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SIMULATION OF THE LONGITUDINAL DYNAMIC FORCES DEVELOPED IN THE BODY OF PASSENGER TRAINS

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Abstract: This paper focuses on a simulation program for evaluating the longitudinal dynamic forces operating upon the traction, collision and coupling devices during the train braking process.The model of a train comprising of an engine and ten coaches under emergency braking from the maximum velocity of 160 km/h has been presented. The numerical results including the values of the forces and strokes occuring on the buffers and the traction device are discussed.
Keywords: longitudinal dynamics of train, brakes, traction, collision and coupling devices

1. INTRODUCTION

Train braking is a quite complex phenomenon, specific to the railway vehicles, with a major impact on the circulation safety. The complexity of the braking process derives from the succession of the mechanical, pneumatic and thermal phenomena occuring on each train coach, during the braking action and with different forces. For the classical braking system, there is always a time delay between the reaction of the front coach and the one in the rear, due to the air compressibility. According to that, the propagation speed of the braking wave will lead to the successive proceeding of the distributors in the train body so that the front coaches are braked and the rear with no braking force developed will hit the braked ones, thus having the longitudinal dynamic phenomena emerge. This action could bring harm to the comfort of the passengers, the integrity of the transported merchandise and, sometimes, to the circulation safety.

The evaluation of the longitudinal dynamic forces depends on a series of parameters, such as: the exploitation conditions of the coaches, the distribution of masses into the train body, the building and functional parameters of the traction, collision and coupling devices, the length of the general pipe, etc. Therefore, a study on the evaluation of the longitudinal dynamic forces to consider all the parameters is rather complicated to devise [1].

Nevertheless, a series of studies on the longitudinal dynamics of the trains under braking action has been conducted, with the explicit purpose to find ways of reducing the longitudinal forces present during braking. *Fukazawa* [2] has calculated the dynamic forces in the traction, collision and coupling devices for the freight trains comprising of two two-axle carloads and reached the conclusion that the emergency braking is not recommended at velocities smaller than 30 km/h so as to avoid the occurence of the high compression forces on these apparatuses. *Zobory* introduces a study concerning the simulation of the longitudinal forces from the track geometry [3] and later carries out the simulation of the longitudinal dynamic forces due to the collision between two trains [4].

Theoretical studies on this type of forces developed within the train bodies under braking have been run by *Karvatski* [5], *Pugi, Fioravanti* şi *Ridi* [6], *Cole* [7], *Belforte, Cheli* [8], *Nasr* and *Mohamndi* [1], *Zobory* [3, 4] for freight trains and *Cruceanu* [9…11] and *Oprea* [12…14] for the passenger trains.

This paper deals with the evaluation of the longitudinal dynamic forces and the relative strokes between the consecutive wagons in a train made up of an engine and ten coaches, based on the mathematical modelling of the characteristics of the traction, collision and coupling devices.

2. THE MODELLING OF THE TRACTION, COLLISION AND COUPLING DEVICES

The mathematical modelling of the traction, collision and coupling devices mainly aims to establish a relation for computation of the forces and the strokes in these apparatuses during train braking.

The deformation force of the buffer depends on the variation in the stroke and the relative velocities between the vehicles, the rigidity of the elastic elements, as well as on the damping degree.

To have a mathematical modelling of the force development in the collision device, the characteristic diagram of the buffer equipping the trailed passenger trains has been taken into account, which was built in Romania by ICPVA-SA, (figure 1), and whose elastic and damping elements is of RINGFEDER type (with metallic discs) [15…19].

