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MODELLING THE CHARACTERISTIC OF THE TRACTION, COLLISION AND COUPLING DEVICE IN RAILWAY VEHILCES

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Abstract: This paper focuses on the modelling of the characteristic in the traction, collision and coupling device for the numerical simulation of the longitudinal dynamic forces in the body of the trains during the braking process. For an easier reference, a model of a three-coach train has been adopted and the disadvantages in the use of the sign function to model the characteristic of the traction, collision and coupling device have been highlighted. An analysis of the results in the numerical simulation of the longitudinal forces has been initiated in case of using the tanh function. The effect of numerical instability will thus be avoided when the basic properties of the longitudinal dynamics of the train during braking are correctly simulated.
Keywords: railway vehicles, buffer, hysteretic model, longitudinal dynamics forces

1. INTRODUCTION

For the traditional braking system, there will always be a time lapse between the reaction of two consecutive coaches, due to the air compressibility and the train length. As a consequence, 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 ones with no braking force developed will hit the braked coaches, thus having the longitudinal shocks emerge. This action could affect the comfort of the passengers, the integrity of the transported merchandise and, sometimes, the circulation safety.

The longitudinal dynamic forces are a function of a range of parameters, such as the running condition of the coaches, the number of coaches in the train and the distribution of their masses, the building and functional parameters of the traction, collision and coupling devices, the length of the general pipe, etc. The large number of these parameters and their variability lades the theoretical evaluation of the forces above [1]. A series of studies on the longitudinal dynamics of the trains under braking action have been nevertheless conducted, with the explicit purpose to find ways of lowering the longitudinal forces present during braking.Theoretical studies on this type of forces developed within the train bodies under braking have been run by Karvatchi [2], Nasr and Mohammadi[1], Zobory [3, 4], Pugi, Fioravanti and Ridi [5...7], Cole[8], Belforte, Cheli [9], Ansari, Esmailzadeh and Younesian [10], Fukazawa [11], El-Sibare [12].

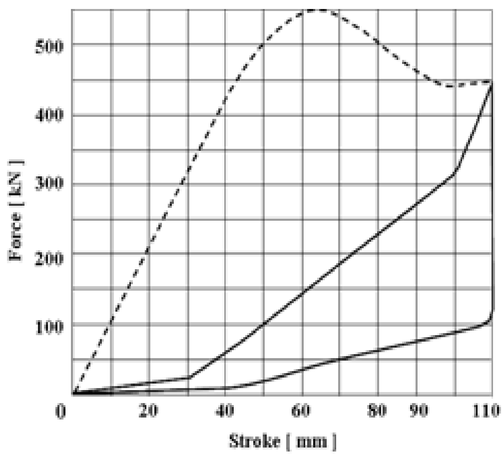

Fig. 1. The characteristics of the Ringfeder-type buffer used on passenger trains, (—, the quasi-static diagram, - - the diagram in a dynamic behavior) [13].

The assessment of the dynamic forces between the consecutive coaches under braking requires the use of a mathematical model pertinent to the traction, collision and coupling devices,whose complexity can directly impact the results of the current study.

The paper recommends a simplified mathematical model to describe the hysteretic feature of the RINDFEDER type traction, collision and coupling device, in order to use simulation programs of the longitudinal dynamic forces in the body of the trains under a braking action.

2. THE MODEL OF THE TRACTION, COLLISION AND COUPLING DEVICE

To have a mathematical modeling of the force 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, (fig.1), and whose elastic and damping elements is of RINGFEDER type (with metallic discs).For the diagram hysteretic shape be maintained, the force model needs to be a function of the elastic characteristic, of the friction forces emerging in the buffer during operation, the maximum stroke and of the speed sign while buffering.

A model to be recommended to assess the force in the collision device is one that uses the *signum* mathematical function, which can indicate the stroke when the buffer operates – namely whether a compression or an expansion stroke.

$$\mathrm{sgn}(x)=\begin{cases}-1 & \text{if} \quad x<0\\ 0 & \text{if} \quad x=0\\ 1 & \text{if} \quad x>0\end{cases} \tag{1}$$

While considering the k_e and k_f constants that depend on the elasticity of the elements within the buffer and on the friction among the metallic discs inside of it, the force in the buffer will have the below value:

$$F_t(x,\dot{x})=\frac{1}{2}\cdot(1-\mathrm{sgn}x)\cdot(k_e-k_f\cdot\mathrm{sgn}\dot{x})\cdot x \tag{2}$$

where x is the displacement and $\dot{x}$ the speed of the collision device.

The fact that the vehicles are fitted with traction and coupling devices during operation that have similar elastic and damping elements with the collision ones, a mathematical model identical with the buffers' has been considered while looking at the constants describing the elastic force k_{ec} and friction force k_{fc}, specific to these devices.

The mathematical model to evaluate the forces in the traction and coupling devices is, therefore:

$$F_c(x,\dot{x})=\frac{1}{2}\cdot(1+\mathrm{sgn}x)\cdot(k_{ec}+k_{fc}\cdot\mathrm{sgn}\dot{x})\cdot x \tag{3}$$

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

The measuring of the longitudinal dynamic forces and reactions between the consecutive coaches in a train – or the assessment of the forces in the collision, traction and coupling devices follows the equation:

$$F(x,\dot{x})=F_t(x,\dot{x})+F_c(x,\dot{x}) \tag{4}$$

To corroborate the relation (4), the Matlab program was used to simulate the functioning of the traction, collision and coupling devices ; the simulation result is featured in figure 2 where for buffer k_e= 2800 kN/m, k_f = 1400 kN/m, and for the traction device, k_{fc} = 2430 kN/m, k_{ec}= 5460 kN/m.

For positive values of the stroke in the collision, traction and coupling devices (x >0), only the collision device will become functional. At x = 0, the force will be null in the collision, traction and coupling, whereas for x <0, the traction and the coupling devices only be operating.

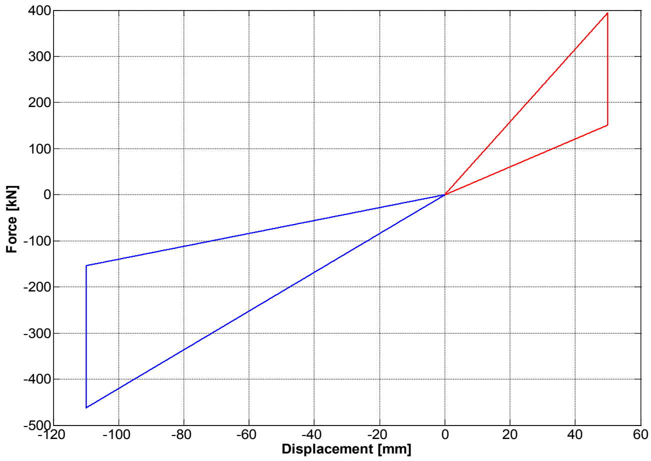

Figure 2. The force-displacement characteristic diagram for the traction, collision and coupling RINGFEDER type device fitting the passenger trains(blue – buffer, red – the traction and coupling device) [14].

3. The simplified train model

The simplified train model with three coaches, connected via the traction, collision and coupling devices, featuring the Ringfeder-type elastic and damping elements is shown in figure 3. The characteristics of such devices have been introduced in the previous section.The forces acting on each coach in the train body are as such:

- inertia forces F_{i1}, F_{i2}, F_{i3};
- braking forces $F_{f1}(t)$, $F_{f2}(t)$, $F_{f3}(t)$;
- forces in the traction, collision and coupling devices $F_1(\Delta x_1,\Delta\dot{x}_1)$, $F_2(\Delta x_2,\Delta\dot{x}_2)$.

