

## THE USE OF RINGFEDER CHARACTERISTICS FOR EVALUATION OF LONGITUDINAL DYNAMIC FORCES IN TRAIN

Ioan SEBEŞAN<sup>1</sup>, Camil CRĂCIUN<sup>2</sup>, Ana Maria MITU<sup>3</sup>

*In this paper the authors propose the validation of a new mathematical model used for the evaluation of the longitudinal dynamic forces which appear on the traction, buffer and fastening devices during passenger trains braking. The following two cases are considered: a vehicle bound by a fixed system and launched on it at various initial speeds and the model of a train composed of two cars, braking at maximum circulation speed. The results are evaluated through the values of the forces and displacement of the buffers and the traction hook; also the results are correlated to those presented in the scientific literature.*

**Key words:** railway vehicles, buffer, longitudinal dynamics forces.

### 1. Introduction

Train braking is a very complex process, specific to rail vehicles and of great importance by the essential contribution to the safety of the traffic.

This complexity results from the fact that during braking, different kinds of phenomena occur - mechanical, thermal, pneumatic, electrical, etc. The actions of these processes take place in various points of the vehicles and act on different parts of the train, with varying intensities. The major problem is that all must favorably interact for the intended scope, to provide efficient, correct and safe braking actions [1].

In the case of classic UIC brake system, due to the air compressibility and to the length of the train, there will always be a time lapse between the reaction of the leading vehicle and the reaction of the rear one. Corresponding to the propagation rate of air pressure signal, the air distributors will come into action successively and the braking of vehicles begins at different times along the train so that, while some cars are slowing down, others are trying to push, still unbraked, from the rear. That is why, during transitional braking stages,

<sup>1</sup> Prof., Department of Railway Vehicles, University POLITEHNICA of Bucharest, Romania, e-mail: ioan\_sebesan@yahoo.com

<sup>2</sup> Assistant Prof. PhD student, Department of Railway Vehicles, University POLITEHNICA of Bucharest, Romania, e-mail: craciun\_camil@yahoo.com

<sup>3</sup> Senior researcher PhD, mat, Institute of Solid Mechanics of Romanian Academy, Bucharest, Romania, e-mail: anamariamitu@yahoo.com

immediately following the command of pressure variation in the brake air pipe, important longitudinal in-train reactions develop causing stress in the couplers and affecting passenger comfort and, sometimes, even the traffic safety [1].

A classical approach for theoretical studies of the dynamic longitudinal forces developed during the braking actions along trains equipped with automated compressed-air brakes is a mechanical cascade-mass-point model in which vertical and lateral dynamics are usually neglected [1], [2], [3].

Studies regarding the longitudinal dynamics of trains during braking actions are mainly focused on long, heavy freight trains, due to the more obvious effects determined by the length of the brake pipe and numerous large masses interconnected [1, 4, 5, 6, 7, 8].

Comparatively, issues regarding the longitudinal dynamic reactions in passenger train body seem to be less important.

In fact, these are generally short, having a constant and much uniform composition than freight trains and there are sufficient arguments to support these assertions, e.g. passenger railcars are typically two axles bogies vehicles and have almost the same length, the mass difference between an empty and fully charged coach is significantly lower [5, 6, 9].

## 2. Modeling the buffer characteristics

Railway vehicles are linked to each other by different kinds of couplers that must have certain elastic and damping characteristics, according to their remarkable contribution not only for the protection of the vehicle's structure and loading's integrity, but also for the passengers comfort.

Buffers and draw-gears, still widely equipping railway vehicles, are based on metallic elastic rings (RINGFEDER® type) [10], using friction elements to fulfill the required damping effects.

Their characteristics have significant influences on the longitudinal dynamics of the train, with running stability implications.

For coaches buffers with 110 mm stroke are used (prescriptions in UIC leaflets no. 526-1, 2, 3 and 528) [11, 12].

The classical RINGFEDER® type buffer characteristics reveal the action of friction forces between the elastic rings (see fig. 2). Accordingly, the forces within the buffer may be valued assuming an algebraic sum of elastic and friction forces [13].