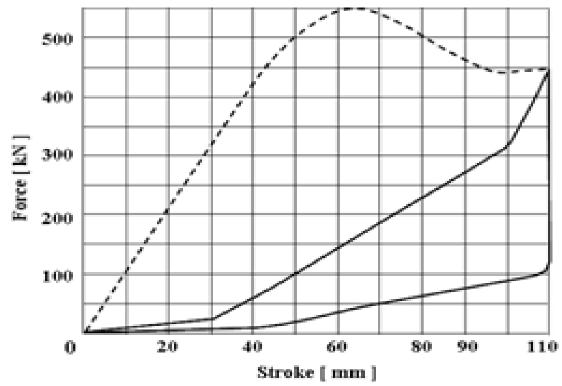

Figure 1. The quasi-static characteristic of the Ringfeder-type buffer used on passenger trains, - the static diagram, - - the diagram in a dynamic behavior [10]

While considering k_e a constant depending on the elasticity of the elements within the buffer and k_f a constant that is a function of the friction between the metallic discs inside of it, the buffer force can be calculated as below:

$$F_t(x,\dot{x}) = \frac{1}{2}\cdot(1+\operatorname{sgn} x)\cdot(k_e x + k_f |x| \tanh(u\cdot\dot{x})) \tag{1}$$

where x represents the stroke of the collision device and $\dot{x}$ its relative speed.

Due to the fact that the coaches are fitted with traction and coupling devices while in traffic, with elastic and damping elements that are similar with the collision's, a mathematical model identical with the buffers' has been considered, looking at the coefficients that define the elasticity kec and the friction k_{fc}, specific to these devices.

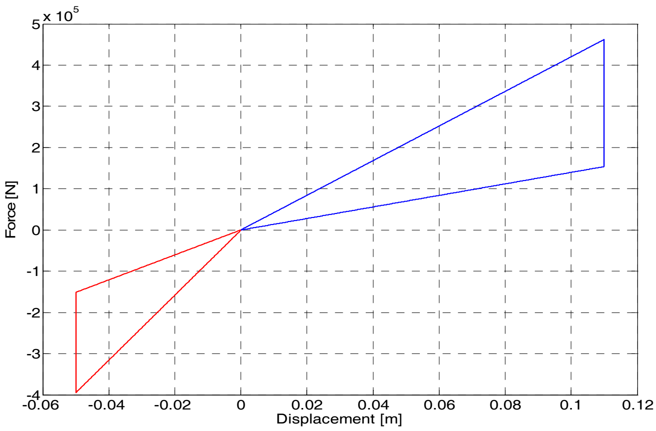

Figure 2. The force-stroke characteristic diagram for the traction, collision and coupling RINGFEDER type device fitting the passenger trains

The force developed in the friction device is as below:

$$F_c(x,\dot{x}) = \frac{1}{2}\cdot(1-\operatorname{sgn} x)\cdot(k_{ec}x + k_{fc}|x| \tanh(u\cdot\dot{x})) \tag{2}$$

where x represents the stroke of the traction and coupling device and $\dot{x}$ its relative speed.

Consequently, the calculation of the forces for the traction, collision and coupling devices follows the below equation:

$$F(x,\dot{x}) = \begin{cases} k_e\cdot x + k_f\cdot|x|\cdot\tanh(u\cdot\dot{x}) & for \quad x<0, \\ 0 & for \quad x=0, \\ k_{ec}\cdot x + k_{fc}\cdot|x|\cdot\tanh(u\cdot\dot{x}) & for \quad x>0 \end{cases} \tag{3}$$

where x is the stroke and $\dot{x}$ the speed of such devices.

Thus, the relation (3) can be used to simulate the operating of the traction, collision and coupling devices and generates the characteristic in figure 2, where for $x > 0$ only the collision device will start functioning, at $x = 0$ the force will be zero for the collision and coupling and for $x < 0$, only the traction and coupling device will be operational.

3. THE MECHANICAL MODEL OF THE TRAIN

To examine the longitudinal forces in the train body under braking, the general model of the train has been established (Figure 3) as a damped elastic system comprising of *n* rigid bodies of mass *m*, representing the train coaches, connected via the collision, traction and coupling devices [23…26].

The following simplifying hypotheses have been considered:

- the initial compression of the elastic elements of the collision and traction devices has been neglected;
- the couplers between the coaches have been tightened up, with no clearance left, in compliance with the regulations for the passenger trains [25];
- the average speed of the braking wave along the train is of 250 m/s, the minimum value stipulated in the regulations [26];
- all the vehicles are fitted with a quick-action brake, where the air distributors in the train body provide the same filling characteristic.

