking systems combinations.](image)

Moreover, analysing the simulations outcomes, it is clear that increased forces in couplers is an important effect of combining vehicles with different braking systems in use in the same train.

To better clarify this aspect, among the results of all the 64 simulations performed, there were selected, as representative combinations, the cases involving the maximum buff and draft forces, corresponding to the ratio between the number of vehicles equipped with cast iron brake blocks (R) and those with disc brakes (D), in the train composition. These results are presented in table 4.

### Table 4. Maximum in-train forces.

| nr. veh. R\(^a\) | coupler combination | Buff forces [kN] | Draft forces [kN] |
|-------------------|----------------------|------------------|-------------------|
|                   | 1 - 2    | 2 - 3    | 3 - 4    | 4 - 5    | 5 - 6    | 1 - 2    | 2 - 3    | 3 - 4    | 4 - 5    | 5 - 6    |
| 0 / 6 | DDDDDD | 29.2 | 34.7 | 35.9 | 33.8 | 26.2 | -11.8 | -11.0 | -9.1 | -6.0 | -3.2 |
| 1 / 5 | RDDDDD | 56.8 | 44.6 | 35.9 | 35.0 | 27.0 | -46.6 | -40.1 | -31.6 | -21.7 | -20.0 |
| 2 / 4 | RRDDDD | 27.2 | 37.5 | 35.8 | 35.1 | 27.4 | -39.3 | -51.4 | -39.6 | -28.4 | -21.9 |
| 3 / 3 | DDRRRR | 29.0 | 36.7 | 43.7 | 45.7 | 25.8 | -14.8 | -21.3 | 28.3 | -34.8 | -17.4 |
| 4 / 2 | DDRRRR | 27.2 | 33.7 | 37.4 | 35.3 | 27.8 | -29.6 | -38.7 | -48.2 | -32.3 | -17.3 |
| 5 / 1 | DRRRRR | 39.5 | 51.5 | 59.0 | 43.4 | 26.3 | -22.0 | -31.5 | -41.6 | -28.0 | -13.9 |
| 6 / 0 | DRRRRR | 47.5 | 62.3 | 50.9 | 35.6 | 26.3 | -29.3 | -41.6 | -30.7 | -20.2 | -10.0 |

\(a\) Vehicles equipped with cast iron brake blocks, high power action mode.

\(b\) Vehicles equipped with disc brakes, fast action mode.
In the assumed conditions, it is to notice that the maximum compression force occurs when the locomotive and first coach are equipped with disc brakes (D) and the other passenger vehicles are featured with cast iron brake shoes (R), while the maximum tensile force occurs in an opposite combination (RRDDDD). It is also to observe a tendency to higher levels of longitudinal forces in first half of the train (couplers 1–2, 2–3 and 3–4).

In the case of uniform composition of the train (DDDDDD, respectively RRRRRR), the maximum in-train forces in couplers are very close in value. Considering these cases as baseline, combinations of brake systems may result in almost doubling the buff forces, respectively in more than four times higher draft forces. Anyway, it is still to notice that in the studied cases, the couplers are still far from breaking limits, so the traffic safety is not affected.

We considered also important to analyse the time evolution of in-train forces, to find out when namely occur the maximum in-train forces during the braking process. In figure 6 are presented these evolutions for the previous discussed representative cases.

![Figure 6](image_url)

**Figure 6.** Time history of in-train forces for six vehicles classical train composed of rolling stock with different systems during emergency braking action from 140 km/h.

It is to notice that in case of uniform train composition regarding the braking systems (disc brake only), the in-train forces follow the classic pattern (see figure 6, a): important compression develops almost immediately after the braking command, continued by a rapid decrease during cylinders filling time period. Train recoil follows, with increasing tensile forces in couplers and oscillatory movements of the vehicles within the train body, rapidly diminished due to the strong damping capacity of the...
couplers elements. Further, the train experience a steady light lowering compression until stopping. In the considered train configuration, the maximum buff force acts at the middle of the train (coupler 3-4) during the first second of the braking process, while the highest draft force develops few seconds later between the locomotive and the first coach (coupler 1-2).