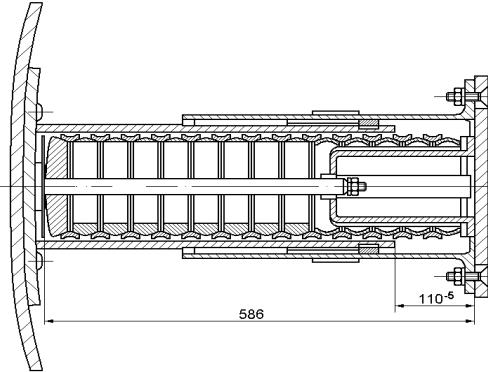


Fig. 1. Buffer used for passenger coach type ICPVA – Romania, according to UIC 528 [11].

The elastic and friction forces depend on the relative displacement between the vehicles, while for the second one, considering a constant friction coefficient according to Coulomb model, the sense of displacement given by the sign of the relative velocity is relevant.

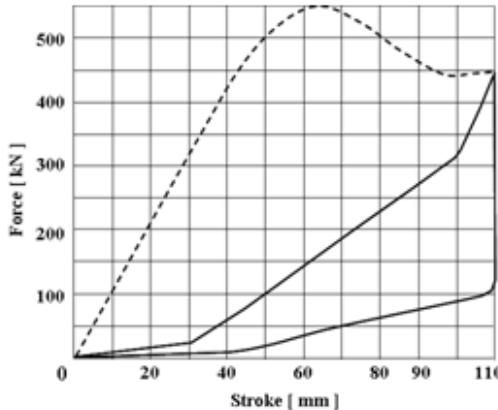


Fig. 2. The experimental quasi- static characteristic of the 110 mm Ringfeder buffer used on passenger coach [9].

Considering  $k_e$  a constant depending on the elasticity of the device and  $k_f$  a constant depending on the inner friction process, the forces within the buffer are:

$$F_b(x, \dot{x}) = \frac{1}{2} \cdot (1 + \text{sgn } x) \cdot (k_e x + k_f |x| \text{sgn } \dot{x}) \quad (1)$$

It is considered that displacement is zero,  $x = 0$  (see fig. 2 and 3) when on the buffer is not acting any force (i.e. elastic and friction forces into buffer modeling equation are zero).

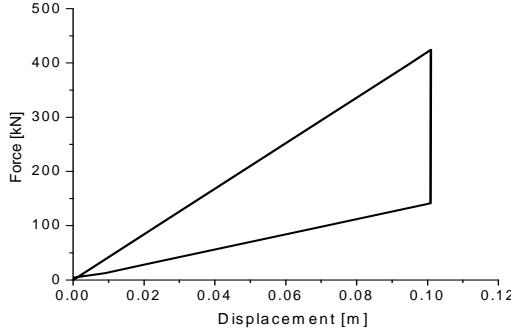


Fig. 3. Force-displacement characteristic of the buffer.

### 3. The analysis of model for buffer and draw-gear

The following situation was considered: a vehicle of mass  $M$ , equipped both with buffers and a traction device fixed by a rigid central coupler to the rigid fixed referential, launched against this referential (see fig. 4) with an initial velocity  $\dot{x}_0$ , adopted with respect to the constructive maximum stroke of the shock devices [13].

For this model, the buffers were remodeled considering a stroke limiter to permit launching simulations with higher initial velocities without exceeding the constructive maximum strike of the shock devices. Consequently, the adopted buffer force model is:

$$F_{b\lim}(x, \dot{x}) = F_b(x, \dot{x}) + F_{\lim}(x, \dot{x}) \text{ [N]} \quad (2)$$

with

$$F_{\lim}(x, \dot{x}) = \frac{1}{2} \cdot k_{\lim} \cdot x^5 \cdot [1 + \text{sgn}(x - x_{\max})] \text{ [N]} \quad (3)$$

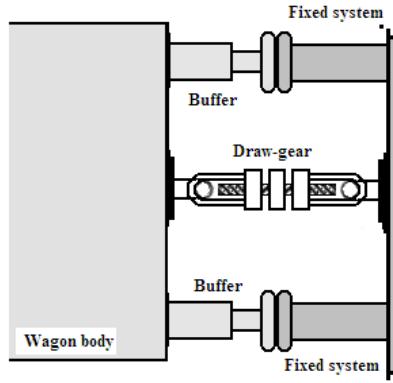


Fig. 4. Schematic wagon connections.

Noting with  $k_{et}$  a constant depending on the elasticity of the traction device and  $k_{ft}$  a constant depending on the inner friction process, the forces within the traction apparatus are:

$$F_t(x, \dot{x}) = \frac{1}{2} \cdot (1 - \operatorname{sgn} x) \cdot (k_{et}x + k_{ft}|x|\operatorname{sgn} \dot{x}) [\text{N}] \quad (4)$$

Considering that the forces developed in the traction device are given by relation (4), the railway vehicle drag is  $R$  (including in that case the braking forces) and  $\ddot{x}$  the acceleration, the equation of motion is:

$$M \cdot \ddot{x} + 2 \cdot F_{b\lim}(x, \dot{x}) + F_t(x, \dot{x}) - R = 0 [\text{N}] \quad (5)$$

In our case, assuming the interaction between a railway vehicle and a rigid fixed system equipped with identical features as the vehicle, the relative displacements and velocities are in fact the displacements  $x$  and the wagon velocities  $\dot{x}$ .

For the performed simulations, the main parameters used for buffers are  $k_e = 2800 \text{ kN/m}$ ,  $k_f = 1400 \text{ kN/m}$ ,  $k_{lim} = 8 \cdot 10^7 \text{ kN/m}$ ,  $R = 45 \text{ kN}$ ,  $M = 45000 \text{ kg}$ .

For draw-gears, in this application, identical characteristics were considered as for buffers:  $k_{et} = 2800 \text{ kN/m}$ ,  $k_{ft} = 1400 \text{ kN/m}$ .

Initial velocities  $\dot{x}$  as required in the application parameter numerical ranges between 0.5 ... 1.4 m/s.

The positive values correspond to compression process, while the negative one to the traction.

Some of the representative results are presented in figs. 5 ... 9.

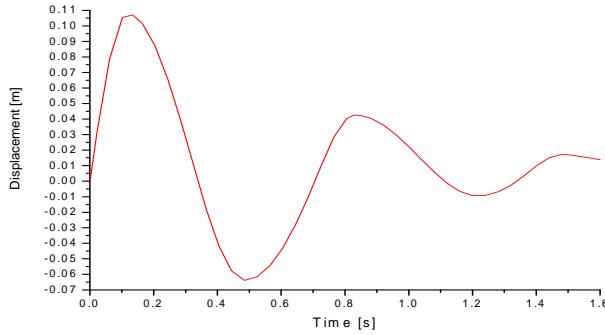


Fig. 5. Vehicle displacement time history for 1.4 m/s initial velocity.

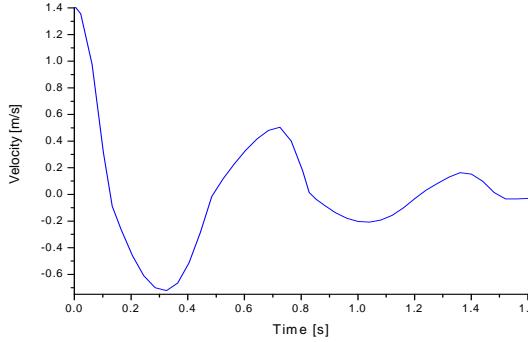


Fig. 6. Vehicle velocity time history for 1.4 m/s initial velocity.

Analyzing the results obtained in fig. 5 and 6, the response of the buffer and draw-gear system on initial imposed conditions is obtained. Also the fact that the movement assured by the model is a cushioned movement and the buffer does not exceed the 110 mm maximum stroke, is pointed out.

For the initial imposed speeds, the successive action of the buffer and draw gear (fig. 7...9) are pointed out. It is observed that as long as the relative displacement between the vehicle and the fixed system is positive, the buffer compresses (for the increase of the displacement) and rebound (for the decrease of the displacement). If  $x \leq 0$  and  $\dot{x} \neq 0$ , the buffer will not function anymore and the draw-gear will come into action.