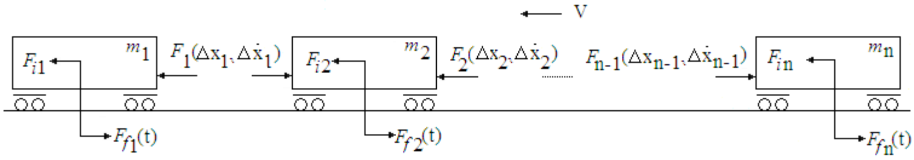

Figure 3. The forces acting on the train

For the train model presented in figure 3, the forces acting on each carriage in the train body are:

- inertia forces $F_{i,1}$, $F_{i,2}$, ... $F_{i,n}$;
- braking forces $F_{f1}(t)$, $F_{f2}(t)$, ... $F_{fn}(t)$;
- forces in the traction, collision and coupling devices $F_1(\Delta x_1, \Delta\dot{x}_1)$, $F_2(\Delta x_2, \Delta\dot{x}_2)$,... $F_{n-1}(\Delta x_{n-1}, \Delta\dot{x}_{n-1})$.

In order to calculate the braking force, it is necessary to know how the air pressure changes in the vehicle's brake cylinder. This has been experimentally determined on the computer-based stand for checking the braking pneumatic equipment of the railway vehicles at the Department of Railway Rolling Stock.

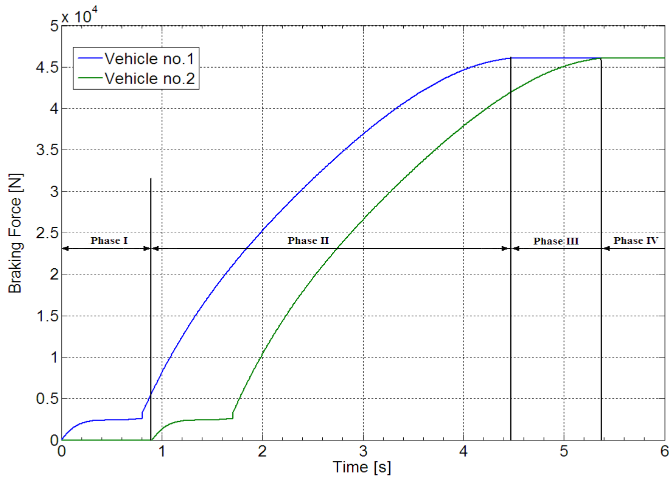

Figure 4. The dynamics of the braking forces in the train body, in a P behavior for emergency braking

The maximum braking force developed by each carriage should not exceed the wheel-rail holding force, according to one of the basic criterion that needs to be met while designing the braking systems of the railway vehicles that is to avoid, under normal circumstances, the blocking of the mounted axles during the braking actions [10, 11, 21].

With the requirement:

$$F_{f,i} \le F_a = \mu_{a,i} \cdot m_i \cdot g \tag{4}$$

where μ_a is the wheel-rail holding coefficient and m_i represents the mass of each carriage in the train body, the development in time of the braking forces can be calculated as such:

$$F_{f,i}(t) = \frac{\mu_a \cdot m_i \cdot g}{p_{cf\,\max}} \cdot p_{cf,i}(t) \tag{5}$$

where $p_{cf\,\max}$ is the maximum pressure occuring in the braking cylinders, $p_{cf,i}(t)$ represents the instantaneous pressure in the braking cylinders, experimentally determined, and g is the gravitational acceleration.

Figure 4 presents the dynamics of the braking forces for the front and rear coaches in the body of a train comprising of 10 identical coaches. The phases of the train braking are featured, with the purpose of pinpointing the differences between the vehicles' braking forces at a certain moment in time.

4. THE ESTABLISHMENT OF THE GENERAL EQUATIONS FOR CALCULATING THE LONGITUDINAL DYNAMIC FORCES

Starting from the facts in the previous paragraph and using the model in Figure 3, the force equilibrium equations for each coach are written as such:

$$\begin{gathered} F_{i,1} + F_1(\Delta x_1, \Delta \dot{x}_1) - F_{f,1}(t) = 0 \\ F_{i,2} + F_2(\Delta x_2, \Delta \dot{x}_2) - F_1(\Delta x_1, \Delta \dot{x}_1) - F_{f,2}(t) = 0 \\ \dots\dots\dots\dots\dots\dots \\ F_{i,i} + F_i(\Delta x_i, \Delta \dot{x}_i) - F_{i-1}(\Delta x_{i-1}, \Delta \dot{x}_{i-1}) - F_{f,i-1}(t) = 0 \\ \dots\dots\dots\dots\dots\dots \\ F_{i,n-1} + F_{n-1}(\Delta x_{n-1}, \Delta \dot{x}_{n-1}) - F_{n-2}(\Delta x_{n-2}, \Delta \dot{x}_{n-2}) - F_{f,n-1}(t) = 0 \\ F_{i,n} - F_{n-1}(\Delta x_{n-1}, \Delta \dot{x}_{n-1}) - F_{f,n}(t) = 0 \end{gathered} \tag{6}$$

where $\Delta x_i = x_i - x_{i+1}$ represent the relative displacement, for the traction, collision and coupling devices.

Upon replacing in the equations (4) the formulas of the inertia forces $F_{i,i} = -m_i \cdot \ddot{x}_i$, where *i = 1…n*, we have:

$$\begin{gathered} m_1 \cdot \ddot{x}_1 = F_1(\Delta x_1, \Delta \dot{x}_1) - F_{f,1}(t) \\ m_2 \cdot \ddot{x}_2 = F_2(\Delta x_2, \Delta \dot{x}_2) - F_1(\Delta x_1, \Delta \dot{x}_1) - F_{f,2}(t) \\ \dots\dots\dots\dots\dots\dots \\ m_i \cdot \ddot{x}_i = F_i(\Delta x_i, \Delta \dot{x}_i) - F_{i-1}(\Delta x_{i-1}, \Delta \dot{x}_{i-1}) - F_{f,i-1}(t) \\ \dots\dots\dots\dots\dots\dots \\ m_{n-1} \cdot \ddot{x}_{n-1} = F_{n-1}(\Delta x_{n-1}, \Delta \dot{x}_{n-1}) - F_{n-2}(\Delta x_{n-2}, \Delta \dot{x}_{n-2}) - F_{f,n-1}(t) \\ m_n \cdot \ddot{x}_n = -F_{n-1}(x_{n-1}, \dot{x}_{n-1}) - F_{f,n}(t) \end{gathered} \tag{7}$$

The movement equation of the train during braking can be determined by the summation of the force equilibrium equations for each coach in the train body, resulting into:

$$m_1 \cdot \ddot{x}_1 + m_2 \cdot \ddot{x}_2 + m_3 \cdot \ddot{x}_3 + \dots + m_n \cdot \ddot{x}_n = -(F_{f1}(t) + F_{f2}(t) + F_{f3}(t) + \dots F_{fn}(t)) \tag{8}$$

According to Figure 3, the number of relations describing the relative displacement between the coaches is *n-1* for a train made up of *n* vehicles connected via the traction, collision and coupling devices.

Thus, the formulation of the relations that characterize the movement in the traction, collision and coupling devices will be done for every group of two consecutive coaches, connected to each other:

- the first group of traction, collision and coupling devices (coaches 1 and 2)

$$\begin{aligned} \ddot{x}_1 - \ddot{x}_2 = -\frac{1}{m_1 \cdot m_2} \cdot [&(m_1 + m_2) \cdot F_1(\Delta x_1, \Delta \dot{x}_1) - m_1 \cdot F_2(\Delta x_2, \Delta \dot{x}_2) + m_2 \cdot F_{f1}(t) \\ &- m_1 \cdot F_{f2}(t)] \end{aligned} \tag{9}$$

Further on, while noting the relative displacement between the two vehicles with $y_1 = \Delta x_1$ and if replacing in relation (9), it will become:

$$\ddot{y}_1 = -\frac{1}{m_1 \cdot m_2} \cdot [(m_1 + m_2) \cdot F_1(y_1, \dot{y}_1) - m_1 \cdot F_2(y_2, \dot{y}_2) + m_2 \cdot F_{f1}(t) - m_1 \cdot F_{f2}(t)] \tag{10}$$

- the second group of traction, collision and coupling devices (coaches 2 and 3):

$$\begin{aligned} \ddot{x}_2 - \ddot{x}_3 = -\frac{1}{m_2 \cdot m_3} \cdot [&(m_2 + m_3) \cdot F_2(\Delta x_2, \Delta \dot{x}_2) - m_3 \cdot F_1(\Delta x_1, \Delta \dot{x}_1) - m_2 \cdot F_3(\Delta x_3, \Delta \dot{x}_3) \\ &+ m_3 \cdot F_{f2}(t) - m_2 \cdot F_{f3}(t)] \end{aligned} \tag{11}$$

Going on, while noting the relative displacement between the two coaches, with $y_2 = \Delta x_2$ and replacing in relation (11), it will become:

$$\ddot{y}_2 = -\frac{1}{m_2 \cdot m_3} \cdot [(m_2 + m_3) \cdot F_2(y_2, \dot{y}_2) - m_3 \cdot F_1(y_1, \dot{y}_1) - m_2 \cdot F_3(y_3, \dot{y}_3) + m_3 \cdot F_{f2}(t) - m_2 \cdot F_{f3}(t)] \quad (12)$$

- the group i of traction, collision and coupling devices (coaches i and $i+1$):

$$\ddot{y}_i = -\frac{1}{m_i \cdot m_{i+1}} \cdot [(m_i + m_{i+1}) \cdot F_i(y_i, \dot{y}_i) - m_{i+1} \cdot F_{i-1}(y_{i-1}, \dot{y}_{i-1}) - m_i \cdot F_{i+1}(y_{i+1}, \dot{y}_{i+1}) + m_{i+1} \cdot F_{f,i}(t) - m_i \cdot F_{f,i+1}(t)] \quad (13)$$

- the group n-1 (the last group) of traction, collision and coupling devices (coaches n-1 and n):

$$\ddot{y}_{n-1} = -\frac{1}{m_{n-1} \cdot m_n} \cdot [(m_{n-1} + m_n) \cdot F_{n-1}(y_{n-1}, \dot{y}_{n-1}) - m_n \cdot F_{n-2}(y_{n-2}, \dot{y}_{n-2}) + m_n \cdot F_{f,n-1}(t) - m_{n-1} \cdot F_{f,n}(t)] \quad (14)$$

The longitudinal forces in the traction, collision and coupling devices will be calculated based on relation (3).

5. NUMERICAL APPLICATION

The case of a train comprising of an engine and 10 coaches will be considered, where the coaches are subjected to a quick action brake from the maximum velocity of 160 km/h, on a track in alignment and bearing. The train was considered to have an engine weighing 120 t and identical coaches, whose maximum mass (including the load) is 47 t. The main parameters in use are:

- for the collision (buffers), the constant that depends on the elastic elements $k_e = 2{,}8 \cdot 10^6$ N/m and the constant depending on the friction force $k_f = 1.4 \cdot 10^6$ N/m;
- for the traction and coupling devices (the mechanical coupler system and the traction hook) the constant that depends on the elastic elements $k_{ec} = 5.46 \cdot 10^6$ N/m and the constant depending on the friction force $k_{fc} = 2.43 \cdot 10^6$ N/m;
- the value of the scale factor $u = 10^4$ [11].

