Computer Aided Study Regarding the Influence of Filling Characteristics on the Longitudinal Reactions within the Body of a Braked Train

CĂTĂLIN CRUCEANU, RĂZVAN OPREA, MARIUS SPIROIU, CAMIL CRĂCIUN, SORIN ARSENE
Faculty of Transports, Rolling Stock Engineering Department
POLITEHNICA University of Bucharest
313 Splaiul Independentei, sect. 6, 77206, Bucharest
ROMANIA
c_cruceanu@yahoo.com, http://www.pub.ro

Abstract: This paper describes a computer aided approach to the study of the longitudinal dynamics of a braked train. The numerical simulation of the phenomena takes into account the delay between the braking action of the cars and the fact that the values of the braking force of each wagon is reached gradually, in certain time after the braking command. The values of the time interval elapsed between the braking command and the moment when the nominal braking force is reached are given by international railway regulations. The time variation of the braking force in this interval is determined both with the aid of a computerized experimental stand and through mathematical functions and its influence is analyzed in the present study.

Keywords: Safety of traffic, Numerical simulation, Train Brake, Longitudinal dynamic forces, Braking delay.

1 Introduction
Train braking is a specific, extremely complex process and of critical importance to ensure the safety of running in train operation. During the braking actions, several mechanical, thermal, pneumatically etc. nature phenomena develop, all of them acting in various places of each vehicle and in the assembly of the train, with different intensities. It is essential that the effects of all these actions favorably interact in order to ensure an efficient, correct and safe braking. Even the train itself is a complex mechanical assembly of different vehicles. According to the international regulations, the main brake system for railway vehicles is the indirect compressed air brake [1]. Due to its specific working principles and corresponding to the complex action of phenomena during the braking actions, within the train’s body there develop certain dynamic reactions, acting on the shock and traction apparatus. While running, under various specific conditions, these forces may reach high values, affecting the passengers’ comfort and, more important, even the ride safety.

So, it is useful to develop suitable models for studying the evolution of the dynamic phenomena developed within the passenger train’s body, taking into account the various parameters that may affect the proportion and the layout of these reactions.

For one of the important parameters involved, the filling time of the brake cylinders, it was developed a computer aided approach to the study of the longitudinal dynamic reactions in the body of a train submitted to braking actions. The numerical simulation of braking forces characteristics is based both on experimental and mathematical established functions for the evolution in time of the air pressure within the brake cylinders of each vehicle.

2 Theoretical bases
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 the mechanical model of an elastic-damped system consisting in \( n \) individual rigid masses \( m_i \), connected through elements having well defined elastic \( c_i \) and damping \( \rho_i \) characteristics (see fig. 1).

A certain \( i \) vehicle of the train is mainly submitted to the following exterior forces: \( F_{f,i} \), the instantaneous braking force of the vehicle, \( R \), the vehicle’s normal resistance, and \( P_{i-1}, P_i \), the cumulated elastic and damping forces acting on the shock and traction apparatus between \( i-1 \) and \( i \), \( i \) and \( i+1 \) respectively.
vehicles. Considering \( x_i \) and \( \dot{x}_i \) the positions and the instantaneous accelerations of the \( i \)th vehicle, the motion equation is [2]:

\[
m_i \cdot \ddot{x}_i = -F_{f,i} - R_i - P_{i-1} + P_i.
\]  

(1)

Because the relative instantaneous velocity differences between the vehicles of the train determine insignificant differences of normal resistances between the cars - especially if the vehicles are considered identical - the \( R_i \) forces were neglected.

The motion equations, considering the simplifying hypothesis above mentioned, are given by:

\[
M \cdot \ddot{X} = -F - P + P_i,
\]  

(2)

where \( M \) is the mass matrix, \( X \) is the state vector, \( F \) is the vector of the braking force, \( P_i \) is the vector of the front shock and traction apparatus forces and \( P_r \) is the vector of the rear forces.

For establishing the \( P \) forces, there were considered non-linear characteristics of the shock and traction apparatus and the commonly used Coulomb friction model [3]. The specific friction forces in the shock and traction apparatus depend on the relative displacement \( x \) and velocity \( v = \dot{x} \):

\[
P = \frac{1}{2} \left[ (c_m - \Delta c_i \tan h(k_i \tanh(v)))x(1 - \psi) - (c_m + \Delta c_i \tan h(k_i \tanh(v)))x(1 + \psi) \right].
\]  

(3)

The index \( c \) is for the buffers, while \( t \) is for the traction apparatus and \( \psi \) is a ponder function. It was assumed that a negative value for \( x \) means that the apparatus is compressed and a negative value of \( v \) means that the cars tend to come closer.

The numerical model of the braked train was previously developed and tested by the authors. All the necessary information may be found in [4].

3 Braking forces evolution

For railway vehicles running over 160 km/h, the basic standard pneumatic disc brake system is compulsory [1].

Fig. 1. Mechanical model of the train.