The initial speeds of the vehicle were increased in order to emphasize the functioning cycles of the buffer and draw-gear, as well as the motion damping. (see fig. 7...9).

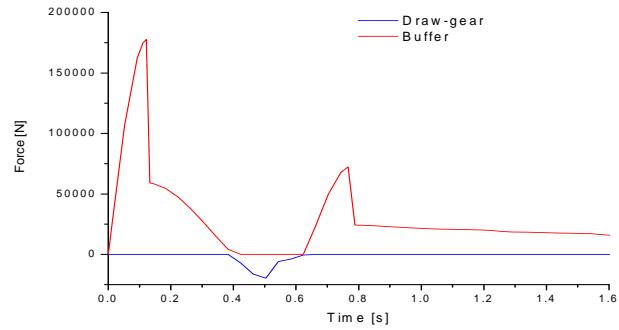


Fig. 7. Buffer and draw-gear force time history for 0.5 m/s initial velocity.

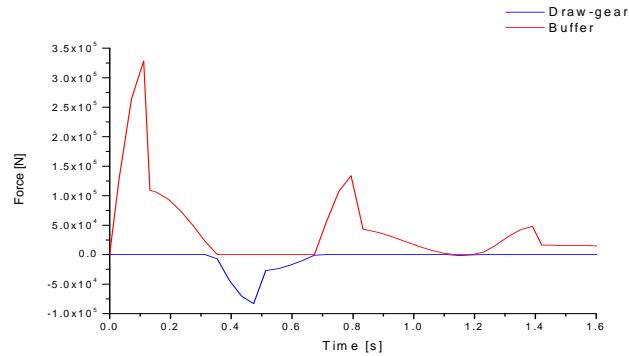


Fig. 8. Buffer and draw-gear force time history for 1 m/s initial velocity.

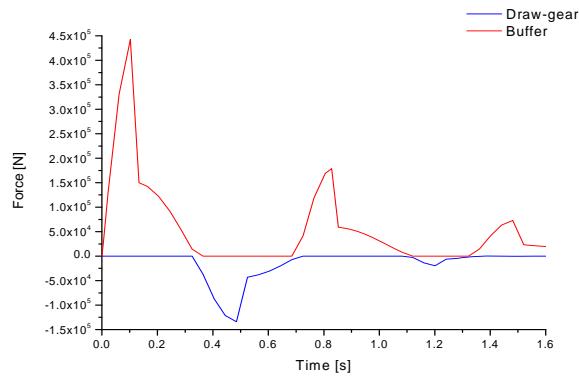


Fig. 9. Buffer and draw-gear force time history for 1.4 m/s initial velocity.

## 5. The analysis of model for buffer and draw-gear during the train braking

Further, the analysis of functioning of the model for the draw-gear, buffer and fastening is performed on a simplified model of train, consisting of two identical vehicles, fastened together and submitted to a braking action from an initial speed (see fig.10).

For this case the following simplifying hypotheses were necessary to be imposed:

- the vehicles are identical from technical and constructive point of view;
- the vehicles have equal masses;
- all brakes are in action;
- the vehicles are equipped with the same type distributors, ergo the pressure in the braking cylinder of each vehicle is identical;
- the filling time of the brake cylinders is identical;
- it is considered that the vehicles are submitted to emergency braking.

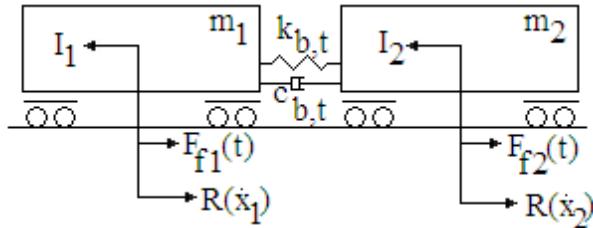


Fig. 10. Simplified model of train.

