Aspects Concerning the Longitudinal Dynamics of Passenger Trains During Braking Actions

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Abstract: - During the braking action, there appear dynamic reactions between the component vehicles of the train, acting on the shock and traction apparatus. Under certain conditions, these forces may reach high values, affecting the passengers' comfort and, more important, even the safety of the traffic. Considering the train as mechanical model of an elastic-damped system with several liberty degrees, consisting in individual rigid masses that represent the component railway vehicles and taking into account the constructive and functional particularities of the brake system equipping it, there were determined the longitudinal dynamic reaction developed into the passenger train's body during the braking actions.

Key-Words: - train braking, dynamic longitudinal reactions, mechanical model of the train, safety of traffic

1 Introduction

Trains' braking is an extremely complex process, specific for railway vehicles and having a great importance in contributing to ensure the safety of traffic. In fact, during the braking actions, there develop several different phenomena of mechanical, thermal, pneumatically, etc. nature. All of them act in various places of each vehicle and in the assembly of the train, with various intensities. It is essential that the effects of all these actions favorably interact in order to ensure an efficient, correct and safe braking.

Moreover, the train itself is a mechanical complex assembly of different vehicles. While running, between them appear and develop dynamic reactions determining, in certain limits, diverse mechanical responses influencing not only the traffic safety, but also the passengers' comfort.

After a braking action command, the speed begins to decrease due both to the kinetic energy's dissipation mainly through the heat developed through friction and to the resistance forces that each vehicle and, accordingly, the whole train, are submitted to.

These processes develop with different intensities in various places of the train assembly. So, in the case of a classical passenger train, equipped with standard pneumatic brake system:

- at each vehicle's level, the braking force increases in time, up to the commanded value, according to the filling time which depends on the brake and air distributor constructive characteristics [1];
- if the component vehicles of the train have antiskid equipments, even if all of them are of the same constructive and functional type, their actions would be particularly dependent of the instantaneous conditions of the wheel-rail adhesion and of the vertical dynamic load of each wheelset, determining divers random fluctuations of the instantaneous braking forces;
- along the train, the effective action of the brakes begins successively, according to the length of the train and the propagation speed of the braking wave, etc.

In the case of classical passenger trains, the component vehicles are usually almost identical, except the locomotive, they are equipped with the same braking systems, which are all active. However, there are inevitable differences, including the load, that determine certain differences between the brake and the inertial instantaneous forces that act on each coach.

Due to these and several more other aspects, during the braking action there appear dynamic reactions between the component vehicles of the train, acting on the shock and traction apparatus.

Under certain conditions, these forces may reach high values, affecting the passengers' comfort and, more important, even the safety of the traffic.

That is why it is necessary to establish the values of these reactions, in order to find out the possibilities to decrease their effects.

2 Mechanical model of the train

In order to achieve a theoretical study of the dynamic longitudinal forces developed during the braking actions along a train equipped basically with automate compressed-air brakes according to [1], one may accept a mechanical model of an elastic-damped system with $n-1$ liberty degrees, consisting in n individual rigid masses m_i, connected through elements having well defined elastic c_i and damping ρ_i characteristics (see fig. 1) [2].

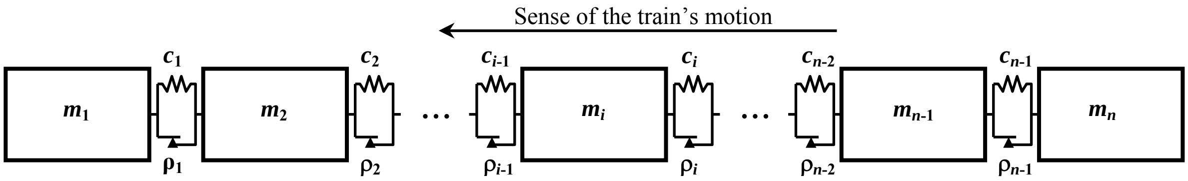

Fig. 1. Mechanical model of the train.

If we consider that x_i are the covered distances and $\ddot{x}_i$ the instantaneous accelerations, on a certain vehicle of the train there will act the following forces (see fig. 2): $F_{f,i}$ the instantaneous braking

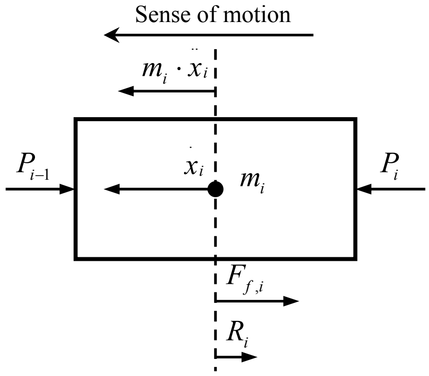

Fig. 2. Forces acting on a vehicle within the train during braking.

force of the vehicle, R_i the vehicle's normal resistance, and P_{i-1}, P_i the forces acting on the shock and traction apparatus between $i-1$ and i, i and $i+1$ respectively masses (vehicles). The latter forces are obviously depending on the elastic and damping characteristics of each above-mentioned apparatus.

Consequently, on a certain i vehicle of the train that is submitted to the influence of the exterior forces, the equation of motion is [3]:

$$m_i \cdot \ddot{x}_i = -F_{f,i} - R_i - P_{i-1} + P_i \quad , \tag{1}$$

considering $i = 1, 2, \ldots, n$ and $P_o = P_n = 0$.

Applied to all component vehicles of the train, relation (1) constitutes a differential nonlinear equation system of second degree.

For each m_i vehicles' mass, the covered distance and instantaneous speed and acceleration, as well as the instantaneous braking, normal resistance and longitudinal developed forces in the train's body are mathematically time dependent.

In accordance with the initial applied conditions, the equation system (1) can be solved applying a numerical integration process. Generally, the main parameters influencing the assembly of the studied problem are:

- the train's composition, the number, mass and type of the vehicles, as well as their disposal in the body of the train;
- the braking system functional characteristics;
- the elastic and damping characteristics of the shock and traction apparatus;
- the evolution of the friction coefficient between the brake shoes and wheels, brake pads and discs respectively, eventually between the electromagnetic track brakes and rail, in accordance with the equipments of the train's vehicles.

3 Analysis of the main parameters

The instantaneous The instantaneous braking force of a vehicle equipped with classically standard pneumatic disc brake with individual brake rigging and brake cylinders having automat brake rigging regulator incorporated into the piston's rod can be calculated as follows:

$$F_{f,i} = \left(F_{p,i} - R_{reg}\right) \cdot C_d \cdot \mu_g \quad , \tag{2}$$

in which $F_{p,i} = \pi \cdot d_{cf}^2 \cdot p_{cf,i} / 4 - F_R$ is the force obtained at the brake cylinder's piston rod, depending on the cylinder's diameter d_{cf}, the relative air pressure within the cylinder $p_{cf,I}$ and the elastic force F_R developed by the brake cylinder's relapse spring. In the same equation, R_{reg} represents the force opposed by the automat brake rigging regulator and $C_d = i_t \cdot n_{cf} \cdot 2 \cdot r_m / D_o \cdot \eta_{tim}$ is a constant where i_t is the amplification ratio of the brake rigging, n_{cf} the number of active brake cylinders of the vehicle, r_m the medium friction radius of the brake discs, D_o the diameter of the wheels measured in the nominal rolling circle's plane and η_{tim} is the mechanical efficaciousness of the brake rigging.

In relation (2), μ_g is the friction coefficient between the disc and the brake pads, which according to [4] is considered to have a constant value during the braking action.

The relative instantaneous pressure within the brake cylinder depends on several parameters and some of the most important are:

- the pressure's evolution during the filling time of the air brake distributor;
- the precise moment of reaching the maximum pressure and its value within the brake cylinder, which from that moment, all along the braking action duration, can be considered constant, except the case if antiskid equipments action occurs;
- the characteristics of the first time duration of braking, defined as the time period of rapid increasing of the brake cylinder pressure, up to approximately 10% of the maximum admitted value.