Upon examining the results derived from the simulation presented in figures 4 and 5, the following conclusions can be made:

- for the time interval 0…. 0.9 s, the first braking phase is characterized by the fact that the braking force starts developing for each coach and only the last in the row senses the decrease of the pressure in the general pipe. This aspect is visible in the diagram of the relative strokes between the consecutive coaches (Figure 4) where these strokes visibly decline due to the differences in the braking force, and the time dynamics is shifted by delay time during the proceeding of every distributor along the train. The emergence of the longitudinal forces in the collision devices between the consecutive coaches in this interval is a successive action;

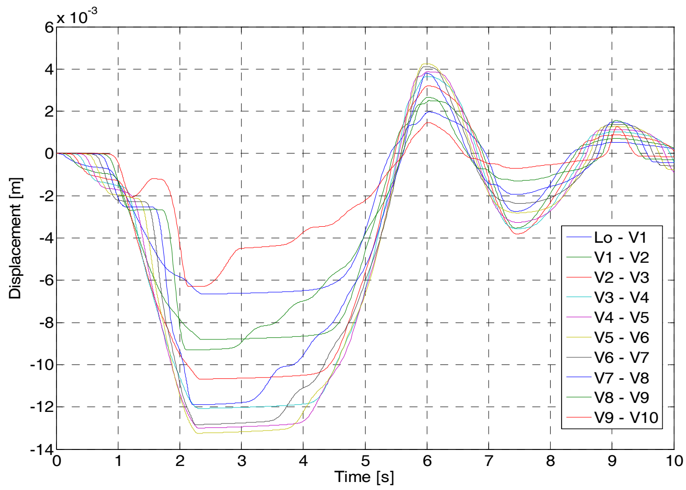

Figure 4. The relative strokes between the coaches

- for the interval 0.9 ...4.5 s, a second braking phase can be identified (that starts at the end of the first phase and continues until the distributor of the first coach reaches the maximum pressure in the brake cylinder [5, 9, 10]). The large differences in the braking forces will result into a maximum compression of every group of collision devices, defined by the lowering of the distance between the coaches and a strong increase of the forces. While the distributor of the first coach tends to reach the maximum pressure, the relative strokes are noticed to remain constant for the first half of the train and lower in the second half. The longitudinal forces come to the maximum values and then drop while the braking forces rise to the maximum value.

- the third phase (interval 4.5...5.4 s) is defined by reaching the maximum values of the braking forces of every coach, with the purpose to release the collision devices and to have the traction and coupling proceed;

- the moment when all the coaches have come to the maximum braking force (specific to the fourth braking phase, in Figure 5 interval 5.4...10 s), characterized by a maximum extension (the relative stroke between the coaches increases) and by the development of the longitudinal forces for the traction and coupling devices. During this time, and due to the elasticity in these devices, oscillating phenomena develop in the train body (release and compression) that are damped in time thanks to the frictions occurring between the component elements of these devices.

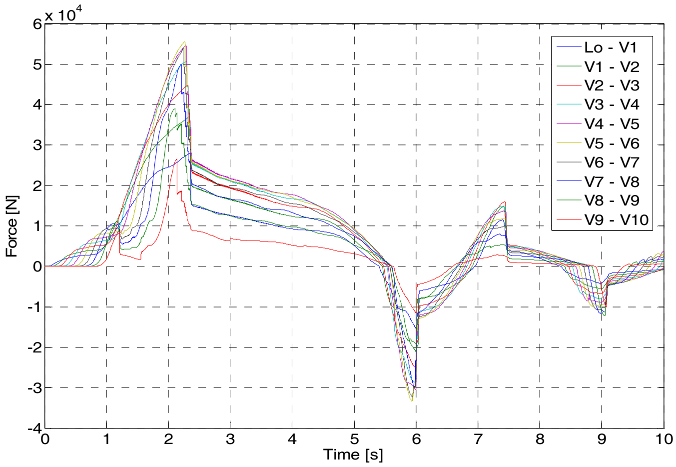

Figure 5. The longitudinal forces between the coaches

From the perspective of the circulation safety, it is necessary to know the maximum values reached by the longitudinal dynamic forces, as well as the distribution of these values along the train body, visible in the bar charts below.