In order to calculate the braking force it is necessary to know the evolution of air pressure in the brake cylinder of the vehicle. This evolution was determined experimentally on the computerised stand for testing pneumatic braking equipment on rail cars at the Department of Railway Vehicles.

The stand was concieved and realized under the CEEX / AMTRANS / X1C02 research contract, that had Prof.Dr.Ing. Ioan Sebeşan as project director. This presents an inovation on national scale, being fitted with pressure gauge and digital manometers, National Instruments aquisition board and the visualisation and storage of data is acomplished by a computer with a specialized software developed under the same contract (see fig.11).

The maximum braking force developed by each vehicle was considered equal to the wheel-track adherence force at maximum travel speed, according to one odd the basic demands that must be respected during designing braking systems on railway vehicles, that of avoiding wheelset blockage during breaking procedure in normal conditions.

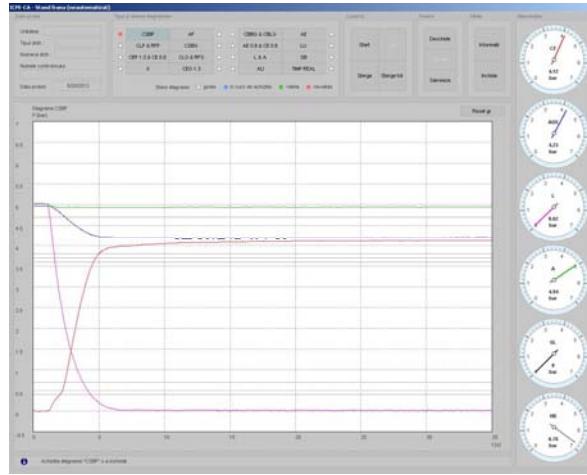


Fig. 11. Determining characteristic brake cylinder filling on the computerized stand.

From a mathematical point of view, that assumes that on the contact surface between the wheel and track, the braking force  $F_{f,i}$  would not exceed the wheel-track adherence forces  $F_{a,i}$  [4]:

$$F_{f,i} \leq F_a = \mu_{a,i} \cdot m_i \cdot g \quad (6)$$

where the wheel-track adherence coefficient was noted as  $\mu_a$ .

Under these conditions, the evolution in time of the braking forces can be calculated, considering the condition (6), [14]:

$$F_{f,i}(t) = \frac{\mu_a \cdot m_i \cdot g}{p_{cf,i} \max} \cdot p_{cf,i}(t) \quad (7)$$

In the previous formula,  $p_{cf,i} \max$  is the maximum pressure developed in the brake cylinders,  $p_{cf,i}(t)$  represents the instantaneous pressure developed within the brake cylinders (determined experimentally) and  $g$  is the gravitational acceleration.

The equations of motion are:

$$\begin{aligned} M_1 \cdot \ddot{x}_1 + 2 \cdot F_{b \lim}(x, \dot{x}) + F_t(x, \dot{x}) &= -F_{f1}(t) - R(\dot{x}_1) \\ M_2 \cdot \ddot{x}_2 + 2 \cdot F_{b \lim}(x, \dot{x}) + F_t(x, \dot{x}) &= -F_{f2}(t) - R(\dot{x}_2) \end{aligned} \quad (8)$$

To determine and evaluate the response of the buffers and draw-gears or in other terms the evaluation of dynamical longitudinal forces in the train, a calculus program was developed in Simulink, for a train formed of two vehicles, under the action of an emergency braking at a maximum speed of 200 km/h. The software takes the evolution of the pressures within the brake cylinder that were determined experimentally, executes the correspondent delay to the development of air pressure within the brake cylinder for the second vehicle due to the length of the general air pipe and the propagation speed of the braking wave, calculates the braking forces and based on the equations of motion (8) determines the evolution between the two vehicles.