It is considered that only at the end of the first time duration of braking begins to develop an effective brake force for the vehicle [1].

While taking into account the vehicle's relative position within the train body, in order to establish the correct instantaneous values of the air pressure in the brake cylinders, it is essential to establish the precise moment of the beginning the air brake distributor's action. That one depends on the distance toward the mechanic's tap that commanded the braking action, the braking wave's propagation velocity and the air brake distributors' sensitivities.

For the studied problem, also very important are the characteristics of the shock and traction apparatus. These have a remarkable influence for the protection of the vehicle's structure and the loading's integrity, moreover to ensure the passenger's comfort. In addition, the shock apparatus is very important for the longitudinal dynamics of the train, with running stability implications. Generally, the classical passenger coaches are equipped with lateral buffers having a 110 mm stroke [5].

Both the traction apparatus and the buffers have elastic and energy absorption systems.

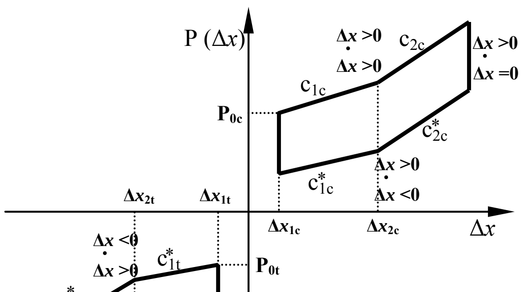

Fig. 3. General elastic characteristics of shock (index **c) and traction (**index **t) apparatus.**

As a rule, the general characteristics of their elastic devices mainly depend on the stroke Δx representing in fact the relative displacement between two successive vehicles, the energy absorption capacity having an important role, too (see fig. 3).

Consequently, from an elastic point of view, the main parameters are: the stiffnesses $c_{i,j}$, the precompression forces $P_{oc,t}$, the length of the stroke defining the inflexion of the elastic characteristics $\Delta x_{2c,t}$ and the precompression of the elastic elements $\Delta x_{1c,t}$ of the shock and the traction apparatus.

The elastic characteristics $c^*_{i,j}$ might be determined either by experiment, or taking into account the damping ratio defined by a β_o percent according to international regulations.

Regarding the elastic characteristics of the passenger wagons' lateral buffers, in the case of RINGFEDER type ones, the static and dynamic characteristic diagrams are presented in fig. 4 [6].

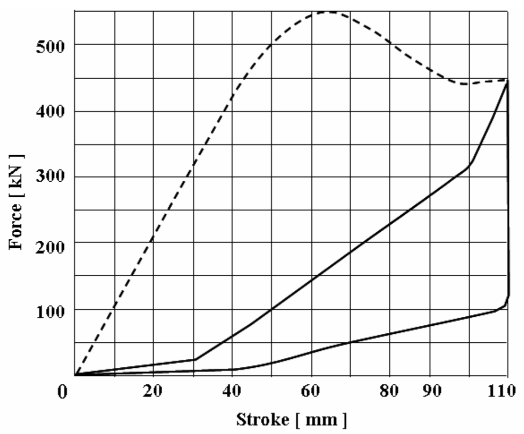

Fig. 4. Theoretical characteristic diagrams of the lateral buffers for passenger vehicle

(——— static; - - - - dynamic **).**

The train's, respectively vehicle's normal resistances are due to all forces opposing their displacement. These forces depend on several parameters, mostly the type and the constructive shape of the vehicles, the running speed, the characteristics and the longitudinal, respectively vertical profile of the track, etc.

In order to simplify the trains' traction calculations, usually the specific main normal resistances r [N/kN] are used. These are defined as the report between the main normal resistances R [N] and the vehicle or train's weight:

$$r = \frac{R}{m \cdot g} \quad , \tag{3}$$

where m [t] is the mass.

As the aim of this study is to determine the relative forces between the train's vehicles, which means that within the calculus it will interest the instantaneous difference in normal resistances between the successive vehicles.

Because the instantaneous speed differences between the component vehicles (considered almost identical) of a train in braking action are quite insignificant, the influence of the main normal resistances is, in this case, insignificant. For this reason, in a first iteration approximation, these forces were not taken into account.

4 Equations of motion of the system

Taking into account the mechanical model of the train presented into §2 and the above mentioned explanations, the equation system (1) becomes:

$$m_1 \cdot \ddot{x}_1 = -F_{f1}(t) + c(\Delta x_1)\cdot(x_1 - x_2)$$

$$m_2 \cdot \ddot{x}_2 = -F_{f2}(t) - c(\Delta x_1)\cdot x_1 + \left[c(\Delta x_1) + c(\Delta x_2)\right]\cdot x_2 - c(\Delta x_2)\cdot x_3$$

$$\cdots\cdots\cdots$$

$$m_i \cdot \ddot{x}_i = -F_{fi}(t) - c(\Delta x_{i-1})\cdot x_{i-1} + \left[c(\Delta x_{i-1}) + c(\Delta x_i)\right]\cdot x_i - c(\Delta x_i)\cdot x_{i+1} \tag{4}$$

$$\cdots\cdots\cdots$$

$$m_{n-1} \cdot \ddot{x}_{n-1} = -F_{fn-1}(t) - c(\Delta x_{n-2})\cdot x_{n-2} + \left[c(\Delta x_{n-2}) + c(\Delta x_{n-1})\right]\cdot x_{n-1} - c(\Delta x_{n-1})\cdot x_n$$

$$m_n \cdot \ddot{x}_n = -F_{fn}(t) - c(\Delta x_{n-1})\cdot(x_{n-1} - x_n) \quad .$$

The instantaneous braking force $F_{f,i}(t)$ of vehicle i depends on the brake type, the time elapsed since the braking command, the moment of brake cylinder's air admission and the filling characteristics.

The calculus stiffnesses c_i depend on the sense and the greatness of the relative displacements $\Delta x_i = x_{i-1} - x_i$ meaning in fact the deformation of the shock and traction apparatus. So, if the deformation $\Delta x_i > 0$, that means that forces act on the buffers, and if the deformation $\Delta x_i < 0$, the forces act on the traction apparatus.

5 Results

We conceived a computer program, based on the above presented theoretical consideration, in order to establish and analyze the effects of the dynamic reactions along the train's body during the braking actions. In particular, the program is designed for the cases of a 10 vehicles passenger train equipped with automate rapid compressed air brake system. We considered each vehicle equipped with two brake discs fixed on each wheelset, according to international regulations for ensuring the traffic safety to run with over 160 km/h [7]. We also considered that the train is submitted to an emergency brake action from the running speed of 250 km/h. We studied several possible situations and some of them are presented below [8].

We studied, e.g., the case of two train composed of 10 identical vehicles, each vehicle having the same mass ($m_i = 30$ t, respectively $m_i = 50$ t) and length (26 m), equipped with identical braking system previously presented. There are four axle vehicles with wheels having a diameter $D_o = 920$ mm. The trains have a general air brake pipe with an interior diameter of 25 mm [1]. The brake discs have an exterior diameter of 610 mm [4]. There are individual brake riggings for each of them, ensuring an amplification ratio $i_t = 2.1$. The interior brake cylinders diameter is $d_{cf} = 245$ mm and the elastic force developed by the brake cylinder's relapse spring is $F_R = 1$ kN. Each brake rigging has an automate regulator incorporated into the piston's rod developing a $R_{reg} = 1.2$ kN force. We adopted a mechanical efficaciousness of the brake rigging $\eta_{tim} = 0.9$.

The study covers the first three braking phases, until all the braking forces along the train become equal (fourth phase), when the longitudinal forces converge to a value corresponding to a slight compression of the train, depending on the damping ratio of the shock and traction apparatus.

Considering the brake cylinders' filling identical and corresponding to the standard filling diagram admitted by regulations, we obtained the longitudinal forces developed along the train as presented in fig. 5 and 6.

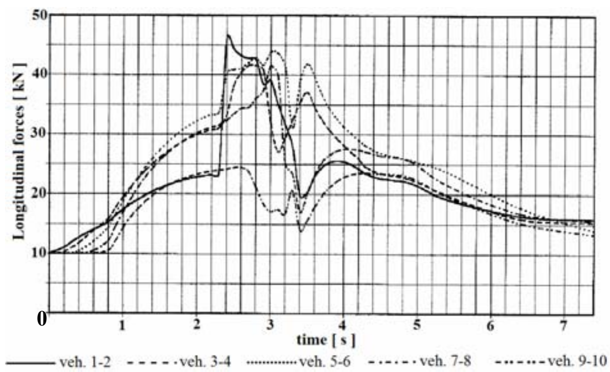

Fig. 5. Evolution of longitudinal dynamic forces within the train body during emergency braking action (10 identical vehicles of 30 t mass each**)**.

It is obvious that in both studied cases, the train is compressed during the whole braking action. At the end of the first braking phase, the higher value of the compression force develops between the 5th and the 6th wagon, and the increase of vehicles' mass of 66% determines its increase with about 14%.

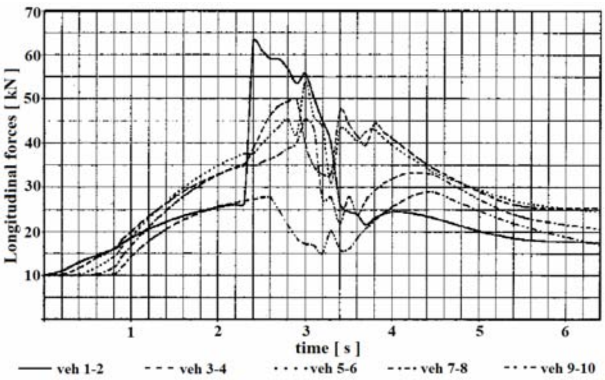

Fig. 6. Evolution of longitudinal dynamic forces within the train body during emergency braking action (10 identical vehicles of 50 t mass each**).**

During the second braking phase, an oscillatory movement develops within the train's body, attempting successive maximum of the longitudinal forces. These maximum values developed between the vehicles of the train during the braking action are presented in fig. 7.

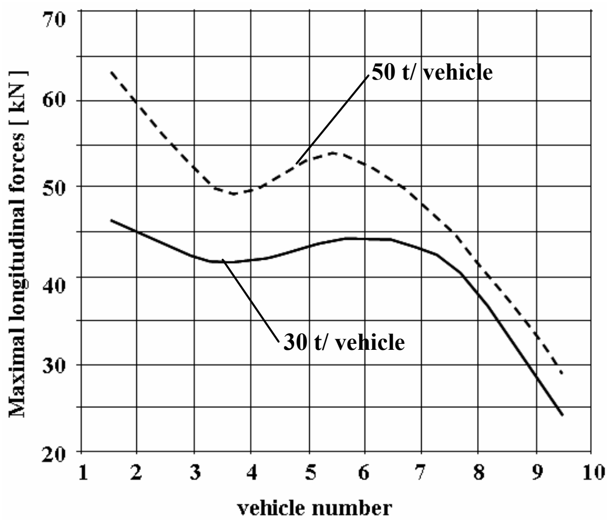

Fig. 7. Assesment of maximal longitudinal forces along the 10 identical vehicles train during an emergency braking action from 250 km/h.

One may also observe that the maximum values repartition's appearance is likewise, the compression forces being higher in the first part and towards the middle of the train.

That means that the vehicles' mass has no significant influence on the repartition of the compression forces. The biggest value of the longitudinal forces occurs between the first and the second vehicles of the train, an increase of 66% of the vehicles' mass determining an increase of about 35% of the maximum values of the dynamic longitudinal forces within the body of the train.

We may also observe a tendency of an oscillating movement during the third braking phase occurring with the equalization of braking forces along the train, but not too important due to the decrease of longitudinal reactions and due to the damping.

We also studied the case of a classical train composed of a six axles locomotive having a mass of 120 t, equipped with disc brakes, tracting 9 wagons that have the same characteristics mentioned above. We considered relevant this case for the study because a locomotive's mass might be 2…4 times bigger than the usual one for a passenger vehicle, constituting a concentrated mass in the front of the train. This fact substantially modifies the mass distribution along the body of the train. Actually, the results presented in fig. 8 illustrate this aspect.

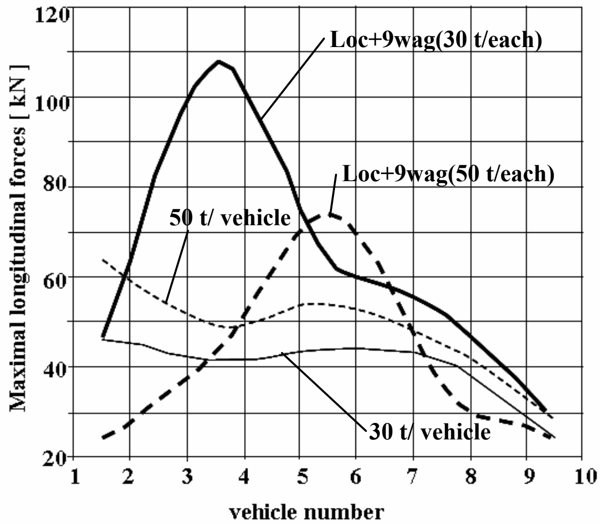

Fig. 8. Assesment of maximal longitudinal forces along the 10 vehicles train during an emergency braking action from 250 km/h
(thick lines – locomotive+9 identical passenger wagons, thin lines – 10 identical vehicles)**.**

At least theoretically, within such a train type one modifies substantially both the appearance and the range of the longitudinal forces developed within the train's body during the braking action.

So, in the case of 9 wagons with an individual mass of 30 t (representing actually 25% of the locomotive's mass), the maximum of forces developed during an emergency braking action from 250 km/h occurs between the 3rd and the 4th vehicles of the train (meaning the 2nd and the 3rd passenger vehicle, taking into account also the locomotive) and there are about 130% bigger than in the case of uniform composition of the 10 identical vehicles of 30 t mass each.

Also, in case of the train composed of locomotive and 9 passenger coaches of each mass 50 t (representing actually 41.5% of the locomotive's mass), the maximum forces developed during the same emergency braking action occurs between the 5th and the 6th vehicles of the train and there are about 20% larger than in the case of uniform composition of the 10 identical vehicles of 50 t mass each.

6 Conclusion

If the vehicles of the train are equipped with a classical, automate compressed air brake system, due to the length of the general air brake pipe, to the fact that the pneumatic command is usually operated from one of its extremities by the engineer and taking into account the air compressibility, during the braking action the commanded forces increase successively along the train's body, beginning normally from the first vehicle.

That is the main reason why during the braking actions there appear dynamic reactions between the component vehicles of the train. These forces may attempt important values depending also on important mechanical parameters including the train's composition, the mechanics of the brake riggings, the friction coefficient between the brake shoes and wheels or between the brake pads and discs, etc.

Considering the train as mechanical model of an elastic-damped system with several liberty degrees, consisting in individual rigid masses that represent the component railway vehicles and taking into account the constructive and functional particularities of the brake system equipping it, there were determined the longitudinal dynamic reaction developed into the train's body during the braking actions.

In the case of a passenger train consisting in 10 vehicles, the theoretical results underline that the evolving of the longitudinal dynamic forces into the train's body due to the emergency braking actions mostly depend on the greatness and the arrangement of masses in the train.

Bigger the difference of vehicles' masses is, the maximums of the longitudinal dynamic reactions increase and, especially because the oscillation movements that develop mainly during the second braking phase, beside the passengers' comfort, the safety of traffic can be affected due to the efforts acting on the traction apparatus that may even result in a breaking of coupling of the train.

In particular, our results point that, especially in the case of high-speed trains, also from the braking point of view, it is more disadvantageous the classical solution of locomotive and passenger wagons than a uniform composition (concerning masses) of the train. It is to prefer as possible equal vehicles' masses, meaning more traction units vehicles in the body of the train, capable to transport also passengers. Nevertheless, it is also obvious that shorter trains are preferred.

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