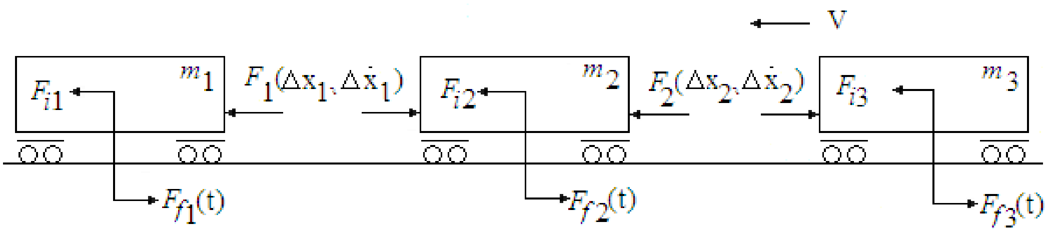

Figure 3. The three-coach train model.

The use of the simplified model of the train requires the introduction of the easier hypotheses below:

≡ all the brakes in the vehicle are active;

≡ the vehicles are equipped with identical distributors, and therefore the time-history of the pressure in the braking cylinder of each coach is the same;

≡ the filling times of the brake cylinders are identical;

≡ the train is considered to be under an emergency braking action.

Further on, the movement equations for the model in fig 3 are as below:

The movement equations for each coach in the train write as such:

- for the coach 1:

$$\begin{aligned} & m_1\cdot\ddot{x}_1+[0.5\cdot(1-\operatorname{sgn}(x_1-x_2))\cdot(k_t\cdot(x_1-x_2)+c_t\cdot|x_1-x_2|\cdot\operatorname{sgn}(\dot{x}_1-\dot{x}_2)) \\ & +0.5\cdot(1+\operatorname{sgn}(x_1-x_2))\cdot(k_c\cdot(x_1-x_2)+c_c\cdot|x_1-x_2|\cdot\operatorname{sgn}(\dot{x}_1-\dot{x}_2)]=-F_{f1}(t) \end{aligned} \tag{5}$$

- for the coach 2:

$$\begin{aligned} & m_2\cdot\ddot{x}_2+[0.5\cdot(1-\operatorname{sgn}(x_2-x_3))\cdot(k_t\cdot(x_2-x_3)+c_t\cdot|x_2-x_3|\cdot\operatorname{sgn}(\dot{x}_2-\dot{x}_3)) \\ & +0.5\cdot(1+\operatorname{sgn}(x_2-x_3))\cdot(k_c\cdot(x_2-x_3)+c_c\cdot|x_2-x_3|\cdot\operatorname{sgn}(\dot{x}_2-\dot{x}_3))] \\ & -[0.5\cdot(1-\operatorname{sgn}(x_1-x_2))\cdot(k_t\cdot(x_1-x_2)+c_t\cdot|x_1-x_2|\cdot\operatorname{sgn}(\dot{x}_1-\dot{x}_2)) \\ & +0.5\cdot(1+\operatorname{sgn}(x_1-x_2))\cdot(k_c\cdot(x_1-x_2)+c_c\cdot|x_1-x_2|\cdot\operatorname{sgn}(\dot{x}_1-\dot{x}_2))]=-F_{f2}(t) \end{aligned} \tag{6}$$

- for the coach 3:

$$\begin{aligned} & m_3\cdot\ddot{x}_3-[0.5\cdot(1-\operatorname{sgn}(x_2-x_3))\cdot(k_t\cdot(x_2-x_3)+c_t\cdot|x_2-x_3|\operatorname{sgn}(\dot{x}_2-\dot{x}_3)) \\ & +0.5\cdot(1+\operatorname{sgn}(x_2-x_3))\cdot(k_t\cdot(x_2-x_3)+c_t\cdot|x_2-x_3|\operatorname{sgn}(\dot{x}_2-\dot{x}_3)))]=-F_{f3}(t) \end{aligned} \tag{7}$$

Upon decoupling the translation movement of the train under braking from the relative movement between the coaches in the train body, there will have the movement equation during the braking behavior:

$$\ddot{x}\sum_{i=1}^{3}m_i=-\sum_{i=1}^{3}F_{f,i}(t) \tag{8}$$

While introducing the following notations,

$$y_1=x_1-x_2 \qquad y_2=x_2-x_3 \tag{9}$$

the relative movement equations between the consecutive coaches in a train can be written as such:

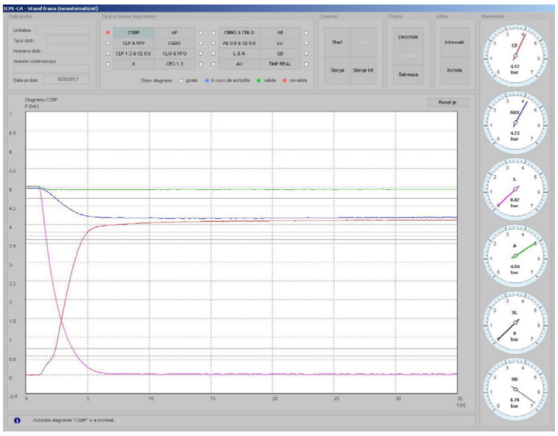

Figure 4. Determiningcharacteristicbrakecylinderfilling on the computerizedstand [14].

$$\begin{aligned} \ddot{y}_1= & \frac{-1}{m_1\cdot m_2}\cdot\{0.5\cdot(m_1+m_2)\cdot[(1-\operatorname{sgn}(y_1))\cdot(k_t\cdot y_1+c_t\cdot|y_1|\operatorname{sgn}(\dot{y}_1)) \\ & +(1+\operatorname{sgn}(y_1))\cdot(k_c\cdot(y_1)+c_c\cdot|y_1|\cdot\operatorname{sgn}(\dot{y}_1))]-0.5\cdot m_1\cdot[(1-\operatorname{sgn}(y_2)) \\ & \cdot(k_t\cdot y_2+c_t\cdot|y_2|\cdot\operatorname{sgn}(\dot{y}_2))+(1+\operatorname{sgn}(y_2))\cdot(k_c\cdot y_2+c_c\cdot|y_2|\cdot\operatorname{sgn}(\dot{y}_2))] \\ & +m_2\cdot F_{f1}(t)-m_1\cdot F_{f2}(t)\} \end{aligned} \tag{10}$$

$$\begin{aligned} \ddot{y}_2= & \frac{-1}{m_2\cdot m_3}\cdot\{0.5\cdot(m_2+m_3)\cdot[(1-\operatorname{sgn}(y_2))\cdot(k_t\cdot y_2+c_t|y_2|\cdot\operatorname{sgn}(\dot{y}_2)) \\ & +(1+\operatorname{sgn}(y_2))\cdot(k_c\cdot y_2+c_c\cdot|y_2|\operatorname{sgn}(\dot{y}_2))]-0.5\cdot m_3\cdot[(1-\operatorname{sgn}(y_1)) \\ & \cdot(k_t\cdot y_1+c_t\cdot|y_1|\cdot\operatorname{sgn}(\dot{y}_1))+(1+\operatorname{sgn}(y_1))\cdot(k_c\cdot y_1+c_c\cdot|y_1|\cdot\operatorname{sgn}(\dot{y}_1))] \\ & +m_3\cdot F_{f2}(t)-m_2\cdot F_{f3}(t)\} \end{aligned} \tag{11}$$

To find out the value of the braking force, the air pressure in the braking cylinder of the vehicle has been experimentally determined on the computerized stand for testing the braking pneumatic equipment on the railway vehicles within the Rolling Stock Department. Figure 4 shows the filling diagram for the braking cylinder for a quick action brake.

With the requirement[13, 14]:

$$F_{f,i}\le F_{a,i}=\mu_a\cdot m_i\cdot g \tag{12}$$

where μ_a is the wheel-rail adhesion coefficient and m_i is the mass of each coach in the train body, the time variation of the braking forces results into:

$$F_{f,i}(t)=\frac{\mu_a \cdot m_i \cdot g}{p_{cf\,max}} \cdot p_{cf,i}(t) \tag{13}$$

where $p_{cf\,max}$ is the maximum pressure developed in the brake cylinders, $p_{cf,i}(t)$ represents the instantaneous pressure emerging in the brake cylinders (from experiments) and g is the gravitational acceleration.

4. NUMERICAL APPLICATION

The case of a short train, comprising of three coaches meant for passenger transport will be considered, 45t in mass, fitted with quick action brakes, which initiates braking at the speed of 160 km/h.

All the coaches are assumed to have the same speed at the initial moment and the action of the brakes on the coaches number two and three is delayed by 0.1 s, due to the length of the general pipe and the mean propagation velocity of the braking wave, regulated at 250 m/s [13, 14, 15].

Some of the representative results are presented in figs. 5 ... 7. Upon examining the results (fig. 5 ... 7) derived from the simulation program, this model used for the traction, collision and coupling can be noticed to introduce integration errors into the calculation of the longitudinal dynamic forces. Nevertheless, the delays between coaches are pointed out, due to the successive proceeding of the brakes in each coach. Similarly, the compression of the collision devices are highlighted during the development of the braking forces and the oscillating phenomena in the train body after the braking forces have become equal, with no indication of the buffers blocking at the change in the stroke.

The errors introduced by the sign function when the relative speed between coaches is zero or close to this value prove that this function is not appropriate to model the characteristic of the traction, collision and coupling devices for the numerical simulation of the longitudinal dynamics of a train under a braking action.

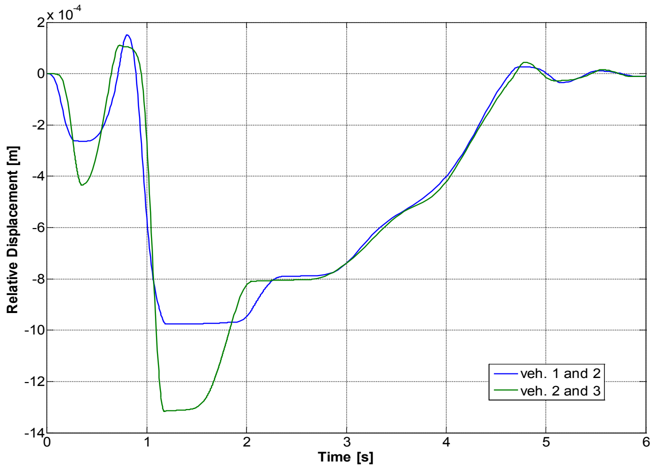

Figure 5. The relative displacement between the train coaches

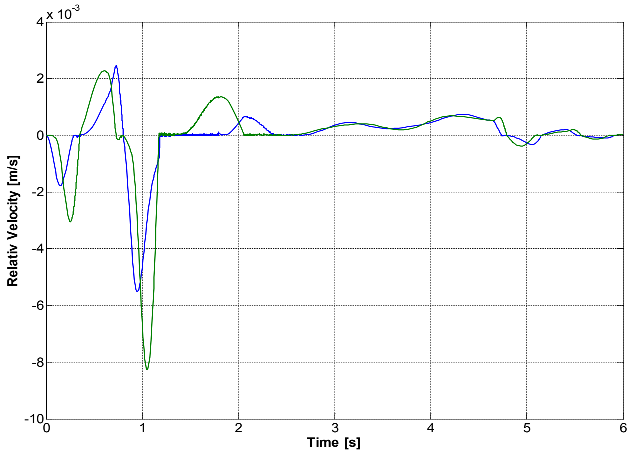

Figure 6. The relative velocity between the train coaches

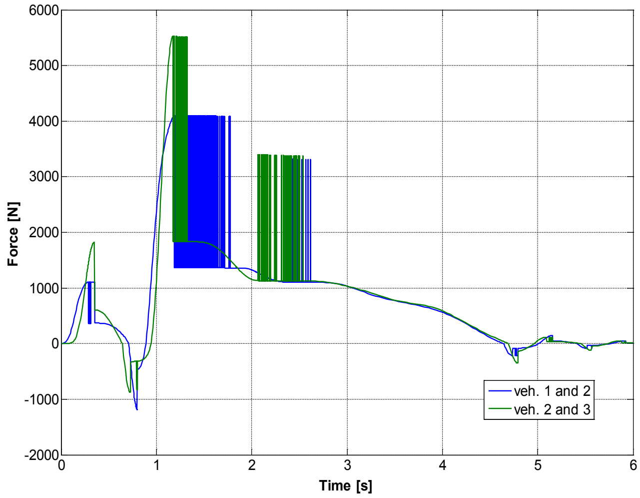

Figure 7. The longitudinal dynamic force between the train coaches

5. THE HYPERBOLIC TANGENT FUNCTION-BASED ALTERNATIVE MODEL

To assess the longitudinal dynamic forces in the train body during braking, the formulas suggested in the previous paragraph need to be continuous and differentiable so as be able to solve the system of differential equations that describe the movement of the coaches during the braking process.