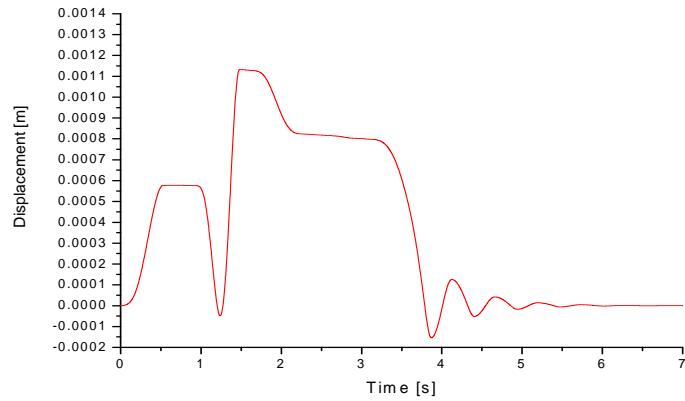


Fig. 12. Relative displacement of the train vehicles.

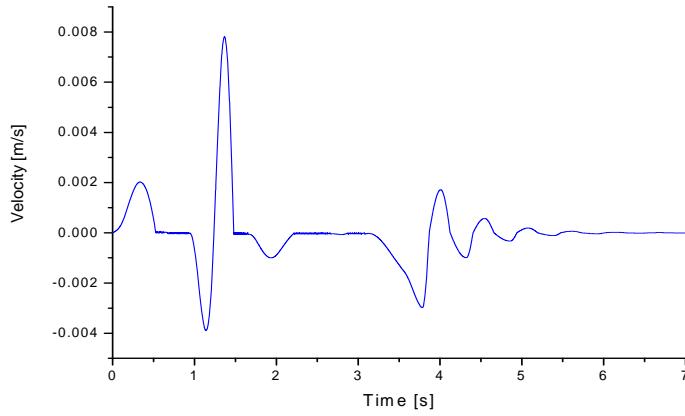


Fig. 13. Relative velocity of the train vehicles.

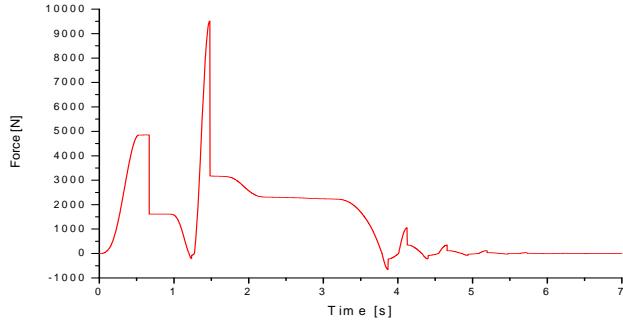


Fig. 14. The evolution of longitudinal forces between vehicles.

Due to the delay of development of the braking force for the second vehicle (due to the construction of the braking installation, the great length of the general pipe, etc. [1, 2, 4, 7]), the second vehicle bumps into the first vehicle (fig. 12...14 in the 0...0,5 s time interval), a rebound appears and during the developing of the braking forces on the second vehicle the compression of the train starts to appear. In the presented diagrams the compression is marked during the 2...3,4 s time interval.

When the braking forces have reached the peak value for both vehicles, the recoil phenomenon takes place in the train, followed by an oscillatory movement marked on the 3,4...6 s time interval. After that period, no longitudinal dynamic phenomena appear in the train any longer, because the forces that act on it have reached an equilibrium, phenomena remains stable if the steady state is no longer retained.

## 6. Conclusions

Taking into account the presented theoretical aspects and the results obtained from the simulation, the following conclusions can be drawn:

- the forces obtained on the buffers during the simulation by imposing initial speeds of 1.4 m/s correspond to those stated in the buffer's functioning characteristics;
- increasing the initial speed over the value of 1.4 m/s (approximately 5 km/h), causes the occurrence of very large forces on the buffer. This aspect does not characterize a normal braking situation of train but rather a situation of collision between the vehicles;
- during braking, due to the connection of the cars composing the train, the relative speeds that appear between two consecutive cars during braking, cannot be higher than the speeds specified earlier;

- the use of the mathematical model for the buffer and draw-gear on a train submitted to an emergency braking on a maximum speed of 200 km/h points out the compression of the train followed by a recoil and an oscillation movement;

- last, we consider that the mathematical model adopted and presented in this study can be utilized to determine the longitudinal dynamic forces that occur during train braking.

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