For such braking system, the instantaneous braking force of a certain \( i \) vehicle is:

\[
F_{f,i}(t) = K \cdot p_{cf,i}(t).
\]  

(4)

In the previous relation \( K \) is a constant depending mainly on specific parameters such as the vehicle’s brake system constructive and functional characteristics (the amplification ratio of the brake rigging, the geometrical dimensions of the brake discs, the nominal diameter of the wheels, the mechanical efficiency of the brake rigging, the friction coefficient between the disc and the brake pads etc.), and \( p_{cf,i}(t) \) is the instantaneous relative air pressure within the brake cylinders [5].

Taking into account that the pressure evolution in the brake cylinder must respect the specific requirements according to [1], it is obvious that the braking characteristic \( F_{f,i}(t) \) is directly depending on the instantaneous relative air pressure within the brake cylinders \( p_{cf,i}(t) \).

It was considered the maximum braking force developed by each vehicle equal to the wheel-rail adhesion forces at the maximum running speed, in respect with the fundamental requirement in designing the basic braking systems of railway vehicles, in order to avoid, in normal conditions, the wheel-rail adhesion forces.

Mathematically, this means that at the wheel-rail contact level, the braking forces \( F_{f,i} \) must not exceed the wheel-rail adhesion forces \( F_a \):

\[
F_{f,i} \leq F_a = \mu_u \cdot m_i \cdot g.
\]  

(5)

\( \mu_u \) denoting the wheel-rail adhesion coefficient.

Under these conditions, the evolution in time of the braking forces may be calculated, taking into account the condition (5) and according to (4):

\[
F_{f,i}(t) = \frac{\mu_u \cdot m_i \cdot g}{P_{cf,\text{max}}} \cdot p_{cf,i}(t).
\]  

(6)

In the previous relation \( P_{cf,\text{max}} \) denotes the maximal pressure developed within the brake cylinders and \( g \) the gravitational acceleration.

It is important to notice that an exploitable braking force develops only after reaching an approx. 0.4 bar pressure within the brake cylinder. Once the pressure gets its maximum, it remains constant.
during the process and, consequently, the same is for
the braking forces of the considered vehicle.

4  Air pressure evolution within the brake cylinders
The dynamic longitudinal reactions developed
during the braking actions along trains are
essentially influenced by the air pressure evolution
within the brake cylinders. With the aim of obtaining
accurate results, the conceived soft offers the
possibility of using pressure information either
directly from a complex computerized system for
testing the air brake valve distributors for railway
vehicles (see fig. 2), or mathematical approximation
functions.

4.1 Data obtained through a complex computerized testing system for brake valve distributors of railway vehicles
In order to ensure a high accuracy related to the real
air pressure evolution in the brake cylinder, the soft
was adapted for direct use of the acquisitioned data
through a complex computerized testing system.
That one is a very performant automate system for
acquisition, analyzing, data processing and diagnose
the compressed air brake valve distributors of
railway vehicles, developed by a high qualified team
of Romanian academics and researchers from
POLITEHNICA University of Bucharest as director,
The Institute of Solids Mechanics of Romanian
Academy, The National Institute of Research-
Development for Electrical Engineering “ICPE-
CA”, within a scientific research project that began
in 2005. The prototype is in final testing in S.C.
Atelierele CFR Grivita S.A., a company with a long
history in passenger coach repairs, modernisation
and rebuilding.

Due to the high accuracy of this testing system, it is
also suitable for scientific research regarding the
pneumatics and mechanics of the air brake
equipments of railway vehicles. The computerized
stand is able to record the pressure variations in the
braking cylinder down to hundredths of a second.
This resolution is absolutely satisfactory as the
numerical integration time step proves to be about
the same size. The solution of the numerical model
was carried out both using an approximating
polynomial for the pressure variation and using
interpolated data. The results did not show any
significant differences.

The diagram of the evolution of air pressure within
the brake cylinder for KE valve distributor generally

---

**Fig. 2. Schematic of the complex computerized system for air brake distributors testing:**
1 – pressure regulator; 2, HB, L, R – air tanks; 3 – air brake distributor; PCP – pneumatic command panel;
5 – brake cylinder; 6A, 6C, 6R, 6L, 6SL, 6 HB – digital display pressure transducers; E10...E13 – electro
faucets with calibrated outlets; Er1...Er9, Ev1...Ev3 – electro faucets; DAB – data acquisition board;
CALC – computer; IMP – printer; r1 – general mechanical admittance tap; F – air filter.
used on the Romanian railway vehicles is presented in fig. 3.