Upon examining the distribution of the forces in the collision devices (see the bar chart in figure 6), the maximum level of the compression forces is noticed to be maximum in the middle of the train, namely in the area of the collision devices 5, 6 and 7 and the maximum value is around 55 kN.

Hence, the conclusion is that the maximum compression forces are located in the middle of the train; for the first and last coaches to be involved in collision), the forces decrease around the value of 26….27 kN, which means a decline by circa 50 % compared to the highest maximum values.

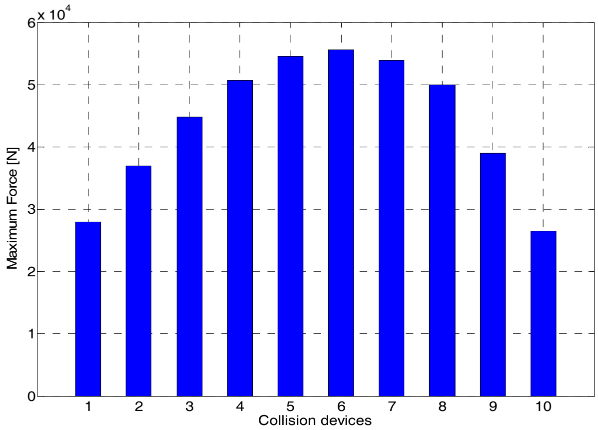

Figure 6. The distribution of the maximum forces of compression in the collision devices

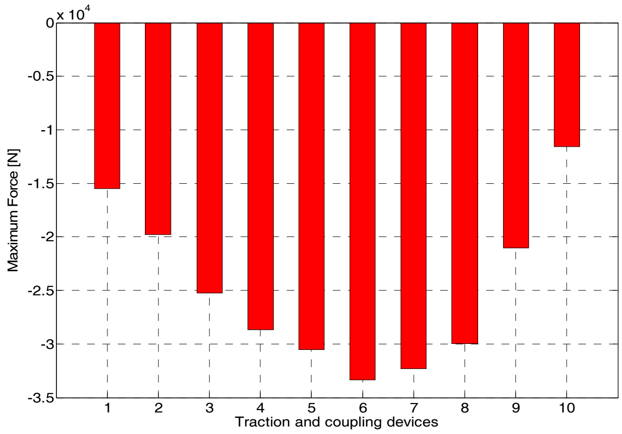

Figure 7. The distribution of the maximum longitudinal extension forces in the traction and coupling devices

For the traction and coupling devices, the maximum extension force occurs at the couplers 6 and 7, at around the values of 33 kN. In figure 7, it can be noticed that the values of the longitudinal forces in the traction and coupling devices are smaller in the first compared to the second half of the train.

6. CONCLUSIONS

This paper focuses on the simulation of the longitudinal dynamic forces occuring in the body of the passenger trains being under an emergency braking, starting from the modelling of the operating features of the traction, collision and coupling devices.

Following the simulation, the below aspects can be mentioned:

- the successive proceeding of the distributors along the train, which involves the succesive development of the braking forces along the train;
- the identification of the phases in the train braking in the diagrams that characterize the time dynamics of the longitudinal forces and also of the relative strokes between the coaches;
- the emergence of the maximum compression of the train during the second braking phase;
- the emergence of the rebound characteristic to the fourth braking phase when all the braking forces have reached the maximum values;

As for the maximum forces reached on the collision devices, it is evident that the maximum compression level is arrived at in the middle of the train, namely in the area of the collision devices 5, 6 and 7, with a value around 55 kN, while for the first and last coaches the values are lower by 50 %.

The extension forces coming from the rebound reach maximum values on the traction and coupling devices in the second half of the train, representing 60 % of the values in the collision ones.

The future reseach studies will have to consider a series of aspects left out in the paper herein, i.e. the drag variations of the coaches.

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