As starting from the definition of the *signum* function, this is noted to be neither continuous nor differentiable (Eq. 1). For avoiding such inconvenience, there is an attempt to approximate the *signum* function via the *hyperbolic tangent* function, as follows:

$$\mathrm{sgn}(x) \cong \tanh(u \cdot x) \tag{14}$$

for $u >> 1$. The parameter u can also be called a scale factor.

Upon introducing the approximation in the relation (14) into the relations (2) and (3) that define the variation of the force in the traction, collision and coupling devices, their equation becomes as below, so as to establish the compression or expansion stroke:

$$F_t(x,\dot{x})=\frac{1}{2}\cdot(1-\text{sgn}x)\cdot(k_e-k_f\cdot\tanh(u\cdot\dot{x}))\cdot x \tag{15}$$

$$F_c(x,\dot{x})=\frac{1}{2}\cdot(1+\text{sgn}x)\cdot(k_{ec}+k_{fc}\cdot\tanh(u\cdot\dot{x}))\cdot x \tag{16}$$

Further on, similar with the paragraph 3, the movement equations for the mechanical model of the train in figure 3 are being written. The equations of the relative displacement between the consecutive strokes of the train are in the following form:

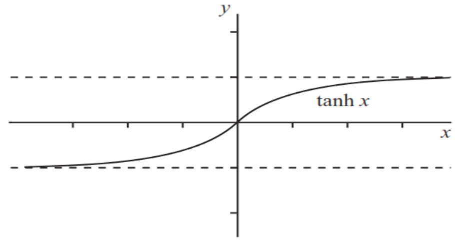

Figure 8. The general graphics of the *tanh* function.

$$\begin{aligned}\ddot{y}_1=\frac{-1}{m_1\cdot m_2}\cdot\{&0.5\cdot(m_1+m_2)\cdot[(1-\text{sgn}(y_1))\cdot(k_t\cdot y_1+c_t\cdot|y_1|\cdot\tanh(u\cdot\dot{y}_1))\\&+(1+\text{sgn}(y_1))\cdot(k_c\cdot(y_1)+c_c\cdot|y_1|\cdot\tanh(u\cdot\dot{y}_1))]-0.5\cdot m_1\cdot[(1-\text{sgn}(y_2))\\&\cdot(k_t\cdot y_2+c_t\cdot|y_2|\cdot\tanh(u\cdot\dot{y}_2))+(1+\text{sgn}(y_2))\cdot(k_c\cdot y_2+c_c\cdot|y_2|\cdot\\&\tanh(u\cdot\dot{y}_2))]+m_2\cdot F_{f1}(t)-m_1\cdot F_{f2}(t)\}\end{aligned} \tag{17}$$

$$\begin{aligned}\ddot{y}_2=\frac{-1}{m_2\cdot m_3}\cdot\{&0.5\cdot(m_2+m_3)\cdot[(1-\text{sgn}(y_2))\cdot(k_t\cdot y_2+c_t|y_2|\cdot\tanh(u\cdot\dot{y}_2))\\&+(1+\text{sgn}(y_2))\cdot(k_c\cdot y_2+c_c\cdot|y_2|\cdot\tanh(u\cdot\dot{y}_2))]-0.5\cdot m_3\cdot[(1-\text{sgn}(y_1))\\&\cdot(k_t\cdot y_1+c_t\cdot|y_1|\cdot\tanh(u\cdot\dot{y}_1))+(1+\text{sgn}(y_1))\cdot(k_c\cdot y_1+c_c\cdot|y_1|\cdot\tanh(u\cdot\dot{y}_1))]\\&+m_3\cdot F_{f2}(t)-m_2\cdot F_{f3}(t)\}\end{aligned} \tag{18}$$

The results of the simulations are displayed in the figs 9... 11.

In order to be able to integrate the movement equations, the new model uses the discontinuous *sign* function only for a change in the operation of the traction devices and coupling with the collision devices; the differentiable *tanh* function is used to change the direction of the force.

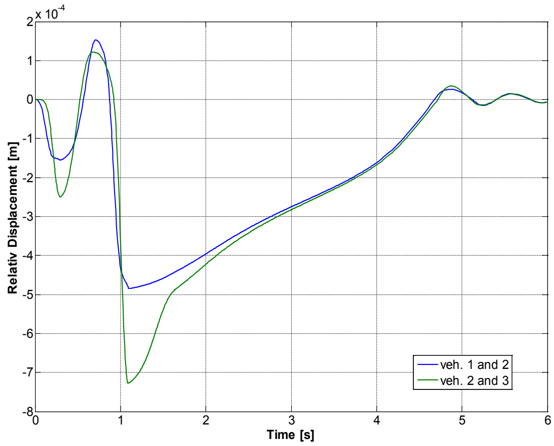

Figure 9. The relative displacement between the train coaches.

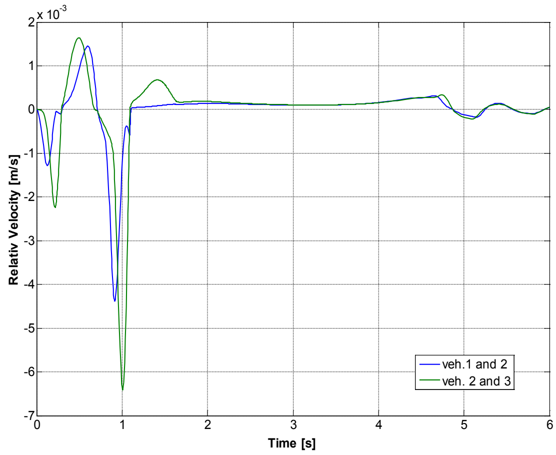

Figure 10. The relative velocity between the train coaches

The diagrams in the figs 9….11 demonstrate that the use of this mathematical model leads to the elimination of the integration errors. Moreover, it accentuates very well the gaps between the longitudinal forces and the shock triggered by the coaches while buffering, due to the delays in the development of the braking force in every coach.

6. CONCLUSIONS

The numerical simulation of the phenomena in the longitudinal dynamics of the trains during braking is linked to the proper modeling of the characteristic of the traction, collision and coupling devices of the coaches.

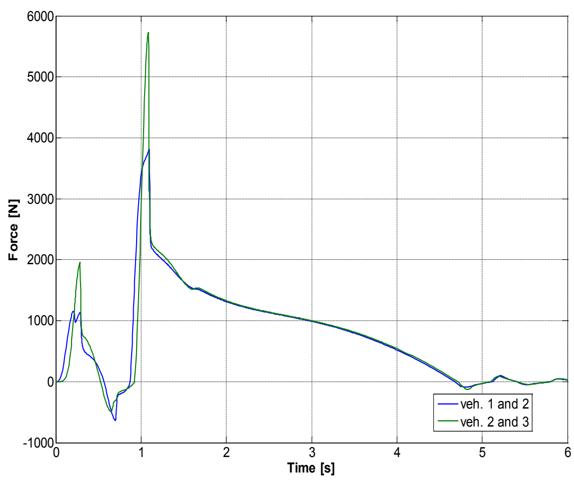

Figure 11. The longitudinal dynamic force between the train coaches

This paper demonstrates that the use of the *sign* function for modeling the direction change of the force in the collision devices is not appropriate because of the numerical instability showing at the calculation of the longitudinal forces between the coaches. This issue can be still solved by using the *tanh* function, without any effect on the accuracy in the simulation of the basic properties of the longitudinal dynamics of the train under a braking action.

The mathematical model, based on the *tanh* function for evaluating the longitudinal dynamic forces in the traction, collision and coupling devices can be used in the programs of braking simulation for the long trains.

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