4.2 Approximation functions
With a view to simulate diverse filling characteristics, using the acquisitioned data, it was first determined an interpolation polynomial function that approximates accurate enough the air pressure evolution within the brake cylinder for the interest domain of $\Delta t = 2.86$ s (see fig. 3), as specified in § 3:

$$p_{cf, 3.16}(t) \approx -0.033 \cdot t^5 + 0.31 \cdot t^3 - 1.081 \cdot t^4 + 1.53 \cdot t^3 - 0.413 \cdot t^2 + 0.88 \cdot t + 0.4 \text{ [bar]}.$$ (7)

This function was extrapolated for the case of a 4 s filling time:

$$p_{cf, 4}(t) \approx -0.005 \cdot t^5 + 0.063 \cdot t^3 - 0.3 \cdot t^4 + 0.59 \cdot t^3 - 0.22 \cdot t^2 + 0.643 \cdot t + 0.4 \text{ [bar]}$$ (8)

and also for a 5 s filling time:

$$p_{cf, 5}(t) \approx -0.001 \cdot t^5 + 0.016 \cdot t^3 - 0.103 \cdot t^4 + 0.26 \cdot t^3 - 0.127 \cdot t^2 + 0.49 \cdot t + 0.4 \text{ [bar]}.$$ (9)

These functions are presented in the diagram presented in fig. 4.

5 Application
As an application, it is presented the case of a six identical vehicles train submitted to an emergency braking action started at a running speed of 180 km/h.

Each 40 t mass vehicle is equipped with rapid indirect compressed air brake system with brake discs in respect with the regulations. The air brake valve distributors are considered to have the filling characteristics as presented in § 4, respecting all the imposed conditions [1].

Moreover, there were also considered the following simplifying hypothesis:
- the initial compressions of the elastic elements of the shock and traction apparatus were neglected;
- a tight coupling within the component vehicles, as regulated for passenger trains;
- an average steady braking wave propagation speed along the train of 250 m/s, the minimum imposed regulated value [1].

Numerical simulations of the system with the different forces models were carried out in Matlab with the solver ode45, until numerical stability and reasonable results were reached. The relative tolerance of the solver proved to be the most important parameter for the numerical model involved and it was finally set to $10^{-9}$.

There were considered the following situations:
- all the vehicles equipped with air brake valve distributors that ensure a 3.16 s filling time;
- all the vehicles equipped with air brake valve distributors that ensure a 4 s filling time;
- all the vehicles equipped with air brake valve distributors that ensure a 5 s filling time;
- vehicles equipped with air brake valve distributors that ensure a diverse filling times spread in different
combinations within the train. Some of the most relevant dynamic longitudinal forces evolutions are presented in fig. 5…8.

6 Analyze of results
The results first of all emphasizes that different brake cylinders filling times conduct to major modifications of compression and traction dynamic forces developed longitudinally in the body of a train submitted to an emergency braking. These variations occur both as forces magnitudes and position among the vehicles along the train. So, for the cases of same filling times for each vehicle of the train, the maximum longitudinal dynamic forces diminish their magnitude in average about 24…43% while increasing the filling time from 3.16 s to 4, respectively 5 s. However, the disposition of the maximum forces remains almost the same along the train (see fig. 9 and 10).

![Fig. 6. Filling time of 5 s for all vehicles.](image)

![Fig. 7. Filling times in order of vehicles in train: 3.16 – 3.16 – 3.16 – 3.16 – 5 – 3.16 s.](image)

![Fig. 8. Filling time in order of vehicles in train: 5 – 5 – 5 – 5 – 3.16 s.](image)

The situation changes if the vehicles of the same train are characterized by different filling times. In that case, the magnitude of the dynamic longitudinal forces and their layout along the train strongly depends on the position and filling time of each vehicle within the train. Regarding the maximum longitudinal dynamic forces between the vehicles, some illustrative results

![Fig. 9. Maximum compression forces and their distribution for different filling times.](image)

![Fig. 10. Maximum traction forces and their distribution for different filling times.](image)
are synthesized in fig. 11 and 12 (the 3.16 s filling time was denoted with “3” and the 5 s one with “5”). It is to notice that longer filling times, wherever placed in the braked train, diminish the traction longitudinal dynamic forces. The same observation is also correct regarding the compression forces, but only if the vehicles having longer brake cylinders filling times are placed in the first half of the train. Otherwise, the longitudinal compression dynamic forces increase, the maximum values being obtained between the vehicles situated in the second part of the train.

Anyway, at least for the studied case, in spite of the almost spectacular forces evolutions due to different filling times, their magnitude can not affect severely the shock, traction and coupling apparatuses.

7 Conclusions
The filling characteristics of the brake cylinders, even respecting the admitted tolerances, have an important influence on the dynamic longitudinal forces developed in the train body submitted to braking actions. The effect is not connected only to the magnitude of the maximum forces, but also influences their distribution among the vehicles along the train.

Regarding the modeling and numerical integration techniques we notice the following aspects:
- both the experimental data and an polynomial approximation may be used to model the pressure variations. If the resolution of the data is high enough, or if a reasonably accurate polynomial is used, the results will be practically the same;
- the interpolation of the experimental data may be linear, when the pressure values are sampled to hundredths of a second. In this case, other types of interpolation prove to be more time consuming and do not bring any improvement in the results;
- the most important parameter of the numerical solver was the relative tolerance. Smaller values than $10^{-9}$ did not bring significant changes in the solution.

Acknowledgement

The present paper was made within the framework of the PN II “IDEl” Program, research contract ID_1711/2009, financed by the Romanian Government through UEFISCSU.

References: