ANNALS OF FACULTY ENGINEERING HUNEDOARA
– International Journal of Engineering
Tome XI (Year 2013) – FASCICULE 4 (ISSN 1584 – 2673)

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EVALUATION OF THE COMFORT INDEX IN RAILWAY VEHICLES DEPENDING ON THE VERTICAL SUSPENSION FEATURES

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ABSTRACT: Numerical simulations can be used to evaluate the comfort index at the railway vehicles. To this purpose, a complex model of the vehicle-track system is used that takes into account a string of factors influencing the level of the vertical vibrations, thus allowing a correct evaluation of the dynamic behaviour in the railway vehicles, mainly at high velocities. The influence of the vertical suspension features upon the vibrating comfort will be examined, as a function of velocity. The notion of the critical point of the carbody vibration behaviour is introduced, as the point where the comfort index is the highest. While assuming the idea to minimize the level of vibrations in the critical point, it is thus proven the possibility of establishing the best damping of the vertical suspension that leads to the best values of the comfort index.
KEYWORDS: railway vehicles, comfort index, vertical vibrations, suspension, track irregularity

INTRODUCTION

The comfort in the railway vehicles is not only a main prerequisite of the modern railway system but also an essential factor in the competition with the other means of transportation. The ride comfort is a complex notion and represents an important criterion while examining the dynamics of the railway vehicles and needs to be taken into account for their modelling and behaviour evaluation. Along other factors, the ride comfort strongly depends on the vibration behaviour of the vehicle [1].

The increase of the velocities will trigger the amplification of the vibration behaviour at the carbody level, with a negative impact on the rolling comfort. This condition is an open issue in the general trend of ‚speeding up', which characterizes the evolution of the railway transport. Hence, maintaining the vibration behaviour at a level that will provide comfort to the passengers requires permanent adjustments in the vehicle construction. A solution relates to the optimization of the suspension characteristics, an element in the railway vehicle construction that plays an important role in providing comfort when it comes to the level of vibrations [2-4].

The evaluation of vibrations in the railway vehicles, and hence of the ride comfort, can be made by measuring and numerical simulations [5-9]. The numerical simulations are an instrument to be used during the designing stage of the railway vehicles, thus providing the opportunity to evaluate the vehicle's dynamic behaviour and estimate the comfort index.

Along the years, many methods were established to assess the ride comfort, with a selective applicability in dependence on the specific of traffic – either urban, suburban, or long distance, etc. These methods are included in a series of standards, such as ISO 2631 [10], BS 6841 [11], Index Sperling Ride [12], ENV-12999 [13] and UIC 513 [14]. All the methods involve the evaluation of the level of vibrations from the comfort perspective based on the acceleration in different directions, via certain weighting filters that trigger the influence of the differentiated sensitivity of the human subjects to vibrations, in dependence on direction and frequency.

The method herein – the comfort index – has been adopted by most railway administrations in Europe. The reason is that it represents a high level and an instrument of finesse that takes into account a number of specific issues of the railway vehicles vibration behaviour, important as they influence the comfort state, but neglected by other methods.

This paper focuses on the numerical simulations to evaluate the comfort index, based on an original model of the vehicle-rolling track system [15]. The model presented here is complex and considers the carbody elasticity, the track elasticity and of the wheel-track, the influence of the system of transmitting the longitudinal efforts, as well as the geometric filter effect derived from the vehicle wheelbase. The paper talks about issues regarding the influence of the vertical suspension *characteristics upon the vibrating comfort, compared to the velocity. In this context, the notion of critical point of the carbody vibration behaviour is being introduced, namely the point along the carbody where the comfort index is the highest. While adopting the idea of minimizing the level of vibrations in the critical point, it is proven possible to establish the best damping of the vertical suspension that will lead to the best values of the comfort index.*

MECHANICAL MODEL AND MOVEMENT EQUATIONS

The case here is of a four-axle vehicle, with two-stage suspension which travels at the constant speed V on a track with irregularities of longitudinal nivelment of a harmonic type. The mechanical model for the study of the vertical vibrations in the vehicle-track system is shown in figure 1.

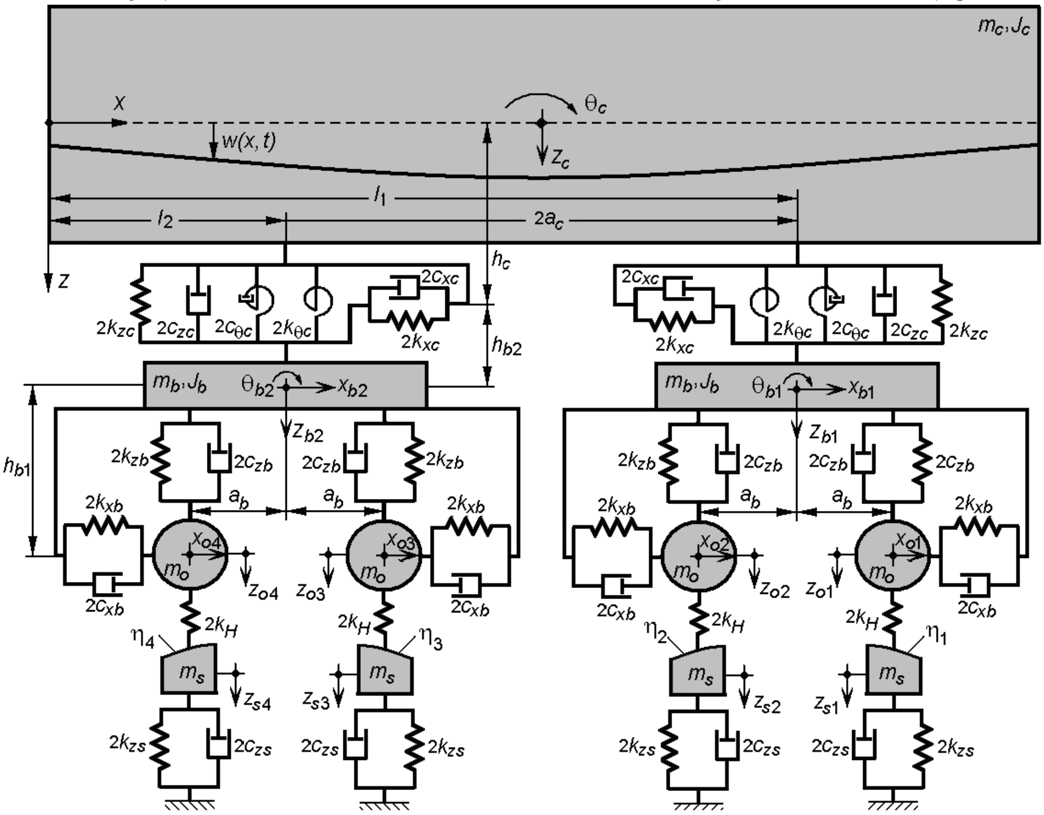


Figure 1. The mechanical model of the vehicle-track system.

The model of the vehicle includes a body with parameters distributed for the carbody and a system of rigid bodies, respectively the wheelsets and the suspended masses of the two bogies.

The carbody of the vehicle with length L and mass m_c is modelled by an Euler-Bernoulli beam of a constant section and a uniformly distributed mass, with the bending module EI, where E is the longitudinal module of elasticity and I is the inertia moment of the beam transversal section. The structural analysis of the carbody structural.

The suspended masses of the bogies are considered rigid bodies on three degrees of freedom, with the following movements: bounce z_{bi}, forward x_{bi} and pitch θ_{bi}, where i = 1, 2. The mass of a bogie is m_b, and its inertia momentum $J_b = m_b i_b^2$, with i_b - the bogie gyration radius.

The wheelsets of mass m_o have two degrees of freedom, thus generating a vertical movement of translation $z_{oj,(j+1)}$ and a longitudinal movement of translation $x_{oj,(j+1)}$, where j = 2i-1, and i = 1, 2, where each bogie is equipped with the wheelsets j and j+1.

Should we neglect the coupling effects between wheels derived from the propagation of the bending waves through rails, for the frequency range that is specific to the vehicle vertical vibrations, then a model equivalent with concentrated parameters will be adopted. Against each wheelset in the bogie i, the track is represented by an oscillatory one- degree of freedom system that can move vertically, and the appropriate bounce is $z_{sj,(j+1)}$, where j = 2i-1 (for i = 1, 2). The track equivalent model has the mass m_s, stiffness k_{zs} and the damping coefficient c_{zs}.

The vehicle suspension, two levels on each bogie, is modelled by means of the Kelvin-Voigt systems. The primary suspension has two Kelvin-Voigt systems that operate on translation, vertically and longitudinally; on the other hand, the carbody suspension has three Kelvin-Voigt systems, two for translation (vertical and longitudinal) and one for rotation. The elements of the Kelvin-Voigt system that take over the angular relative travelling between the carbody and bogie consider the influence of the secondary suspension. The Kelvin-Voigt system on the longitudinal direction between the carbody and the bogie models the system of transmission of the longitudinal forces, located at distance h_c from the carbody neutral fiber and at distance h_{b2} from the center of gravity of the bogie's suspended *mass. The longitudinal Kelvin-Voigt system located in the axle plan, at distance h_{b1} from the mass center of the bogie suspended mass, models the elastic steering of the wheelsets.*

The features of the elastic connections of the secondary suspension, related to a bogie, are represented by the elastic constant values on vertical direction $2k_{zc}$, longitudinal $2k_{xc}$ and the pitch angular stiffness $2k_{\theta c}$, as well as the damping constant values $2c_{zc}$, $2c_{xc}$ and $2c_{\theta c}$. For the primary level of suspension, corresponding to an wheelset, the elastic constant values are noted with $2k_{zb}$ - on vertical direction and with $2k_{xb}$ - on longitudinal direction, while the damping ones are $2c_{zb}$, and $2c_{xb}$ respectively.

To calculate the frequency response, the track irregularities are considered to have a harmonic shape, with the wave length Λ and amplitude ▫▫

$$\eta(x) = \eta_0 \cos(2\pi / \Lambda)x , \tag{1}$$

where x is the coordinate along the track.

The size of the irregularities against each axle depends on their position. While considering the above equation to correspond to the track irregularity in the middle of the vehicle and that the position of the vehicle reported to the track is given in the relation x = Vt, then the defects against the axles are dephased by 2▫($a_c \pm a_b$)/Λ - against the front bogie and by 2▫($-a_c \pm a_b$)/Λ against of the axles of the rear bogie, where $2a_c$ is the distance between bogies and $2a_b$ stands for the bogie wheelbase. As a result, the deviations of irregularity will be

$$\eta_{1,2}(t) = \eta_0 \cos\omega\left(t + \frac{a_c \pm a_b}{V}\right);\ \eta_{3,4}(x) = \eta_0 \cos\omega\left(t - \frac{a_c \mp a_b}{V}\right), \tag{2}$$

where ▫ = $2\pi V/L$ is the angular frequency induced by the track excitation.

Further, once the mechanics laws are implemented, the movement equations of the vehicle-track system are as follows:

- for the carbody bending

$$EI\frac{\partial^4 w(x,t)}{\partial x^4} + \mu I\frac{\partial^5 w(x,t)}{\partial x^4 \partial t} + \rho_c \frac{\partial^2 w(x,t)}{\partial t^2} = \sum_{i=1}^{2} F_{zi}\delta(x - l_i) - \sum_{i=1}^{2}(M_i - h_c F_{xi})\frac{\mathrm{d}\delta(x - l_i)}{\mathrm{d}x}, \tag{3}$$

where $\delta(.)$ is Dirac's delta function, and F_{xi}, F_{zi} and M_i are the forces and the moments derived from the secondary suspension of the bogie i.

The carbody movement equation, which includes partial derivatives, can be changed into equations with ordinary derivatives by implementing the method of modal analysis. For this purpose, the rigid and bending carbody modes are taken into account as

$$w(x,t) = z_c(t) + \left(x - \frac{L}{2}\right)\theta_c(t) + \sum_{n=2}^{\infty} X_n(x)T_n(t), \tag{4}$$

where $z_c(t)$ and $\theta_c(t)$ are the carbody vibration rigid modes, namely the bounce and pitch, $T_n(t)$ is the time coordinate and $X_n(x)$ is the natural function of the mode n of vibration while bending

$$X_n(x) = \sin\beta_n x + \sinh\beta_n x - \frac{\sin\beta_n L - \sinh\beta_n L}{\cos\beta_n L - \cosh\beta_n L}(\cos\beta_n x + \cosh\beta_n x); \tag{5}$$

where $\beta_n = \sqrt{\omega_n^2 m/(EI)}$, with ω_n as the natural angular frequency of the vibration mode n.

- for the bogie i bounce movement

$$m_b \ddot{z}_{bi} = \sum_{j=2i-1}^{2i} F_{zbj} - F_{zi}, \text{ for i = 1,2;} \tag{6}$$

- for the bogie i pitch movement

$$J_b \ddot{\theta}_{bi} = a_b \sum_{j=2i-1}^{2i}(-1)^{j+1}F_{zbj} - h_{b1}\sum_{j=2i-1}^{2i}F_{xbj} - M_i - h_{b2}F_{xi} \text{ for i = 1,2;} \tag{7}$$

- for the bogie i rebound movement

$$m_b \ddot{x}_{bi} = \sum_{j=2i-1}^{2i} F_{xbj} - F_{xi}, \text{ for i = 1,2,} \tag{8}$$

where F_{zbj} represents the forces due to the primary suspension and F_{xbj} the forces derived from the system of the axles elastic guidance.

- for the vertical movement of the axle j, and axle j +1 respectively

$$m_o \ddot{z}_{oj,(j+1)} = 2\Delta Q_{j,(j+1)} - F_{zbj,(j+1)}; \tag{9}$$

- for the longitudinal movement of the axle j, and for the axle j +1

$$m_o \ddot{x}_{oj,(j+1)} = -F_{xbj,(j+1)}, \text{ for j = 2i-1 and i = 1, 2.} \tag{10}$$

To calculate the vertical dynamic forces, the hypothesis of the linear hertzian wheel-rail contact has been adopted

$$\Delta Q_{j,(j+1)} = -k_H [z_{oj,(j+1)} - z_{sj,(j+1)} - \eta_{j,(j+1)}], \text{ for } j = 2i-1 \text{ and } i = 1, 2 \qquad (11)$$

where $\eta_{j,(j+1)}$, are the defects of the track longitudinal irregularity against the axle j, and j+1, and k_H - the rigidity of the wheel-rail contact.

- for the vertical movement of the rail against the axle j, and j +1

$$m_s \ddot{z}_{sj,(j+1)} = F_{zsj,(j+1)} - 2\Delta Q_{j,(j+1)} . \qquad (12)$$

While considering only the first two natural modes of the carbody bending, symmetrical and anti-symmetrical, it results that the vibration of the vehicle-track system is described by a set of 22 equations, coupled with ordinary derivatives. Nevertheless, a correct choice of the coordinates and an appropriate interpretation of the system of equations will lead to its division into two independent systems of ten equations each, which systems describe the symmetrical and anti-symmetrical movements of the vehicle-track system and two decoupled movement equations [16].

THE ASSESSMENT OF THE COMFORT INDEX

As shown earlier, the study of the vehicle vertical vibrations involves the hypothesis that this vehicle runs on a track with random defects of a longitudinal nivelment. Also, it is assumed that the rolling track random defects are stationary in nature.

The spectral density power of the track irregularities can be approximated by a theoretical curve and the literature in review mentions various computation relations that generally concern the track quality. The recommended form by ORE [17] is featured below,

$$S(\Omega) = \frac{A\Omega_c^2}{(\Omega^2 + \Omega_r^2)(\Omega^2 + \Omega_c^2)} , \qquad (13)$$

where Ω is the wave number, Ω_c = 0,8246 rad/m, Ω_r = 0,0206 rad/m, and A = $4{,}032 \cdot 10^{-7}$ rad m or A = $1{,}080 \cdot 10^{-6}$ rad m, depending on the track quality.

While noticing that the track defects become an excitation factor for the vehicle travelling at speed V, the spectral density power of the nivelment defects has to be expressed in dependence of the angle frequency $\omega = V\Omega$, as per the general relation

$$G(\omega) = \frac{S(\omega / V)}{V} . \qquad (14)$$

The equations (13) and (14) will give

$$G(\omega) = \frac{A\Omega_c^2 V^3}{[\omega^2 + (V\Omega_c)^2][\omega^2 + (V\Omega_r)^2]} . \qquad (15)$$

Figure 2 shows the spectral density power of the track irregularities for the velocities of 100 and 200 km/h. The excitation power of the track irregularities can be noticed to increase along the velocity, for frequencies higher than circa 0.25 Hz and decreases at smaller frequencies. This is the first reason why the vehicle level of vibrations goes up with the velocity.

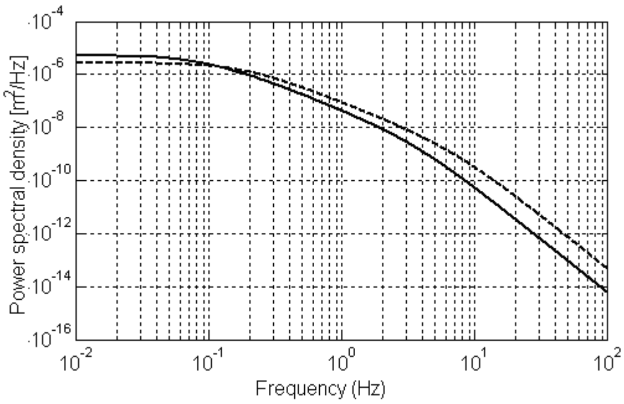


Figure 2. The power spectral density of the track nivelment:
——, 100 km/h; – – –, 200 km/h.

When starting with the vehicle frequency response and the range of the track defects, the spectral density power of the carbody vertical movement can be computed. Thus, for the response factor $\overline{H}_c(x, \omega)$ in a random point x along the carbody

$$\overline{H}_c(\omega, x) = \overline{H}_{z_c}(\omega) + \left(\frac{L}{2} - x\right)\overline{H}_{\theta_c}(\omega) + \sum_{n=2}^{3} X_n(x)\overline{H}_{T_n}(\omega) , \qquad (16)$$

where $\overline{H}_{z_c}(\omega)$, $\overline{H}_{\theta_c}(\omega)$, $\overline{H}_{T_i}(\omega)$ are the response factors corresponding to the rigid modes of vibration - bounce and pitch (z_c şi θ_c) and of the natural modes of symmetrical and anti-symmetrical bending ($T_{2,3}$), then the spectral density power of the carbody vertical movement will be

$$G_c(\omega, x) = G(\omega)\left|\overline{H}_c(\omega, x)\right|^2 . \tag{17}$$

The equation (17) can be customized for various points along the carbody. Thus, in the middle of the carbody, the response factor is

$$\overline{H}_{cm}\left(\omega, \frac{L}{2}\right) = \overline{H}_{z_c}(\omega) + X_2\left(\frac{L}{2}\right)\overline{H}_{T_2}(\omega) , \tag{18}$$

and above the two bogies

$$\overline{H}_{cbi}(\omega, l_i) = \overline{H}_{z_c}(\omega) \pm a_c \overline{H}_{\theta_c}(\omega) + \sum_{n=2}^{3} X_n(l_i)\overline{H}_{T_n}(\omega) , \text{ for } i = 1, 2. \tag{19}$$

The power spectral density of the carbody vertical acceleration are calculated by the relation

$$G_{ca}(\omega, x) = \omega^4 G(\omega)\left|\overline{H}_c(\omega, x)\right|^2 , \tag{20}$$

valid for any point along the carbody.

This is the fundament that underlies the efficient acceleration

$$a_c(x) = \sqrt{\frac{1}{\pi}\int_0^{\infty} G_{ca}(x, \omega)\mathrm{d}\omega} . \tag{21}$$

To quantify the comfort to vibrations, a parameter is needed, i.e. the comfort index, and a scale to connect the values of this parameter and the comfort feeling. Thus, a conventional scale of the comfort index has been set up [14].

Table 1. The significance of the comfort index

Comfort index N	Significance
N < 1	very good comfort
1 ≤ N < 2	good comfort
2 ≤ N < 4	good comfort
4 ≤ N < 5	poor comfort
N ≥ 5	very poor comfort

The method of the comfort index applies differentially, depending on the type of vehicle. For the very good comfort vehicles, this index should be smaller than 2. As for the classical trains, the value is N = 3 the most, and in the passenger cars of high capacity for small distances, $N \leq 4$.

In general, the comfort index is a function of the magnitude of the accelerations in those three directions. When following the simplified method [14], the comfort index can be the result of the below equation

$$N_{MV} = 6\sqrt{\left(a_{c\,XP95}^{W_d}\right)^2 + \left(a_{c\,YP95}^{W_d}\right)^2 + \left(a_{c\,ZP95}^{W_b}\right)^2} , \tag{22}$$

where a_c is the efficient acceleration, X, Y, Z - the directions of measuring the accelerations, P - the floor, 95 represents the quantile of a 95% order, and W_b refers to the weighting filter of the accelerations on a vertical direction.

Weighting filter has a transfer function that reads

$$H_b(s) = \frac{\left(s + 2\pi f_3\right)\cdot\left[s^2 + \frac{2\pi f_5}{Q_3}s + \left(2\pi f_5\right)^2\right]2\pi K f_4^2 f_6^2}{\left[s^2 + \frac{2\pi f_4}{Q_2}s + \left(2\pi f_4\right)^2\right]\left[s^2 + \frac{2\pi f_6}{Q_4}s + \left(2\pi f_6\right)^2\right]f_3 f_5^2} , \tag{23}$$

where f_3 = 16 Hz, f_4 = 16 Hz, f_5 = 2.5 Hz, f_6 = 4 Hz, Q_2 = 0.63, Q_4 = 0.8, K = 0.4.

This filter takes into account the human increased sensitivity to the vertical vibrations in the frequency range of 5.6 - 13 Hz. This filter is used along with another pass band filter W_a with the following transfer function

$$H_a(s) = \frac{s^2\left(2\pi f_2\right)^2}{\left[s^2 + \frac{2\pi f_1}{Q_1}s + \left(2\pi f_1\right)^2\right]\left[s^2 + \frac{2\pi f_2}{Q_1}s + \left(2\pi f_2\right)^2\right]} , \tag{24}$$

with f_1 = 0.4 Hz, f_2 = 100 Hz and Q_1 = 0.71.

While overlooking the influence of the longitudinal and transversal vibrations, the influence of the vertical vibrations upon the comfort index can be estimated as

$$N_{MV} = 6a_{cZP95}^{W_b}. \tag{25}$$

By agreeing to the hypothesis that the vertical accelerations comply with the Gaussian distribution law with a zero average value, the comfort index can be therefore calculated as

$$N_{MV}(x) = 6\Phi^{-1}(0{,}95)\sqrt{\frac{1}{\pi}\int_0^\infty \omega^4 G(\omega)\left|\overline{H}_c(x,\omega)\right|^2\left|\overline{H}_{ab}(\omega)\right|^2 \mathrm{d}\omega}\ , \tag{26}$$

where x places the point along the carbody in which the comfort index is evaluated and $\Phi^{-1}(0{,}95)$ represents the quantile of the standard Gaussian distribution wtih a probability of 95%, and $\overline{H}_{ab}$ is the weighting factor of the two filters W_a and W_b.

To determine the comfort index by a simplified method, the accelerations in three reference points will be considered: in the middle of the carbody and above the two bogies.

NUMERICAL APPLICATION

This section concentrates on the results of the numerical simulations regarding the influence of the suspension characteristics upon the vibrating comfort, evaluated by the comfort index. Such simulations rely on the model and method in the previous sections. The parameters of the numerical simulation have been properly defined for a passengers car equipped with bogies Y 32R, with the maximum velocity of 200 km/h.

In order to facilitate the analysis, the damping ratio of each suspension level is introduced

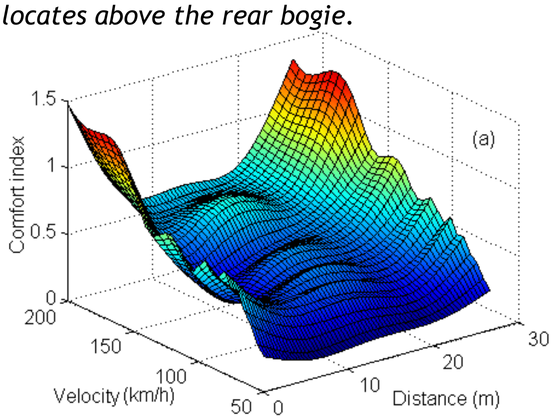
$$\zeta_{zb,c} = \frac{4c_{zb,c}}{2\sqrt{4k_{zb,c}m_{b,c}}}. \tag{27}$$

Similarly, the concept of the critical point of the carbody vibration behaviour is introduced, namely the point along the carbody where the comfort index is the highest.

Figure 3 (diagram (a)) represents the distribution of the comfort index along the carbody when the vehicle travels at speeds between 50 and 200 km/h on a good quality track ($A = 4.032\cdot10^{-7}$ radm) with nivelment irregularities described by the spectral density power, as in relation (16). The diagram (b) shows the comfort index in the reference points, in the middle of the carbody and above the two bogies.

The distribution of the comfort index is noticed to be asymmetrical with respect to the carbody middle, with different laws in dependence on the velocity (fig. 3, diagram (a)). In general, the comfort index increases along with the velocity, but this amplification is not uniform, due to the geometric filtering effect [18], which is visible in the middle of the carbody.

As a rule, the comfort index is lower in the middle of the carbody and increases against the two bogies and towards its ends. The diagram (b) demonstrates that the comfort index has high values above the rear bogie up to the speed of 90 km/h – a speed at which the critical point of the comfort index moves to the middle of the carbody; then, for a large range of speed, 97....180 km/h, this critical point is above the front bogie. Between 180 and 200 km/h, the critical point of the comfort index locates above the rear bogie.



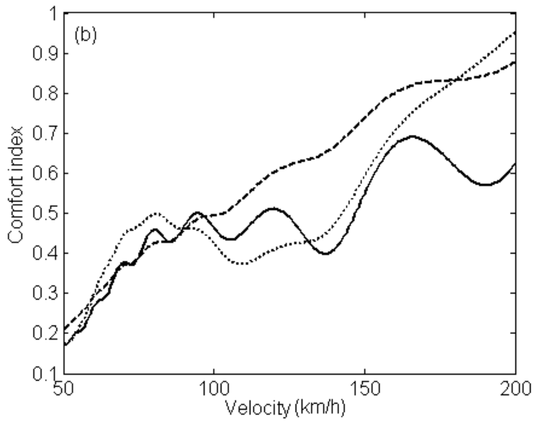


Figure 3. The comfort index: (a) the comfort index along the carbody; (b) the comfort index in the reference points: —— in the middle of the carbody, – – above the front bogie, · · above the rear bogie.

The damping of the secondary suspension exerts a significant influence upon the comfort to vibrations, as seen in figure 4 (diagrams (a), (b) and (c)), where the comfort index is presented in the middle of the carbody and above the two bogies, for velocities between 50 and 200 km/h. Values in *the range of 0.05...0.5 have been taken into account for the damping ratio of the secondary suspension, while the reference values have been maintained for the other parameters of the vehicle.*

Table 2. The damping ratio of the secondary suspension that minimizes the comfort index, depending on velocity

Velocity [km/h]	*100*	*150*	*200*
Reference point	*The damping ratio of the secondary suspension*		
in the middle of the carbody	*0.22*	*0.11*	*0.16*
above bogie 1	*0.28*	*0.27*	*0.22*
above bogie 2	*0.15*	*0.21*	*0.28*

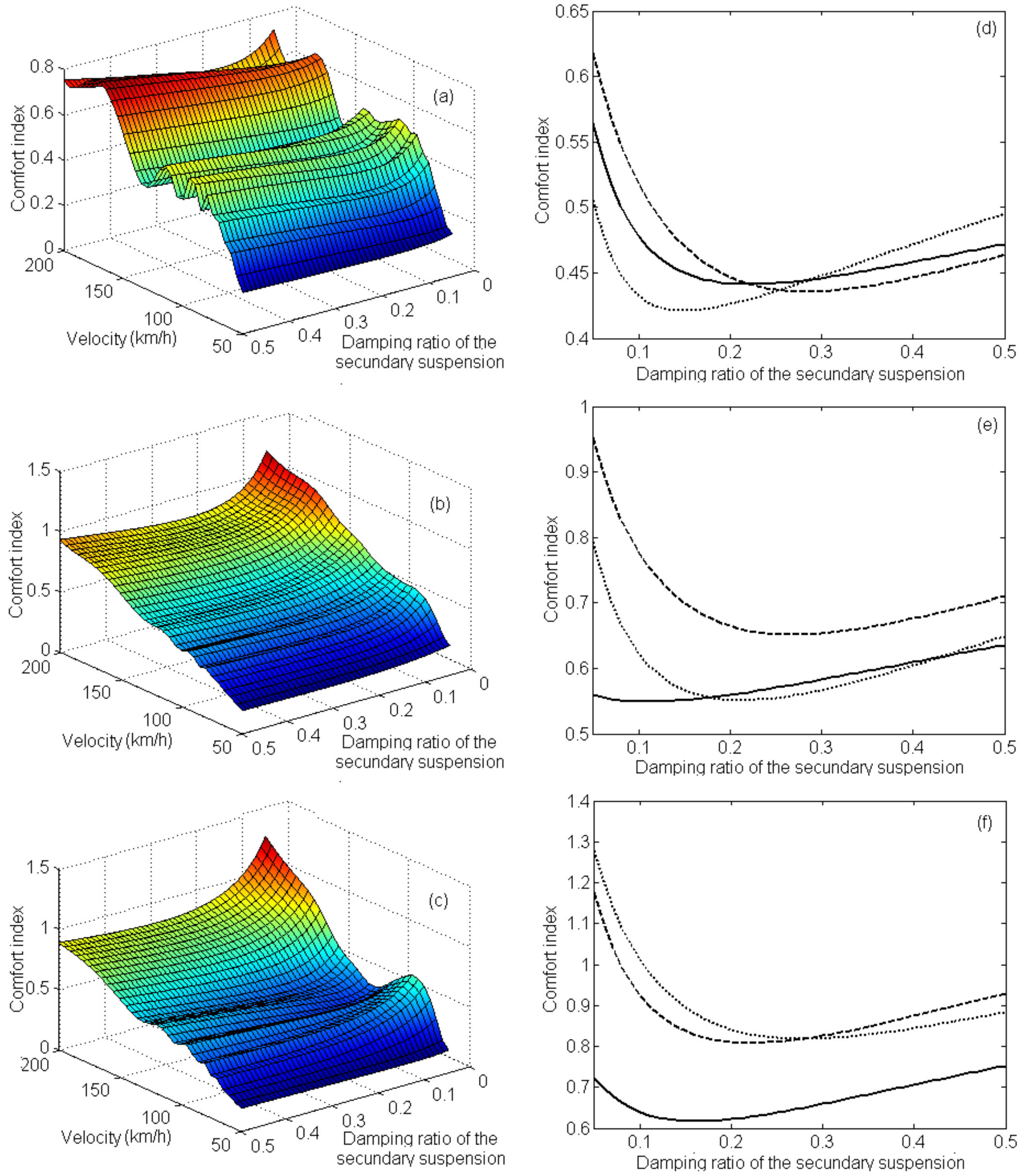


Figure 4. Influence of the secondary suspension damping and of the velocity upon the comfort index: (a) in the middle of the carbody; (b) above the front bogie; (c) above the rear bogie; (d) 100 km/h; (e) 150 km/h; (f) 200 km/h: ——, in the middle of the carbody, – – –, above the front bogie, · · · ·, above the rear bogie.

In fact, there is a value for damping that minimizes the carbody vibration behaviour in each of the reference points, no matter the velocity. This shows better in diagrams (d), (e) and (f), which feature the comfort index in all three reference points for the velocities of 100, 150 and 200 km/h. The examination of the diagrams truly proves that the comfort index in these points has a minimum value that corresponds to a certain damping ratio of the suspension. This one depends on both the position along the carbody (in the middle or above the bogies) and on the velocity, as presented in Table 2.

It can be noticed that the comfort index is minimum in the middle of the carbody, at high velocities and for small values of the damping ratio. On the contrary, the index is higher above the bogies and its minimalization is done for $\square_c$ = 0.22 and $\square_c$ = 0.28, respectively.

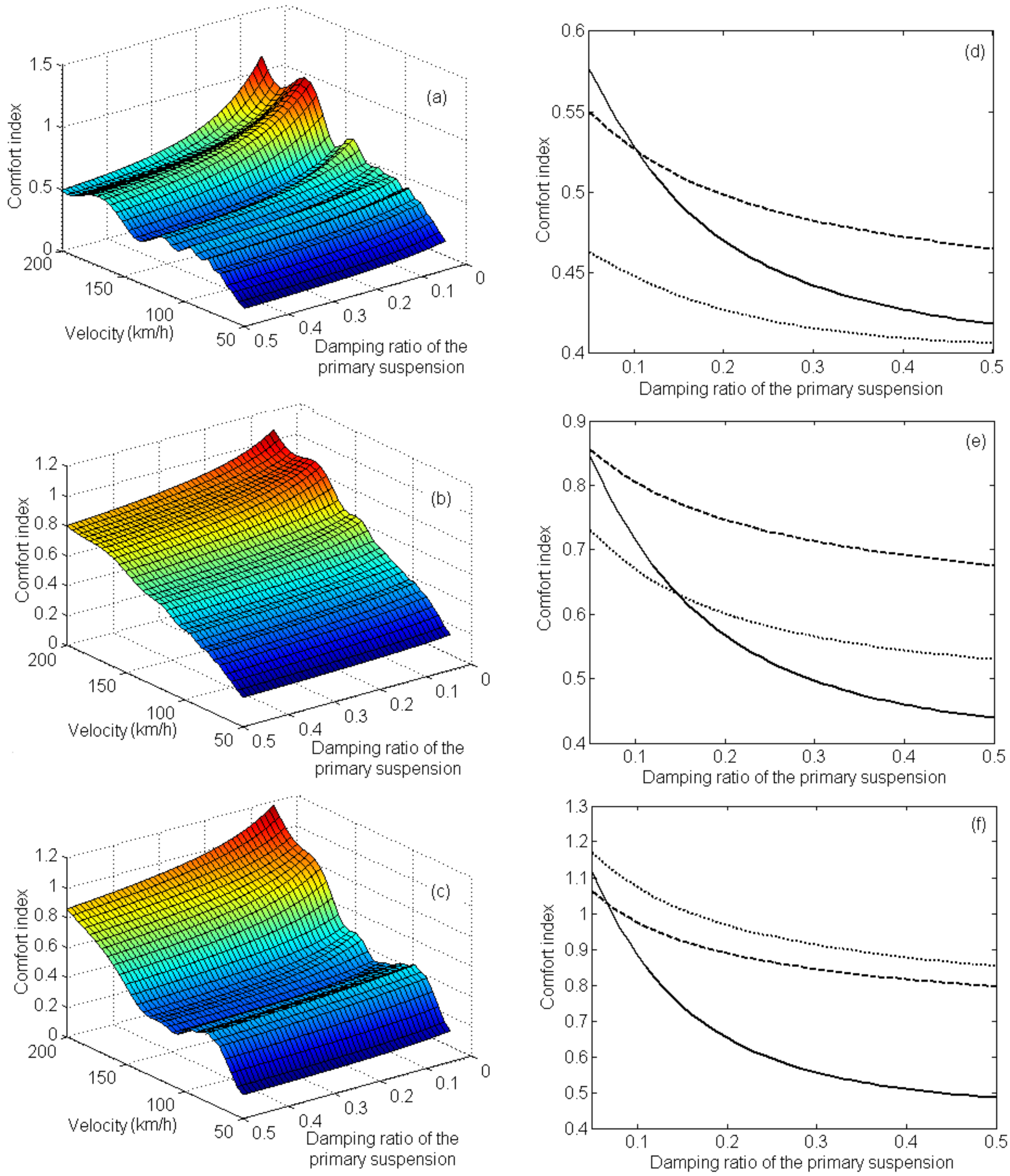


Figure 5. Influence of damping upon the primary suspension and of the velocity upon the comfort index: (a) in the middle of the carbody; (b) above the front bogie; (c) above the rear bogie; (d) 100 km/h; (e) 150 km/h; (f) 200 km/h: ——, in the middle of the carbody, – – –, above the front bogie, · · · ·, above the rear bogie.

The conclusion is that an increase in the damping up to a certain value will trigger a general improvement of the vibrating comfort. Beyond this limit, the damping has opposite effects – for example, it improves the comfort above the bogies, while the value in the middle of the carbody is lowered. For low damping values, the comfort to vibrations worsens in any of the reference points. The explanation is that, on the one hand, small damping values limit the vibration to the resonance frequency and, on the other hand, when damping is high, this will lead to an increase of the dynamic rigidity of the system and, therefore, the vibration behaviour is more intense.

As a rule, the idea of minimizing the level of vibrations in the carbody critical point can be adopted (where the comfort index is the highest) in order to obtain the best damping. For example, in the diagram (e), the critical point is above the front bogie and the damping ratio that minimizes the comfort index in this point is $\square_c$ *= 0.27. There are conditions where the identification of the critical point is not obvious and the selection of the best damping difficult, since the vibration behaviour depends on the carbody damping along the carbody in a different manner, as already shown.*

Thus, in the diagram (d), it can be noted that the critical point is above the front bogie for small damping, around the middle of the carbody at around $\square_c$ *= 0,25 and then further to move* *towards the rear bogie as the damping goes up. Similarly, at a velocity of 200 km/h, the diagram (f) shows how the critical point moves along the increase of ζ_c over the value of 0.28, from the rear to the front bogie.*

Another reference would be related to the fact that the best damping value varies differently, as a function of velocity (Table 2). Thus, the best damping ratio of the secondary suspension over the rear bogie that leads to minimum values of the comfort index increases along the velocity, while for the front bogie this ratio decreases under the same circumstance.

Should for the secondary suspension there is a damping ratio that, no matter the velocity and the point considered along the carbody, will lead to the minimization of the comfort index, things go different for the primary suspension.

As seen in figure 5 (diagrams (a), (b) and (c)), within the standard interval of 0.05 ... 0.5 looked at for ζ_b, there is no damping value that will minimize the vibration level, at no velocity or in any reference point.

The increase of the damping means a lower comfort index, and this decrease is practically the same in all the points. Hence, there are no sensitive changes in the critical point position of the vibration behaviour in the carbody. This is the conclusion from the details of diagrams (d), (e) and (f), where the variation of the comfort index while damping raises is represented for the three velocities adopted as reference, namely 100, 150 and 200 km/h.

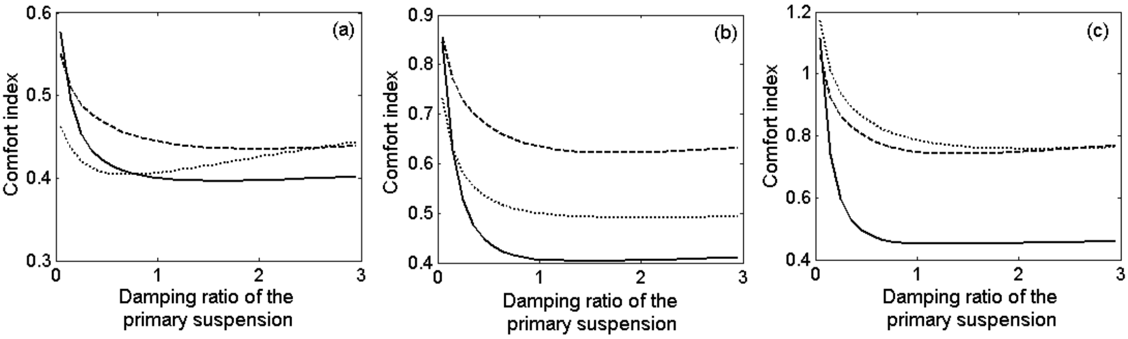


Figure 6. The damping ratio of the primary suspension that minimizes the comfort index:
(a) in the middle of the carbody; (b) above the front bogie; (c) above the rear bogie;
——, in the middle of the carbody, – – –, above the front bogie, · · · ·, above the rear bogie.

It is worthwhile mentioning that this case also includes a damping value that minimizes the carbody vibration level in any of its points - but this value is extremely high, as in figure 6, and on that account it has no practical interest.

CONCLUSIONS

Estimating the vibration behaviour is a relevant stage in designing the railway vehicles. The suspension, thanks to its features, plays an important role in terms of the vibration behaviour at the carbody and so in fulfilling the conditions brought by achieving comfort in the passenger vehicles.

Relying on the numerical simulations, the paper focuses on the influence of the vertical suspension damping upon the comfort in the railway vehicles, evaluated by the comfort index. To this end, a complex discrete-continuum model of the vehicle-track has been used - the model includes a beam for the carbody, a system of rigid bodies namely the wheelsets and the suspended mass of the bogies and also a system equivalent with parameters concentrated for the track. The complexity of the model lies in the fact that it considers a series of factors that influence the level of the vertical vibrations: the carbody elasticity, the elasticity of the rolling track and of the wheel-rail contact, the influence of the transmission system of the longitudinal efforts, as well as the geometric filtering effect introduced by the vehicle axle base.

The issues here have pointed out at a string of specific aspects of the vibration behaviour in the railway vehicles. Thus, is has been shown that the level of vibrations is not uniform along the carbody, based on the asymmetrical distribution of the comfort index along the carbody. In general, the comfort index is smaller in the middle of the carbody and increases against the two bogies and towards its ends. Similarly, the distribution of the comfort index along the carbody has been demonstrated to follow different laws, depending on velocity.

The investigation regarding the influence of the secondary suspension damping upon the comfort index has proven that, as a rule and no matter the velocity, there is a damping value that minimizes the level of the carbody vibration in any of its points, a value that depends though on the position along the carbody and velocity.

The selection of a damping ratio for a railway vehicle can be a sensitive issue, as a too small damping can compromise the vehicle dynamic performance, while a too high value will lead to an *increase in the dynamic rigidity of the system and, hence, to an intensification of the vibration behaviour.*

While introducing the concept of critical point of the carbody vibration behaviour as being the point along the carbody where the comfort index is the highest, the idea came to minimize the level of vibrations in this point so as to obtain the best damping. In this context, it has been proven that the best value of the secondary suspension damping varies differently as a function of velocity and the position of the critical point that is usually found against one of the two bogies. As for the primary suspension, when the damping has its usual numbers, there is no value to lead to the minimization of the level vibration in the carbody. Nevertheless, the increase in the damping ratio takes to a smaller comfort index, and this decrease is practically uniform in all the points along the carbody.

When comparing the influence of the damping of the two-level suspension upon the comfort index, it can be relatively said that the primary suspension damping has a lower influence. This aspect can be explained by the fact that the bogie mass will interfere between the primary suspension and the carbody and its inertia reduces the efficiency of the primary suspension damping.

REFERENCES

[1] Mazilu, T., Confortul la materialul rulant (Comfort at the rolling stock), Matrix Rom Publishing House, 2003.

[2] Sebeșan I., Mazilu, T., Vibrațiile vehiculelor feroviare (Vibrations of the railway vehicles), MatrixRom Publishing House, Bucharest, 2010.

[3] Liao, Y., Wang, T., Yang, Y., Optimization of suspension parameters on ride quality of railway passenger vehicles, International Conference on Electrical and Control Engineering (ICECE), 4001 – 4004, 2011.

[4] Younesiana, D., Nankali, A., Spectral optimization of the suspension system of high-speed trains, International Journal Vehicle Structures & Systems, 1(4), 98-103, 2009.

[5] Carlbom, P., Carbody and passengers in rail vehicle dynamics, Doctoral thesis, Railway Technology, Department of Vehicle Engineering, Royal Institute of Technology, Stockholm, 2000.

[6] Sugahara, Y., Kazato, A., Takigami, T., Koganei, R., Suppression of vertical vibration in railway vehicles by controlling the damping force of primary and secondary suspensions, QR of RTRI, Vol. 49, No. 1, 2008, pp. 7–15.

[7] Sugahara, Y., Kazato, A., Koganei, R., Sampei, M., Nakaura, S., Suppression of vertical bending and rigid-body-mode vibration in railway vehicle car body by primary and secondary suspension control: Results of simulations and running tests using shinkansen vehicle, Journal of Rail and Rapid Transit, Vol. 223, No. F6, 2009, pp. 517–531.

[8] Suzuki, H.:, Research trends on riding comfort evaluation in japan, Journal of Rail and Rapid Transit, Vol. 212, No. F1, 1998, pp. 61–72.

[9] Dumitriu, M., Influence of the suspension damping on ride comfort of passenger railway vehicles, UPB Scientific Bulletin, Series D: Mechanical Engineering, iss. 4, vol. 74, 2012, pp. 75-90.

[10] ISO 2631-4, Mechanical vibrations and shock- evaluation of human exposure to whole body vibrations - Part 4: Guidelines for the evaluation of the effects of vibration and rotational motion on passenger and crew comfort of fixed guide way transport systems, 2004.

[11] BS 6841, 1987, Guide to measurement and evaluation of human exposure to whole-body mechanical vibration and repeated shock, BSI, 1987.

[12] Sperling, E., Betzhold, C., Beitrag zur beurteilung des fahrkomforts in schienenfahrzeugen, Glasers Annalen, Vol. 80, pp. 314-320, 1956.

[13] ENV12299, Railway applications: Ride comfort for passengers, 1999.

[14] UIC 513R, Guidelines for evaluating passenger comfort in relation to vibration in railway vehicle, International Union of Railways, 1994.

[15] Dumitriu, M., Modelling of vehicle-track system to study the vertical vibrations, Annals of the "Constantin Brâncuși" University of Târgiu Jiu issue. 4, 2012, Editura "Academica Brâncuși", pp. 53-64.

[16] Dumitriu, M., Influence of the vertical suspension on the vibration behavior in the railway vehicles, Annals of the University of Petroșani, Mechanical Engineering, vol. 13, 2011, pp. 35-50.

[17] C 116, Interaction between vehicles and track, RP 1, Power spectral density of track irrgularities, Part 1: Definitions, conventions and available data, Utrecht, 1971.

[18] Dumitriu, M., Geometric filtering effect of vertical vibrations of railway vehicles, Analele Universității "Eftimie Murgu" Resița, nr. 1, 2012, pp.48-61.