In the other presented situations, evolutions of in-train forces are significantly different, even if the compression during the first seconds is almost identical (see figure 6, b and c). The continuous evolving differences in braking forces, evident in cases b and c depicted in figure 7, determine the specific correspondent response of the longitudinal dynamic of the train. It is also obvious the effect of high/low pressure modification in brake cylinders at 50 km/h for the vehicles equipped with high power system and cast iron brake shoes.

**Figure 7.** Time history of braking forces in the case of six vehicles classical train composed of rolling stock with different systems during emergency braking from 140 km/h.

During the braking action, if front vehicles are equipped with R (high power, cast iron shoes) system, mostly draft forces act in couplers; if the front vehicles have disc brakes, during the braking process, the train is mostly compressed (see figure 6 b and c). It is also to underline that, in no uniform brake systems composition, the highest in-train forces may occur either around the speed of 50 km/h, due to the change high/low level pressure in brake cylinders, of just before the train stops. This assertion is sustained not only by the values and diagrams presented in table 4 and figure 6, but also by the results of all 64 simulations performed.
As regard the stopping distances, a synthesis of the results are presented as diagrams in figure 8.

![Stopping distance for six vehicles classical train composed of rolling stock with different systems, submitted to emergency braking from 140 km/h.](image)

Figure 8. Stopping distance for six vehicles classical train composed of rolling stock with different systems, submitted to emergency braking from 140 km/h.

It can be seen that braking systems designed according to the wheel-rail adhesion condition have important influence on trains stopping distance. In our case, in uniform braking systems composition, there is a consistent difference between 687.1 m (only disc brake – D) and 839.2 m (only high power with cast iron brake shoes – R), meaning about 20 % shorter for disc brake equipment. More vehicles featured with cast iron brake blocks in train composition results in increased braking distance.

These aspects were predictable if taking into account the results of our first stage of numerical simulations (see tables 2 and 3). One interesting observation is referring to the case of cast iron equipment: higher weight and wheel-rail adhesion condition makes more influential the effect of the locomotive on the stopping distance, which is 6...7 % longer compared to all situations when the pulling vehicle has disc brakes (see figure 8).

It is to underline that, while the braking system of each vehicle is designed to cover all requirements according to the destined top speed and, in mixed composition, the maximum admitted speed of the train is given by the less performant one, any presence of disc brake featured vehicle together with rolling stock equipped with cast iron shoes enhances the total braking power of the train assembly. So, the discussed combinations do not affect the traffic safety in terms of braking capacities.

5. Conclusions

In this research was comprehensively studied the influence of different braking systems on the braking process of railway passenger vehicles, considered individual, as well as in train composition, using numerical simulations. Six axle locomotive and four axle passenger coaches equipped with either disc brakes or high power system with cast iron brake blocks were considered, operating single or as classical six vehicles train. Constructive and operational specific mechanical, as well as pneumatic characteristics were taken into account and there were not neglected the propulsion resistances and the limits imposed by wheel-rail adhesion to maximum designed braking forces. Regarding the train composition, all the 64 possible combinations of vehicles featured with the above enunciated braking systems were supposed.

As results of the research, stopping distances and decelerations are the main outcomes for individual vehicles, while for the train, the interest was focused mainly on the longitudinal compression and tensile forces acting in couplers, as magnitude, position and time evolutions.

The study reveals that higher braking power of vehicle equipped with discs and the different, less advantageous braking characteristics specific for high power system with cast iron brake blocks, result in notable modifications of the entire longitudinal dynamics if associated in the same train composition. Such combinations determine important increase of in-train reactions, more significant
for the tensile forces than for the compression ones. Most affected couplers are those between the
vehicles situated in the first half of the train. During the process, the maximum longitudinal forces
occur either in short time after the braking command, or when the running speed falls below 50 km/h
and even just before the train stops. The arrangement of the vehicles with different braking systems in
train is essential in this respect.

As regard the traffic safety, in all considered situations, the longitudinal forces in couplers are low
enough that there are no reasons to inquire about train break apart and the stopping distances are in the
required limits.

Still, it is to highlight that in the case of combinations of the discussed brake systems in the body of
six vehicles train, grouping the identical equipped vehicles at the front or at the end of the train will
induce the highest longitudinal reactions.

Even if the outcomes of our research seem not very spectacular in terms of traffic safety for
passenger trains operations, we consider that it opens an interesting frame for scientific knowledge and
understanding of the complex process of braking.

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