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ABOUT ADOPTION OF THE VERTICAL SUSPENSION PARAMETERS FOR PASSENGER WAGONS

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Abstract: *The suspension's projection implies to establish the elastic and shock absorbent constants of the conditions to ensure the rolling vehicle's dynamic performances. The necessary vertical rigidities´ adoption implies also a certain correlation between the constructive masses. The geometrical and mechanical characteristics of the railway affect significantly the suspension's response. Moreover, during the braking actions, due to the friction force between discs and brake pads, the axles suspension´s shock absorption factor increases.*

Key words: *suspension, suspension's rigidity, shock absorption factor, braking*

INTRODUCTION

An important role of suspension is that of contributing to the decreasing of mutual forces between the vehicle and the track, with the view to maintain these forces into the limits that are established by the conditions of security of traffic and the necessity of ensuring the protection both of the running gear and the track. That is why the correct adoption of the suspension parameters plays a major part for safety operating and when it comes about passenger coaches, there must be considerate the necessity of ensuring a vibratory comfort as better as possible.

In point of suspension, a railway vehicle can be modeled through a complex, oscillating system, formed by continuously distributed and concentrated masses joined together through elastic and shock absorber elements. Such model implies however a big number of unknown elements, situation which makes it more difficult the resolving of both the motion equations and extracting the conclusions that are useful in designing. That is why, considering the peculiarities of railway vehicles vibrations and the present engineering of the bogies, there is a model of a oscillating two-liberty degrees system which reflects, in an acceptable extent, the behavior of the studied system and which can be adopted with a view to vertical suspension study. (see fig. 1) [1,2].

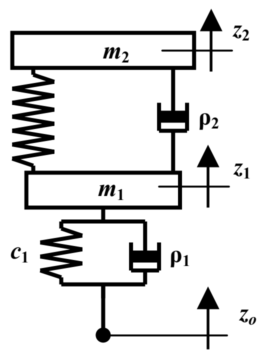

Fig. 1. Elementary linear model of a vertical, two-liberty degree oscillating

Generally, the suspension's design implies the establishment of elastic and shock absorber constants in conditions concerning the insurance of rolling vehicle's dynamic performances. The necessary rigidities can theoretically be determined by the condition of imposed natural frequencies, but by considering also the necessary limitation of oscillation amplitudes, which the vehicle box can develop on vertical direction, and by taking notice of the elasticity repartition between the two suspension levels.

Since the suspension's state of excitation is given by the answer of unsprung masses, the value of these masses and the quality of the track defined by its irregularities, as well as its elastic and shock absorbent characteristics, constitute a very important aspect of projection that must be taking into account. One must also consider the fact that, while braking actions, due to the friction force that appears between the discs and brake pads, the shock absorption factor of the axles' suspension increases.

CORRELATIONS BETWEEN MASSES

In the case of vertical suspension with two liberty levels, starting from the natural pulsation equation:

$$m_1 m_2 \omega^4 - (m_2 c_1 + m_1 c_2 + m_2 c_2)\omega^2 + c_1 c_2 = 0 \quad (1)$$

where m_1 represents the bogie's sprung mass, $m_2 = (m_c + m_i)/2$, m_c represents the box vehicle's mass, m_i the charge mass, and c_1 and c_2 the equivalent rigidities of the axles, respectively central bogie's suspension levels, it follows the adequate expression of the deflection [1,2]:

$$f_{1,2} = \frac{g}{8\pi^2} \cdot \frac{\nu_1^2 + \nu_2^2 \mp \Delta}{\nu_1^2 \nu_2^2} \quad (2)$$

$$\Delta = \sqrt{(\nu_1^2 + \nu_2^2)^2 - 4(1 + m_2/m_1)\nu_1^2 \nu_2^2}$$

In these relations, ν_1 and ν_2 represents the natural low, respective high frequency of suspension, which, for avoiding the resonance phenomena and for comfort conditions there are recommended to be valorized, in the case of high speed vehicles, at 1 Hz, respectively between 5…7 Hz; g is gravitational acceleration's value.

Considering these, the suspension's total static deflection can be figured out with the following relation:

$$f = f_1 + f_2 \approx \frac{g}{4\pi^2 \nu_1^2} \approx \frac{1}{4\nu_1^2} \quad (3)$$

By establishing the rigidities of the suspension's levels, there are also other aspects that must be considered, namely the condition that the total deflection Δh caused by the charge not to exceed the accepted difference of elevation Δh_{max} between the buffers of two successive vehicles:

$$\Delta h = \frac{m_i}{4(m_1 + m_2)\nu_1^2} \le \Delta h_{\max} \quad (4)$$

and the known technical recommendation to ensure a $(85...75)\%/(15...25)\%$ ratio between the primary suspension's rigidity and the rigidity of the secondary one [2].

On the basis of determined points, in the case of railway coaches having a bogie's sprung mass $m_1 = 1...3$ t, a box mass $m_2 = 20...40$ t and a charge mass $m_i = 5...12$ t, we determinate the coherence range of the necessary equivalent rigidities of the two levels, enforcing the low natural frequency $\nu_1 = 1$ Hz and the high one according to $\nu_2 = 5...7$ Hz.

From the analysis and the mathematical processing of the obtained results it has been determined that the two levels rigidities of suspension can indeed keep all the conditions previously imposed if between the vehicle's masses, expressed in tons, the following condition is respected:

$$m_1 = 55(m_2 + m_i) + 10 \quad (5)$$

THE VERTICAL SUSPENSION'S EXCITATION

The excitation of vertical suspension depends on the track's elastic system constants, on its irregularities and on the vehicle's unsuspended mass. This study limits itself at the vertical motion, supposing that the longitude leveling irregularities of the two tracks are in the same phase and, for simplifying, that the oscillations are harmonious. The vehicle—railway oscillatory system was studied based on a simplified one liberty level mechanical model. (see fig. 2) [1].

If m_n is considered to be the unsprung mass, c_c and ρ_c rigidity, respective track's shock absorption factors, z_o and η_o the amplitude of the unsuspended mass answer, respectively of the vertical irregularities, λ the pulsation disaccord, the system's, answer factor in amplitude is, in simplifying hypotheses:

$$H_{\eta z} = \frac{z_o}{\eta_o} = \sqrt{\frac{1 + 4D_c^2\lambda^2}{\left(1 - \lambda^2\right)^2 + 4D_c^2\lambda^2}} \quad , \qquad (6)$$

$$z_o = \eta_o H_{\eta z} \, . \qquad (7)$$

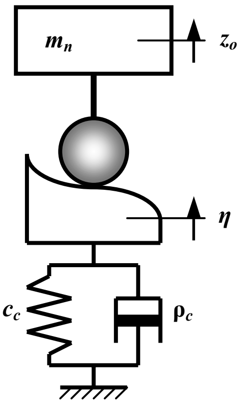

Fig. 2. Mechanical model for establishing the vehicle's suspension excitation.

The influence that the main parameters have on the transfer function on amplitude of the vertical displacements at the level of mounted axles, which determines the amplitude of suspension's received excitation, was the central problem. Thus, it has been considering the track irregularities wave length $L = 0.001...30$ m, unssprung masses $m_n = 1000...5000$ kg (corresponding, in an extreme case, to the blocked braking axle), various rigidities and shock absorption factors of the track.

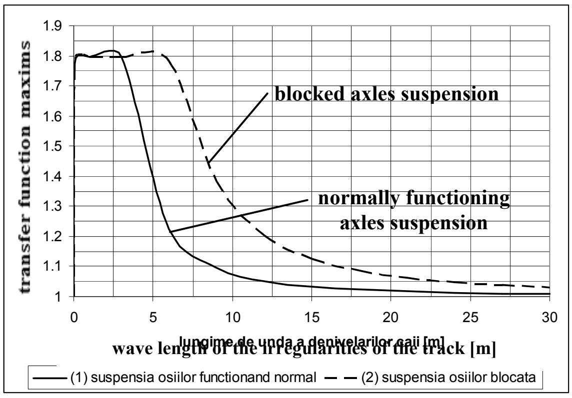

Fig. 3. Dependence of transfer function's maxims in amplitude of vertical oscillation on the track irregularities' wave length.

The main results point that, in the considered domains, the transfer function on amplitude of the vertical displacements of the unsuspended masses reaches a maximum which recedes with the increment of the irregularities track's length wave, becoming smaller than 1.1 for length waves $L \geq 7.5$ m (see fig. 3, curve 1). These maximums move themselves with the increasing of the unsuspended mass, due e.g. to the fitting-out the mounted axles with brake discs. For the extreme case of a blocked axles suspension, the transfer function $H_{\eta z}$ decreases under 1.1 for wavelength $L \geq 17$ m (see fig. 3, curve 2).

It is important to know these sizes both for the excitation's value transmitted to vehicle's suspension and for the fact that it determines the dynamic load's size at the wheel - rail contact, which influences at the same time the adhesion force that can affect the braking and the traction ones, the latter in the case of driving vehicles.

ABOUT THE RIGIDITY OF THE SUSPENSION'S LEVELS

The natural frequencies of the two suspension's levels depend on the concerned mass proportion: the bogie's suspended mass, the box mass and the charge mass, theoretically, the passengers and luggage's mass. It is obvious the fact that, observing the conditions pertinent to impose natural frequencies can be achieved only through determinate areas of specified mass's values. Thereby, in the particular case of a railway carriage with a 25 tons mass, for the 1000…3000 kg bogie's sprung masses, the possibilities of ensuring 1…1.2 Hz natural low frequencies, respective of a 5…7 Hz high frequencies is presented in fig. 4 and 5. E.g., for a 1000 kg bogie's suspended mass, the high frequency exceeds 7 Hz value that is why it does not appear in fig. 5.

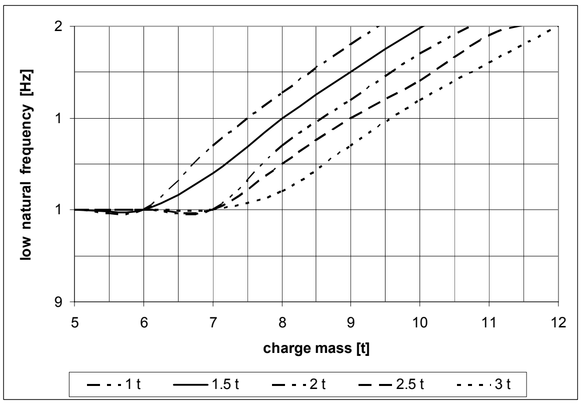

Fig. 4. Dependence of low natural frequency on charge mass and sprung bogie mass (vehicle's box mass 25 t)**.**

The imposed natural frequencies determine, in the whole suspension system, certain values of the two levels' rigidities. The high frequency increase realizes itself through the increment of the rigidity of axles' suspension, simultaneous with the decrease of central suspension's rigidity, as one can see in fig. 6 (for the prior particular case above mentioned).

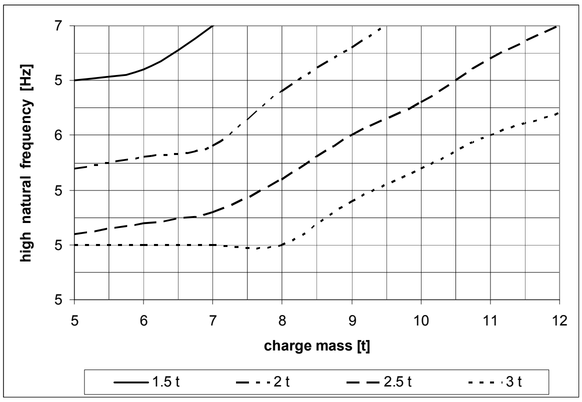

Fig. 5. Dependence of high natural frequency on charge mass and sprung bogie mass (vehicle's box mass 25 t)**.**

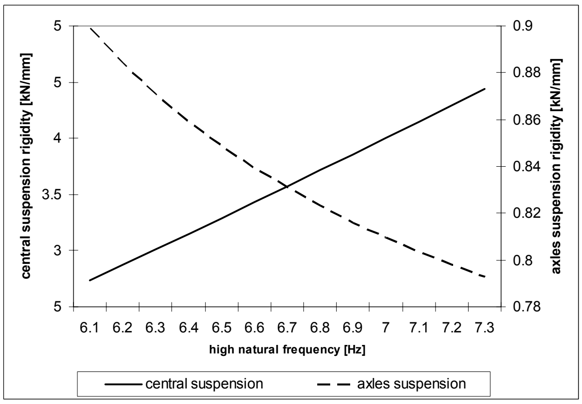

Fig. 6. Dependence of equivalent rigidities on high natural frequency.

It might be interesting to analyze the influence that the axles suspension rigidity may have upon the phenomena during the braking actions. In the particular case of disc brake use, which is a compulsory element for the vehicles running with over 160 km/h [3], the braking pads, which are fixed through hangers on the bogie's frame, will have vertical direction displacements accountable to the discs that are usually mounted on the axle.

Therewith, the bogie's suspended mass will acquire vertical acceleration, determining a vertical forces development. During the braking action, if the sum of friction forces between the braking discs and the pads exceeds the value of these vertical forces, the suspension's primary level can obstruct itself, situation leading on to a substantial bigger unsprung mass [4].

This determines the development of certain vertical dynamic forces at the rail-wheel contact. These forces increase while the running speed increases and might become dangerous for the ssafety of running.

From this point of view, as a principle, we can say that the recession of the axles' suspension level obstruction assumes the reach of some vertical dynamic forces big enough at the bogie's frame level.

Taking into account the two levels mechanical suspension model (see fig. 1), the vertical dynamic forces at the bogie's frame level can be determined with the relation [1,2]:

$$u_1 = z_o \cdot m_1 \cdot \omega^2 \cdot \frac{1}{N} \cdot \sqrt{A^2 + B^2} \quad , \qquad (8)$$

where: z_o represents the unsuspended mass's amplitude of oscillation which can be calculated with relation (7), $A = \gamma \cdot \lambda^2 (1 + \mu) - \mu \cdot \lambda^4$, $B = \lambda^3 (1 + \mu) \cdot \delta_2$, δ_1 and δ_2 the dumping ratio of the axles, respectively central suspension levels, $N = \sqrt{a^2 + b^2}$, $a = (\Gamma - \lambda^2 \cdot \delta_1 \cdot \delta_2), \gamma = \frac{c_2}{c_1}$, $b = [\Lambda \cdot \lambda \cdot \delta_2 + (\gamma - \lambda^2) \cdot \lambda \cdot \delta_1], \lambda = \frac{\omega}{\omega_o}, \mu = \frac{m_1}{m_2}$, $\omega_o = \sqrt{\frac{c_1}{m_2}}$, $\Gamma = (1 - \mu \cdot \lambda^2) \cdot (\gamma - \lambda^2) - \gamma \cdot \lambda^2$, $\Lambda = 1 - \mu \cdot \lambda^2 - \lambda^2$, $\omega = 2 \cdot \pi \cdot \frac{V}{3,6 \cdot L}$, V [km/h] the instantaneous circulation speed, L [m] the rails' irregularities wave length.

On the calculus programmers ground, we observed that once with the increase of the axles suspension's rigidity, there is also recorded an increase of the bogie's vertical dynamic forces. The percentage growth of the force maxims, accountable to the minimum accepted value of 3 kN/mm is roughly linear, reaching almost 15% at the maximum accepted value of 3.9 kN/mm, how is presented in the diagram in fig. 7 [4].

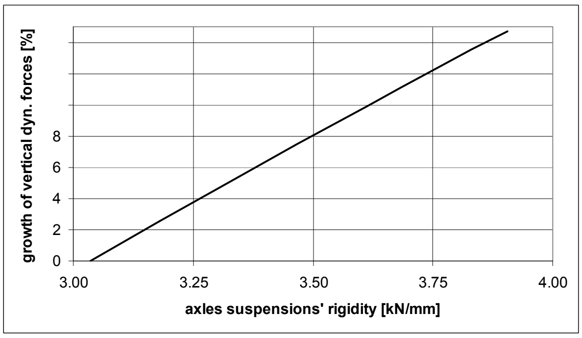

Fig. 7. Percentage growth of vertical dynamic forces at the bogie's frame level in terms of axles suspension rigidity.

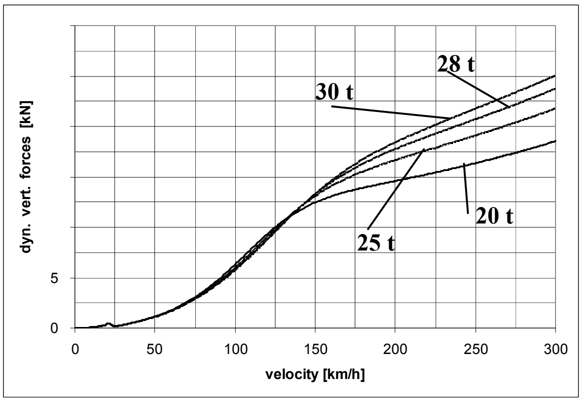

Fig. 8. Dependence of vertical dynamic

Therefore, from this point of view too, follows the necessity of a bigger rigidity concerning the axles suspension's elastic elements and, implicitly, a natural high frequency regarding the acceptable superior limit.

We considered the case of an emergency braking action starting at 300 km/h, at the adhesion limits. In the present case, the use of a 3.9 kN/mm rigidity within the axles suspension would lead on a possible blocking of that suspension level at a speed about 15% smaller (195…220 km/h accountable to 230…255 km/h). We specify that, during a service braking action, commanded with a 0.6…0.7 bar depression in the main brake-pipe, the effect of the axles suspension rigidities is insignificant; theoretically, the suspension level's blocking tendency manifests itself under 50…60 km/h [4].

For almost the same ratio of the two suspension levels rigidities, we also observed the influence of the vehicle's box mass, on the ground of 20…30 t. We considered the same characteristics of the rail and the same masses. The main calculus results are reported in the diagram presented in fig. 8. We noticed that, for running speeds exceeding 150 km/h, the box mass's influence becomes important and determines an increase of the vertical dynamic forces maxims developed at the bogie's level with 35% at a box mass growth from 20 to 30 tons. The increase is almost linear [4].

We also made a study regarding the effects of the suspension levels' rigidities modifications on the amplitude of the vertical oscillations and on the vertical acceleration develpped at the vehicle´s box level.

Within the study we considered a passenger vehicle characterised by a box mass of 28 t, a bogie's suspended mass of 2.5 t, a charge mass of 9 t and an 1.5 t unsprung mass for one mounted axle.The calculus presumed pairs of the suspension's levels rigidities, which respect all the prior conditions and recommendations.

The main results are summarized in the diagrams presented in fig. 9 and 10.

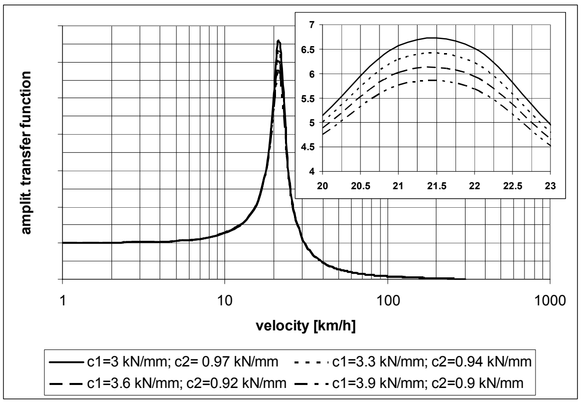

Fig. 9. Amplitude transfer functions of the vertical oscillations of the vehicle's box for different suspension's levels rigidities.

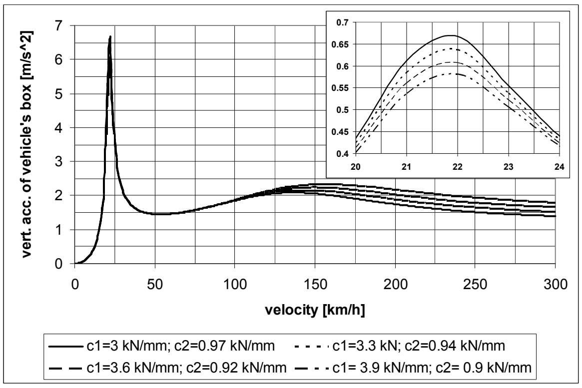

Fig. 10. Vertical accelerations of the vehicle's box for different suspension's levels rigidities.

It can be observed that a increase of the axles suspension's rigidity (implicitly a remission of the central suspension's rigidity) does not affect in a major way the vertical oscillation amplitude of the vehicle's box. Overpassing 125 km/h (in this particular case), we noticed a vertical acceleration's increase which, at the maxim velocity, becomes with almost 30% bigger.

ABOUT THE DUMPING RATIO OF THE SUSPENSION LEVELS

As it is known, the establishment of the optimal dumping ratio reckons on the condition of obtaining minimal values of the oscillating amplitudes. In the two levels suspension, usually the dumping ratio of the axles suspension must be enough for the pitching motion vibrations and the central suspension's dumping ratio is determined in accordance with the maximum accepted values of the oscillation amplitudes or accelerations for the vehicle's [1, 2].

During the braking actions however, the axles suspension's dumping ratio may substantially increase and may possibly reach the blocking situation. That is why, for this particular case, we considered theoretically the influence of an axles suspension's dumping ratio between 0.2...0.5. The main results are summarized in the diagram presented in fig. 11.

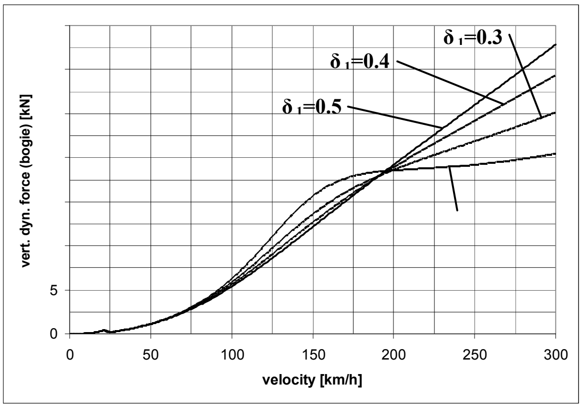

Fig. 11. Vertical dynamic forces at the bogie's frame level for different dumping ratios δ_1.

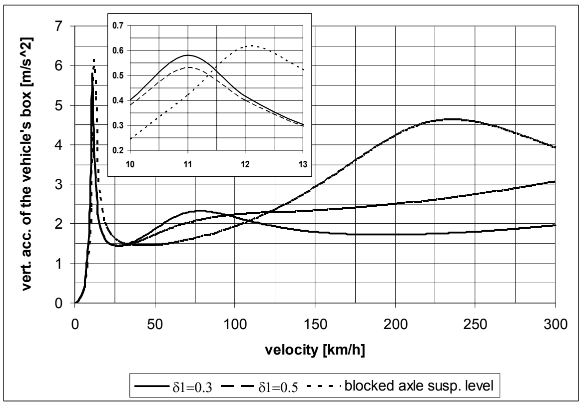

Fig. 12. Vertical accelerations of the vehicle's box for different axle suspension's dumping ratio.

We observed that once with the increase of the dumping ratio, at high speed running can be noticed a substantial vertical dynamic forces increase with more than 30% at the bogie's frame level. At the same time, in the slow speed area, it is noticed a decrease of these with almost 12%.

At the vehicle's box level it can be observed that increasing the dumping ratio from 0.3 to 0.5 and even when the axles suspension is blocked, at least for the studied case, there are no major influences on the vertical oscillating amplitude of the vehicle's box. Instead, its vertical accelerations modify enough both as value and through an obvious displacing of the velocity corresponding to the second maximum (see fig. 13).

CONCLUSIONS

Respecting the imposed conditions and recommendations regarding the adoption of the suspension's elastic characteristics for railway vehicles requires the existence of some correlations between the unssprung mass, the bogie's and box suspended mass and charge mass, which have been presented in this study. Regarding the suspension functioning during the braking actions based on the rail-wheel adhesion, there appear some changes of the dumping ratio of the axles suspension, which affects the vehicle's vibratory regime.

With a view of projecting in an optimal manner the suspension, specially in designing the high speed vehicles which requires a quite long distance for braking and consequently a longer time for running speed decreasing, we consider that it would be necessary to take into consideration such studies of the aspects that we have been pointed out in our paper.

REFERENCES

[1] **Sebeşan, I.**, *Dinamica vehiculelor de cale ferată*, Ed. Tehnică, Bucureşti, 1995, *317 p*.

[2] **Sebeşan, I. Hanganu, D.**, *Proiectarea suspensiilor pentru vehicule pe şine*, Ed. Tehnică, Bucureşti, 1993, *244 p*.

[3] **Fiche Nr. 546**: *FREIN – Freins à haute puissance pour trains de voyageurs*, 5e édition 01.01.1967, Tirage du 01.01.1980, mise à jour 01.01.1983, Union Internationale des Chemins de Fer.

[4] **Cruceanu, C.**, *Frâne pentru vehicule feroviare*, Ed. MATRIXROM, Bucureşti, 2006, 2007, *388 p*.

ОТНОСНО ПРИЕМАНЕТО НА ПАРАМЕТРИТЕ ЗА ВЕРТИКАЛНО ОКАЧВАНЕ ЗА ПЪТНИЧЕСКИТЕ ВАГОНИ

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РУМЪНИЯ

Резюме: *Проектирането на окачването прилага установени еластични и поглъщащи удара константи за условията за осигуряване на действието на динамиката на подвижния състав. За приемането на необходимата вертикална твърдост се прилага определена корелация между конструктивните маси. Геометричните и механичните характеристики на железницата оказват значително въздействие върху реакцията на окачването. Нещо повече – при задействане на спирачки поради силата на триене между дисковете и подложките на спирачките се увеличава факторът за поглъщане на удара при осевото окачване.*

Ключови думи: *окачване, твърдост на окачването, фактор на поглъщане на удара, задействане на спирачки.*