# AN APPROACH TO IMPROVING THE RIDE COMFORT OF THE RAILWAY VEHICLES

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*The paper addresses the issue of improving the ride comfort of the railway vehicles in terms of isolating the vibrations conveyed to the carbody through the best choice of the vertical damping of the secondary suspension of the vehicle. To evaluate the ride comfort in the relevant points of the carbody - in the centre and above the bogies, the partial comfort index for vertical vibrations is being used, derived from numerical simulations. The results of the numerical simulations point out at the manner in which the damping ratio of the secondary suspension affects the ride comfort index in these points, depending on velocity. Based on these observations, correlated with those linked to the possibilities to minimize the ride comfort index, a series of aspects is highlighted to underlie the selection criteria for the best damping of the secondary suspension so that the ride comfort performance of the vehicle be improved in all three relevant points of the carbody.*

**Keywords**: railway vehicle, vertical vibration, ride comfort, suspension damping, ride comfort index

## 1. Introduction

While train riding, the comfort of passengers can be affected by various factors, where some of them derive from the movement of the railway vehicle, such as vibrations and noise, whereas others from the environment conditions within the vehicle - temperature, humidity and air speed, lighting - or the outside fitting and facilities (e.g. the shape and placement of chairs) [1].

Of all the factors influencing the comfort of passengers, vibrations are paid a great attention since their effect upon the human body is extremely important [2-5]. In dependence on the intensity of vibrations, direction and frequency of occurrence, as well as the exposure time, vibrations can harm the human health or his ability to conduct certain sedentary activities. Generally, the vibrations in the railway vehicle are considered the main factor to determine the ride comfort [6-8].

Ride comfort is one of the criteria for evaluating the dynamic behaviour in railway vehicles [8, 9]. This is used to describe the degree of the passengers'

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comfort from the point of view of mechanical vibrations, taking into account the physiological characteristics of the human body [10].

The issue of improvement in the ride comfort has turned into an active research topic considering that a decrease in the weight of the vehicle, particularly in the carbody that is the most important component in the total weight of the vehicle, has become an essential criterion for designing the railway vehicles so that they reach the highest possible velocities, at the lowest energy consumption. Moreover, by reducing the weight of the vehicle the vibrations transmitted through the ground and the manufacturing costs are reduced.

The weight lightening design of carbody implies using light materials and altering mechanical structures [11], which often leads to a lower carbody structural stiffness. The lighter the vehicle carbody, the higher its flexibility that will thus facilitate an easy excitation of the carbody structural vibrations that have a negative effect upon the ride comfort.

The literature of review features more concepts regarding the improvement in the ride comfort of the railway vehicles. Such concepts can be classified depending on how they are dealt with, namely [12, 13]: approaches concerning the isolation of vibrations, with passive, semi-active and active concepts, aiming to reduce the transmission of the excitation of the carbody via suspension, and approaches regarding the damping of the carbody vibrations, which on their turn have passive or active concepts, intending to lower the amplitude in the structural vibrations of the carbody. As a rule, the ride comfort can be improved through either the optimization of the suspension parameters, or the identification of certain solutions targeting the reduction of the structural vibrations in the carbody [14].

The issue of improving the ride comfort of the railway vehicles is addressed in the paper from the perspective of isolating the vibrations conveyed to the carbody through the best possible choice of the damping of the secondary suspension. The work has as its starting point an important aspect highlighted by the main author in several previous works. It regards the fact that, for any of the points in the vehicle carbody – relevant for the ride comfort evaluation – the value of the damping of the secondary suspension which minimizes ride comfort index at the vertical vibrations can be identified [15-18]. The points that are relevant in terms of the ride comfort evaluation to be found on the longitudinal axis of the carbody that passes through its centre of gravity and are located as such – one point at the centre of the carbody and two points against the two bogies (against the leaning points of the carbody on the secondary suspension).

The results included in this paper are derived from numerical simulations developed as based on a rigid-flexible coupled type vehicle model, where the modes of vertical vibrations relevant for the ride comfort are taken into account – the bounce, pitch and bending of the carbody. The results of the numerical

simulations indicate a number of aspects to underlie the criteria of selecting the best damping of the secondary suspension, so that the ride comfort performances of the vehicle to be improved in all three relevant points of the carbody.

## 2. The mechanical model of railway vehicle

To study the possibility of improving the ride comfort at the vertical vibrations through the best choice for the damping of the secondary suspension, a rigid-flexible coupled type vehicle model is adopted. This model includes the carbody modelled through a free-free equivalent Euler-Bernoulli beam, with constant section and uniform distributed mass, Euler-Bernoulli type, and six rigid bodies representing the bogies and the axles of the vehicle (see figure 1). This type of model is often used in studies regarding the vertical vibrations of the railway vehicles and the ride comfort [15-21], thanks to the fact that it displays a good agreement between the numerical simulations and the field tests [22]. The parameters of the vehicle model are presented in table 1.

Relevant points from the point of view of the ride comfort evaluation are marked on the vehicle model, namely point $C$ – at the centre of the carbody and points $B_1$ and $B_2$ – above the bogies (against the leaning points of the carbody on the secondary suspension).
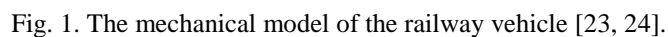


Fig. 1. The mechanical model of the railway vehicle [23, 24].

The parameters of the vehicle model.

| Symbol | Definition | Symbol | Definition |
|--------|-----------|--------|-----------|
| $m_c$ | carbody mass | $L_c$ | carbody length |
| $m_b$ | bogie suspended mass | $2a_c$ | carbody wheelbase/ |
| $J_c$ | carbody inertia moment | $2a_b$ | bogie wheelbase |
| $J_b$ | bogie inertia moment | $l_{1,2}$ | supporting points position of the carbody on the suspension |

The secondary suspension corresponding to a bogie is modelled via three Kelvin-Voigt systems, i.e. two for translation (vertical and longitudinal) and one for the rotation. The model of the secondary suspension comprises important elements - the pitch angular stiffness and the stiffness of the connection between the carbody and the bogie – by which the bogie pitch vibrations are transmitted to the carbody and excite its vertical bending modes. In using the Kelvin - Voigt system, with the elastic constant $2k_{\theta c}$ and the damping constant $2c_{\theta c}$, the rotation couple between the carbody and the bogie introduced by the springs of the secondary suspension is considered when the laying plans are not parallel. The Kelvin-Voigt system operating in translation, with the elastic constant $2k_{xc}$ and the damping constant $2c_{xc}$, models the system of transmission for the longitudinal forces between the carbody and the bogie. The Kelvin-Voigt system for translation on a vertical direction has the elastic constant $2k_{zc}$ and the damping constant $2c_{zc}$.

The primary suspension corresponding to an axle is modelled through a single Kelvin-Voigt system for translation on the vertical direction, with the elastic constant $2k_{zb}$ and the damping constant $2c_{zb}$.

As for the track model is concerned, the rigid track hypothesis is accepted. This simple approach is justified by the fact that the eigenfrequencies of the vehicle carbody affecting the ride comfort are much lower than the ones in the axle-track system.  In the vehicle axles, the rigid track triggers vertical displacements equal with its irregularities. In figure 1, the track vertical irregularities are described against each axle through the functions $\eta_{1...4}$.

The carbody vibration modes of the carbody relevant for the ride comfort in the vertical plan are considered, such as the bounce $z_c$, pitch $\theta_c$, and the first bending mode. The vertical displacement of the carbody $w_c(x, t)$ is the result of having the rigid vibrations modes - bounce and pitch overlapped, and the vertical bending

$$w_c(x,t) = z_c(t) + \left( x - \frac{L_c}{2} \right)\theta_c(t) + X_c(x)T_c(t) , \tag{1}$$

where $T_c(t)$ is the coordinate of the carbody bending and $X_c(x)$ represents the natural function of this vibration mode, described in the equation

$$X_c(x) = \sin \beta x + \sinh \beta x - \frac{(\sin \beta L_c - \sinh \beta L_c)(\cos \beta x + \cosh \beta x)}{\cos \beta L_c - \cosh \beta L_c} , \qquad (2)$$

with
$$\beta = \sqrt[4]{\omega_c^2 \rho_c /(EI)} , \; \cos \beta L_c \cosh \beta L_c - 1 = 0 , \qquad (3)$$

where $\omega_c$ is the natural pulsation of the carbody bending.

The motion equation of the carbody vehicle has the general form

$$EI \frac{\partial^4 w_c(x,t)}{\partial x^4} + \mu I \frac{\partial^5 w_c(x,t)}{\partial x^4 \partial t} + \rho_c \frac{\partial^2 w_c(x,t)}{\partial t^2} =$$
$$= \sum_{i=1}^{2} F_{zci} \delta(x - l_i) + \sum_{i=1}^{2} (M_{ci} - h_c F_{xci}) \frac{\mathrm{d}\delta(x - l_i)}{\mathrm{d}x} , \qquad (4)$$

where $EI$ is bending modulus (with $E$ - longitudinal modulus of elasticity, and $I$ - inertia moment of the beam's transversal section), $\mu$ - structural damping coefficient, and $\rho_c = m_c/L_c$ – beam mass per length unit; $w_c(x, t)$ represents the vertical displacement in a random point of the carbody; $\delta(.)$ is Dirac's delta function; $F_{zci}$, $F_{xci}$ and $M_i$ stand for forces and moments, respectively, due to the secondary suspension of bogie $i$ (with $i = 1, 2$)

$$F_{zci} = -2c_{zc} \left( \frac{\partial w_c(l_i,t)}{\partial t} - \dot{z}_{bi} \right) - 2k_{zc} \left[ w_c(l_i,t) - z_{bi} \right], \qquad (5)$$

$$F_{xci} = 2c_{xc} \left( h_c \frac{\partial^2 w_c(l_i,t)}{\partial x \partial t} + h_b \dot{\theta}_{bi} \right) + 2k_{xc} \left( h_c \frac{\partial w_c(l_i,t)}{\partial x} + h_b \theta_{bi} \right) \qquad (6)$$

$$M_{ci} = -2c_{\theta c} \left( \frac{\partial^2 w_c(l_i,t)}{\partial x \partial t} - \dot{\theta}_{bi} \right) - 2k_{\theta c} \left[ \frac{\partial w_c(l_i,t)}{\partial x} - \theta_{bi} \right] . \qquad (7)$$

The application of the modal analysis and the orthogonality property of the eigenfunctions of the carbody bending can help infer the bounce, pitch and vertical bending equations,

$$m_c \ddot{z}_c + 2c_{zc}[2\dot{z}_c + 2\varepsilon \dot{T}_c - (\dot{z}_{b1} + \dot{z}_{b2})] + 2k_{zc}[2z_c + 2\varepsilon T_c - (z_{b1} + z_{b2})] = 0 \qquad (8)$$

$$J_c\ddot{\theta}_c + 2c_{zc}a_c[2a_c\dot{\theta}_c - (\dot{z}_{b1} - \dot{z}_{b2})] + 2k_{zc}a_c[2a_c\theta_c - (z_{b1} - z_{b2})] +$$
$$+ 2c_{xc}h_c[2h_c\dot{\theta}_c + h_b(\dot{\theta}_{b1} + \dot{\theta}_{b2})] + 2k_{xc}h_c[2h_c\theta_c + h_b(\theta_{b1} + \theta_{b2})] + \qquad (9)$$
$$+ 2c_{\theta c}[2\dot{\theta}_c - (\dot{\theta}_{b1} + \dot{\theta}_{b2})] + 2k_{\theta c}[2\theta_c - (\theta_{b1} + \theta_{b2})] = 0 ;$$

$$m_{mc}\ddot{T}_c + c_{mc}\dot{T}_c + k_{mc}T_c +$$
$$+ 2c_{zc}\varepsilon[2\dot{z}_c + 2\varepsilon\dot{T}_c - (\dot{z}_{b1} + \dot{z}_{b2})] + 2k_{zc}\varepsilon[2z_c + 2\varepsilon T_c - (z_{b1} + z_{b2})] +$$
$$+ 2c_{xc}h_c\lambda[2h_c\lambda\dot{T}_c + h_b(\dot{\theta}_{b1} - \dot{\theta}_{b2})] + 2k_{xc}h_c\lambda[2h_c\lambda T_c + h_{b2}(\theta_{b1} - \theta_{b2})] + \qquad (10)$$
$$+ 2c_{\theta c}\lambda[2\lambda\dot{T}_c - (\dot{\theta}_{b1} - \dot{\theta}_{b2})] + 2k_{\theta c}\lambda[2\lambda T_c - (\theta_{b1} - \theta_{b2})] = 0 ,$$

where $k_{mc}$ is the carbody modal stiffness, $c_{mc}$ - carbody modal damping and $m_{mc}$ - carbody modal mass

$$k_{mc} = EI\int_0^L \left(\frac{\mathrm{d}^2 X_c}{\mathrm{d}x^2}\right)^2 \mathrm{d}x ; \quad c_{mc} = \mu I\int_0^L \left(\frac{\mathrm{d}^2 X_c}{\mathrm{d}x^2}\right)^2 \mathrm{d}x ; \quad m_{mc} = \rho_c\int_0^L X_c^2\mathrm{d}x . \qquad (11)$$

The notations $\varepsilon$ and $\lambda$ have been introduced, based on the symmetry properties of the eigenfunction $X_c(x)$,

$$X_c(l_1) = X_c(l_2) = \varepsilon , \quad \frac{\mathrm{d}X_c(l_1)}{\mathrm{d}x} = -\frac{\mathrm{d}X_c(l_2)}{\mathrm{d}x} = \lambda . \qquad (12)$$

The vibration modes of the bogies in the vertical plan are the bounce $z_{b1,2}$ and the pitch $\theta_{b1,2}$. The equations describing the bounce and pitch motions of the bogies are

$$m_b\ddot{z}_{b1} = \sum_{i=1}^2 F_{zbi} - F_{zc1} , \quad m_b\ddot{z}_{b2} = \sum_{i=3}^4 F_{zbi} - F_{zc2} , \qquad (13)$$

$$J_b\ddot{\theta}_{b1} = a_b\sum_{i=1}^2 \pm F_{zbi} - h_bF_{xc1} , \quad J_b\ddot{\theta}_{b2} = a_b\sum_{i=3}^4 \pm F_{zbi} - h_bF_{xc2} , \qquad (14)$$

where $F_{zb1\dots4}$ stands for the forces due to the primary suspension, given in the relations below

$$F_{zb1,2} = -2c_{zb}(\dot{z}_{b1} + a_b\dot{\theta}_{b1} - \dot{\eta}_{1,2}) - 2k_{zb}(z_{b1} + a_b\theta_{b1} - \eta_{1,2}) , \qquad (15)$$

$$F_{zb3,4} = -2c_{zb}(\dot{z}_{b2} - a_b\dot{\theta}_{b2} - \dot{\eta}_{3,4}) - 2k_{zb}(z_{b2} - a_b\theta_{b2} - \eta_{3,4}) . \qquad (16)$$

The motion equations of the bogies write as such:

$$m_b \ddot{z}_{b1,2} + 2c_{zb}[2\dot{z}_{b1,2} - (\dot{\eta}_{1,3} + \dot{\eta}_{2,4})] + 2k_{zb}[2z_{b1,2} - (\eta_{1,3} + \eta_{2,4})] +$$
$$+ 2c_{zc}(\dot{z}_{b1,2} - \dot{z}_c \mp a_c\dot{\theta}_c - \varepsilon\dot{T}_c) + 2k_{zc}(z_{b1,2} - z_c \mp a_c\theta_c - \varepsilon T_c) = 0 \; ; \qquad (17)$$

$$J_b \ddot{\theta}_{b1,2} + 2c_{zb}a_b[2a_b\dot{\theta}_{b1,2} - (\dot{\eta}_{1,3} - \dot{\eta}_{2,4})] + 2k_{zb}a_b[2a_b\theta_{b1,2} - (\eta_{1,3} - \eta_{2,4})] +$$
$$+ 2c_{xc}h_b[h_b\dot{\theta}_{b1,2} + h_c(\dot{\theta}_c \pm \varepsilon\dot{T}_c)] + 2k_{xc}h_b[h_b\theta_{b1,2} + h_c(\theta_c \pm \varepsilon T_c)] = 0 \qquad (18)$$

The motion equations for the carbody and bogies (8-10 and 17-18) make up a 7-equation system with ordinary derivatives, which can be solved numerically using MATLAB code. The system can be matrix-like written

$$\mathbf{M\ddot{p}} + \mathbf{C\dot{p}} + \mathbf{Kp} = \mathbf{P\dot{\eta}} + \mathbf{R\eta} , \qquad (19)$$

where $\mathbf{M}$, $\mathbf{C}$ and $\mathbf{K}$ are the inertia, damping and stiffness matrices, $\mathbf{p} = [z_c \quad \theta_c \quad T_c \quad z_{b1} \quad \theta_{b1} \quad z_{b2} \quad \theta_{b2}]^\mathrm{T}$ represents the vector of the displacements' coordinates and $\mathbf{\eta}$ is the vector of the heterogeneous terms.

### 3. The ride comfort index-based to evaluate the ride comfort

To quantify the comfort to vibrations, a parameter is needed, i.e. the ride comfort index, and a scale to connect the values of this parameter and the comfort feeling. Thus, the conventional scale of the ride comfort index has been set up, as seen in figure 2 [25, 26].

To evaluate only the ride comfort in the vertical direction, the partial comfort index is used, calculated with [25]

$$N_{MV} = 6a_{95}^{W_{ab}} \qquad (20)$$

where $a$ is the root mean square of the vertical carbody acceleration, 95 refers to the quantile of order 95%, and $W_{ab} = W_a \cdot W_b$ represents the weighting filter of the vertical acceleration.

To evaluate the ride comfort at the carbody centre and above the bogies, the root mean square of the vertical acceleration is calculated

$$a_m = \sqrt{\frac{1}{\pi}\int_0^\infty G_{cm}(\omega)\mathrm{d}\omega} \;\; , \;\; a_{b1,2} = \sqrt{\frac{1}{\pi}\int_0^\infty G_{cb1,2}(\omega)\mathrm{d}\omega} \;\; , \qquad (21)$$

Fig. 2. The significance of the ride comfort index.

where $G_{cm}(\omega)$ is the power spectral density of the carbody vertical acceleration in the point $C$ on the carbody and $G_{cb1.2}(\omega)$ is the power spectral density of the carbody vertical acceleration in the points $B_{1,2}$, which is calculated depending on the power spectral density of the track vertical irregularities $G(\omega)$ and the frequency response function of the carbody vertical acceleration at the carbody centre $\overline{H}_{cm}(L_c / 2, \omega)$, and above the two bogies, respectively $\overline{H}_{cb1,2}(l_{1,2}, \omega)$

$$G_{cm}(\omega) = G(\omega)\left|\overline{H}_{cm}(L_c / 2, \omega)\right|^2, \ G_{cb1,2}(\omega) = G(\omega)\left|\overline{H}_{cb1,2}(l_{1,2}, \omega)\right|^2. \qquad (22)$$

The theoretical curve of the power spectral density is deemed representative for the average statistical properties of the European railways [27]

$$G(\omega) = \frac{A\Omega_c^2 V^3}{[\omega^2 + (V\Omega_c)^2][\omega^2 + (V\Omega_r)^2]}, \qquad (23)$$

where $\Omega_c = 0.8246$ rad/m, $\Omega_r = 0.0206$ rad/m, and $A$ is a coefficient depending on the track quality. For a high-level quality track, $A = 4.032 \cdot 10^{-7}$ radm, whereas for a low-level quality, the coefficient $A$ is $1.080 \cdot 10^{-6}$ radm.

To calculate the frequency response function, the track vertical irregularities against each axle are considered to be in a harmonic shape, with the wavelength $\Lambda$ and amplitude $\eta_0$. The track vertical irregularities are out of phase against the axles corresponding to the distances between them, $2a_c$ and $2a_b$.

Hence, the functions $\eta_j$ describing the track irregularities against the four axles (see figure 1) are in the form of

$$\eta_{1,2}(t) = \eta_0 \cos\omega\left(t + \frac{a_c \pm a_b}{V}\right), \quad \eta_{3,4}(t) = \eta_0 \cos\omega\left(t - \frac{a_c \mp a_b}{V}\right). \tag{24}$$

where $\omega = 2\pi V/\Lambda$ represents the track excitation-induced angular frequency.

The frequency response function of the carbody vertical acceleration at the carbody centre (in the point $C$) is calculated with the relation

$$\overline{H}_{cm}(L_c/2,\omega) = \omega^2\left[\overline{H}_{z_c}(\omega) + X_c(L_c/2)\overline{H}_{T_c}(\omega)\right] \tag{25}$$

and above the two bogies (the points $B_1$ and $B_2$)

$$\overline{H}_{cb1,2}(l_{1,2},\omega) = \omega^2\left[\overline{H}_{z_c}(\omega) \pm a_c\overline{H}_{\theta_c}(\omega) + X_c(l_{1,2})\overline{H}_{T_c}(\omega)\right], \tag{26}$$

where $\overline{H}_{z_c}(\omega)$, $\overline{H}_{\theta_c}(\omega)$, $\overline{H}_{T_c}(\omega)$ are the frequency response functions corresponding to the rigid vibration modes of bounce and pitch ($z_c$ and $\theta_c$), and to the vertical bending in the carbody ($T_c$).

The filter $W_a$ is a passband filter with the following transfer function [25, 26]

$$H_a(s) = \frac{s^2(2\pi f_2)^2}{\left[s^2 + \frac{2\pi f_1}{Q_1}s + (2\pi f_1)^2\right]\left[s^2 + \frac{2\pi f_2}{Q_1}s + (2\pi f_2)^2\right]}, \tag{27}$$

with $s = i\omega$ ($i^2 = -1$), $f_1 = 0.4$ Hz, $f_2 = 100$ Hz, $Q_1 = 0.71$.

The weighting filter $W_b$ takes into account the high human sensitivity to the vertical vibrations within the frequency interval of 3 ... 13 Hz and has the transfer function in the form of [25, 26]

$$H_b(s) = \frac{(s + 2\pi f_3)\cdot\left[s^2 + \frac{2\pi f_5}{Q_3}s + (2\pi f_5)^2\right]2\pi K f_4^2 f_6^2}{\left[s^2 + \frac{2\pi f_4}{Q_2}s + (2\pi f_4)^2\right]\left[s^2 + \frac{2\pi f_6}{Q_4}s + (2\pi f_6)^2\right]f_3 f_5^2}, \tag{28}$$

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

Adopting the hypothesis that the vertical accelerations have a Gaussian distribution with the zero mean value, the partial comfort index derives from:
- at the carbody centre

$$N_{MV_m} = 6\Phi^{-1}(0.95)\sqrt{\frac{1}{\pi}\int_0^\infty G(\omega)\omega^2 \left[\overline{H}_{z_c}(\omega) + X_c(L_c/2)\overline{H}_{T_c}(\omega)\right] \left|H_{ab}(i\omega)\right|^2 d\omega}$$

(29)

- above the bogies

$$N_{MV_{b1,2}} = 6\Phi^{-1}(0.95)\sqrt{\frac{1}{\pi}\int_0^\infty G(\omega)\omega^2 \left[\overline{H}_{z_c}(\omega) \pm a_c\overline{H}_{\theta_c}(\omega) + X_c(l_{1,2})\overline{H}_{T_c}(\omega)\right] \left|H_{ab}(i\omega)\right|^2 d\omega}$$

(30)

where $\Phi^{-1}(0.95)$ represents the quantile of the standard Gaussian distribution with the probability of 95%, and the transfer function $H_{ab}(i\omega)$ is calculated as below

$$H_{ab}(i\omega) = H_a(i\omega)H_b(i\omega) .$$

(31)

## 4. Numerical simulations results

This section deals with the possibilities of improving the ride comfort through the best choice of the damping of the secondary suspension. To this purpose, the results derived from the numerical simulations on the comfort index at vertical vibrations are used, calculated for velocities ranging from 50 km/h to 300 km/h during running on a low quality track ($A = 1.080\cdot10^{-6}$ radm).

The parameters of the numerical model for the vehicle are included in Table 2. The model parameters are similar to the ones at an empty passenger vehicle (top speed of 300 km/h).

*Table 2.*

The parameters of the vehicle numerical model

| | | |
|---|---|---|
| $m_c = 34.0\cdot10^3$ kg | $L_c = 26$ m | $k_{zc} = 0.60$ MN/m |
| $m_b = 3.20\cdot10^3$ kg | $EI = 3.02\cdot10^9$ Nm$^2$ | $k_{xc} = 2.00$ MN/m |
| $m_{mc} = 35.2\cdot10^3$ kg | $h_c = 1.30$ m | $k_{\theta_c} = 128$ kN/m |
| $k_{mc} = 89.0$ MN/m | $h_b = 0.20$ m | $c_{xc} = 25.0$ kNs/m |
| $c_{mc} = 53.1$ kNm/s | $2a_c = 19.0$ m; $2a_b = 2.56$ m | $c_{\theta_c} = 1$ kNm |
| $J_c = 1.96\cdot10^6$ kg·m$^2$ | $l_1 = 3.5$ m | $k_{zb} = 1.10$ MN/m |
| $J_b = 2.05\cdot10^3$ kg·m$^2$ | $l_2 = 22.5$ m | $c_{zb} = 13.05$ kNs/m |

Influence of the secondary suspension damping upon the ride comfort is analyzed using the damping ratio

$$\zeta_c = \frac{4c_{zc}}{2\sqrt{4k_{zc}m_c}},$$

(32)

considering values between 0.05 and 0.4.

Figure 3 shows the comfort index calculated in the relevant points in terms of the ride comfort – at the carbody centre and above the bogies, for velocities between 50 km/h and 300 km/h and four values of the damping ratio of the secondary suspension.
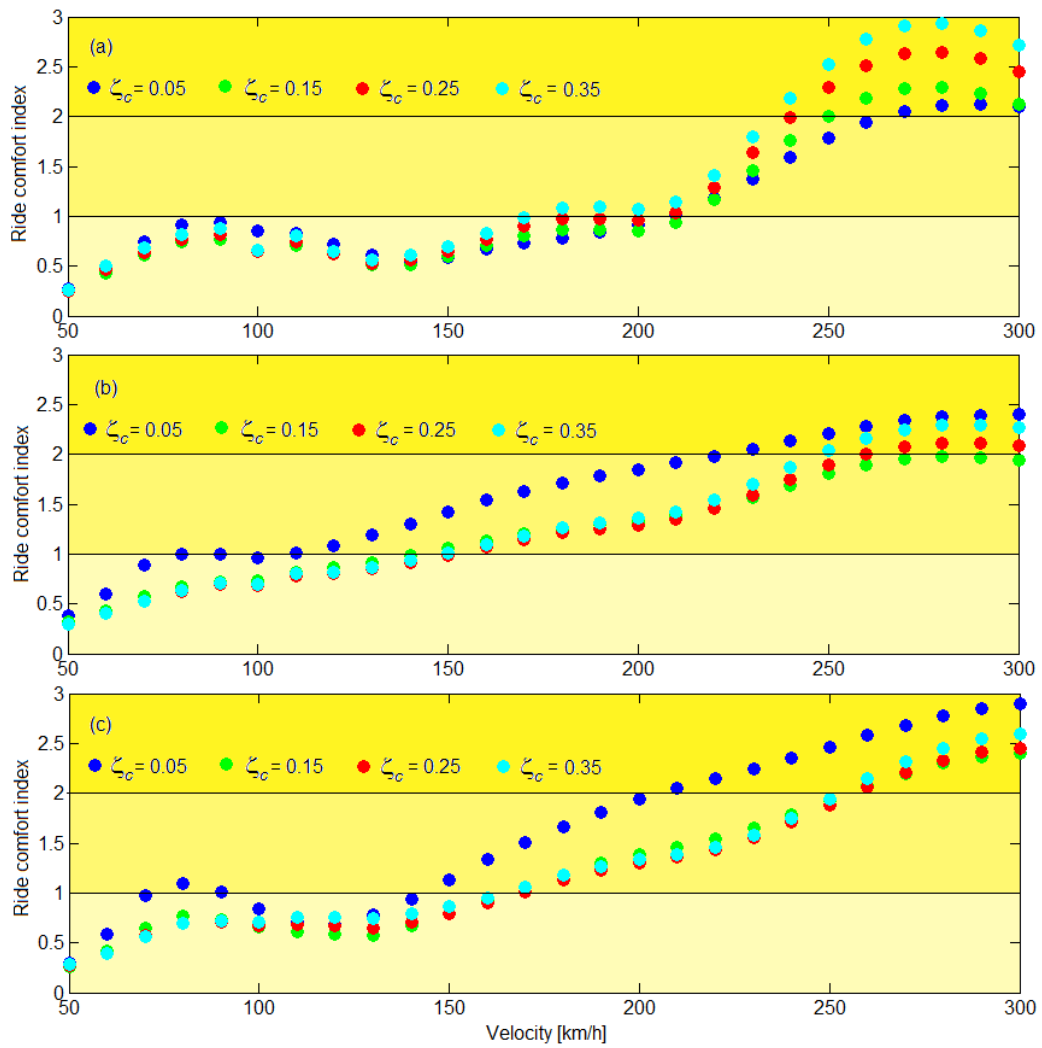


Fig. 3. Ride comfort index – influence on the damping ratio of the secondary suspension:
(a) at the carbody centre; (b) above the front bogie; (c) above the rear bogie.

The damping ratio of the secondary suspension is noticed to differently affect the ride comfort index at the carbody centre and above the bogies. At the carbody centre, up to the velocity of 150 km/h, the damping ratio of the secondary suspension does not sensitive influence the value of the ride comfort index. The higher the velocity, the more important the effect of the damping ratio of the secondary suspension on the ride comfort index, becoming visible at velocities higher than 230 km/h – 240 km/h. For the velocity interval of 150 – 300 km/h, the best comfort performances of the vehicle are recorded for a reduced damping of the secondary suspension ($\zeta_c = 0.05$). Above the bogies, for the entire interval of velocities under study, the ride comfort index does not significantly change for certain values of the damping ratio of the secondary suspension ($\zeta_c$ between 0.15 and 0.35). The highest values for the ride comfort index are present when the damping ratio of the secondary suspension is very small ($\zeta_c = 0.05$).
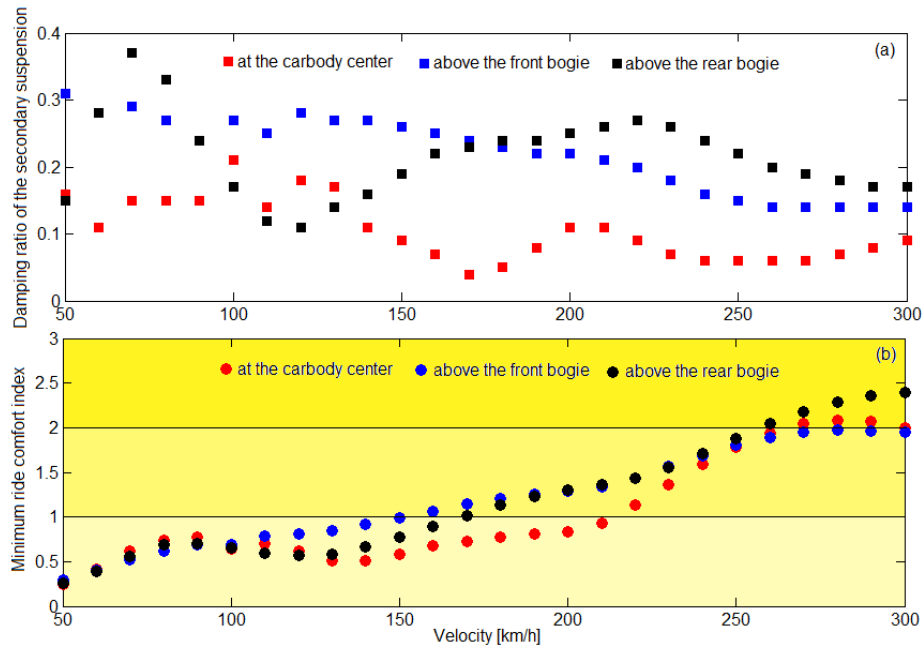


Fig. 4. (a) The damping ratio of the secondary suspension that minimizes the ride comfort index;
(b) The minimum comfort index

The diagrams in figure 4 feature the values of the damping ratio of the secondary suspension for which minimum values of the ride comfort index are obtained at the carbody centre and above the two bogies, depending on velocity. The values of the damping ratio of the secondary suspension to minimize the ride comfort index are noticed to be dispersed out into a large interval, between 0.04 and 0.37. The values of $\zeta_c$ to minimize the ride comfort index at the carbody

centre range from 0.04 to 0.21. For the smallest values of the ride comfort index above the front bogie, the damping ratio of the secondary suspension is required to be included between 0.14 and 0.31 and between 0.11 and 0.37 for the rear bogie.
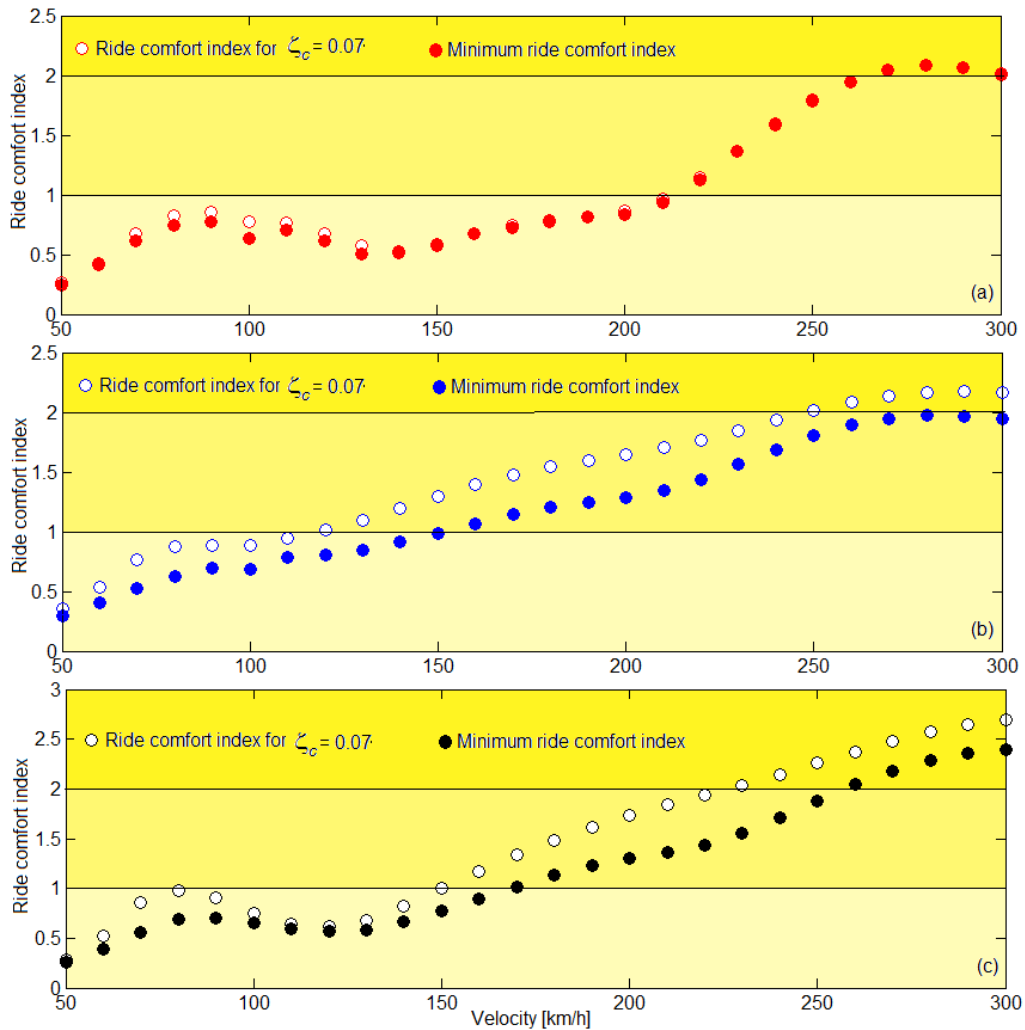


Fig. 5. Ride comfort index for $\zeta_c = 0.07$: (a) at the carbody centre;
(b) above the front bogie; (c) above the rear bogie.

It is difficult, therefore, to establish a value of the damping ratio of the secondary suspension to minimize the ride comfort index in all three points of the carbody that are relevant for the ride comfort.

Under such circumstances, certain criteria are necessary to be formulated to support the choice of the best damping of the secondary suspension, so that the

ride comfort performances of the vehicle are improved in all three points of the carbody. For instance, these criteria should consider the following aspects evidenced by the results above:

1. The ride comfort worsens at high velocities; according to figure 3, at the carbody centre, the ride comfort is very good and good, respectively, up to the velocity of 230 km/h, irrespective of the damping ratio of the secondary suspension. Above the bogies, for $\zeta_c \geq 0.15$, the ride comfort is very good and good up to the velocity of 250 km/h.

2. For the interval of 250 - 300 km/h, the highest ride comfort index is recorded at the carbody centre and above the rear bogie. To minimize the ride comfort index at the carbody centre within this velocity interval, the damping ratio is required to fall within 0.06 - 0.08; for the rear bogie, $\zeta_c$ should be between 0.17 and 0.22 (see figure 4, diagram (a)).

In line with the observations above, the damping of the secondary suspension should be selected in such a way to lead to the improvement in the ride comfort at high velocities, in the points where the ride comfort index has the highest values. Nevertheless, to have a ride comfort index minimized at the carbody centre, a very low damping ratio of the secondary suspension is necessary, which as shown above (see figure 3, diagram (b) and diagram (c)), triggers an increase in the ride comfort index in the points above the bogies, for the entire interval of velocities. For instance, in diagrams (b) and (c) in figure 5, the difference between the minimum ride comfort index and the ride comfort index calculated for $\zeta_c = 0.07$ is evident.

On the other hand, the damping ratio of the secondary suspension, which minimizes the ride comfort index above the rear bogie does not bring significant improvements of the ride comfort at the carbody centre. This aspect is visible in diagram (a) in figure 6 where the ride comfort index was first calculated for an average value of the damping ratio of the secondary suspension in the interval 0.17 ... 0.22, i.e. $\zeta_c = 0.20$. Then, while considering the observations according to which the best comfort performances at the carbody centre are possible for the lowest possible values of the damping ratio of the secondary suspension and that the ride comfort index does not greatly change above the bogies for $\zeta_c$ between 0.15 and 0.35 (see figure 3), a lower value of the damping ratio of the secondary suspension is adopted, namely $\zeta_c = 0.15$ (see figure 6). The decrease in the ride comfort index at the carbody centre is noticed, with no obvious changes of this index above the two bogies.

The figure 7 features the ride comfort index calculated for more values of the damping ratio of the secondary suspension between 0.05 and 0.40. The minimum values of the ride comfort index are shown in red, while the values in green are for $\zeta_c = 0.15$, which can be herein considered the best value of the damping of the secondary suspension.
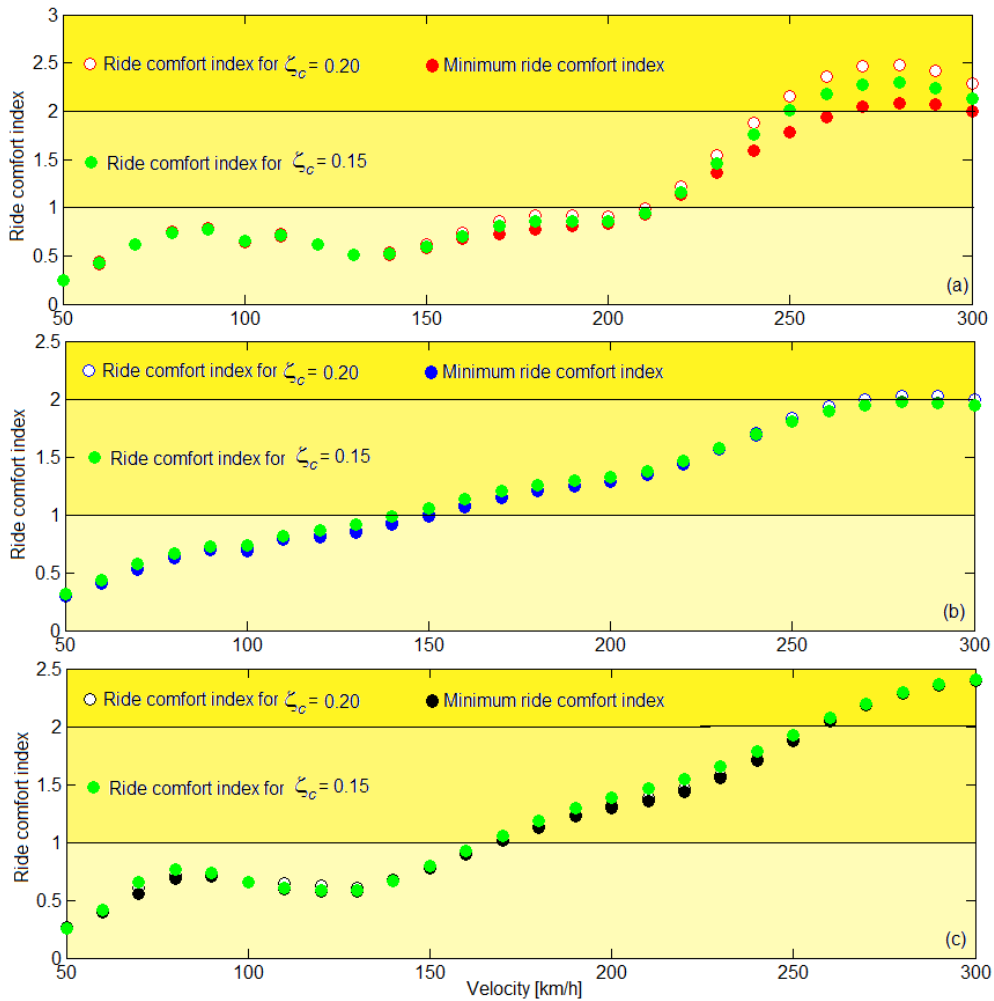
Fig. 6. Ride comfort index for $\zeta_c = 0.20$ and $\zeta_c = 0.15$: (a) at the carbody centre;
(b) above the front bogie; (c) above the rear bogie.

## 6. Conclusions

This paper deals with the issue of improving the ride comfort in the railway vehicles, starting from the concept that a value of the damping of the secondary suspension to minimize the ride comfort index at the vertical vibrations can be identified for each of the three relevant points of the carbody in terms of evaluating the ride comfort.

The results of the numerical simulations concerning the ride comfort derived for an extended interval of velocities, between 50 and 300 km/h, have highlighted the fact that the values of the damping ratio of the secondary suspension for which minimum value of the ride comfort index are obtained at the carbody centre and above the two bogies are very dispersed.
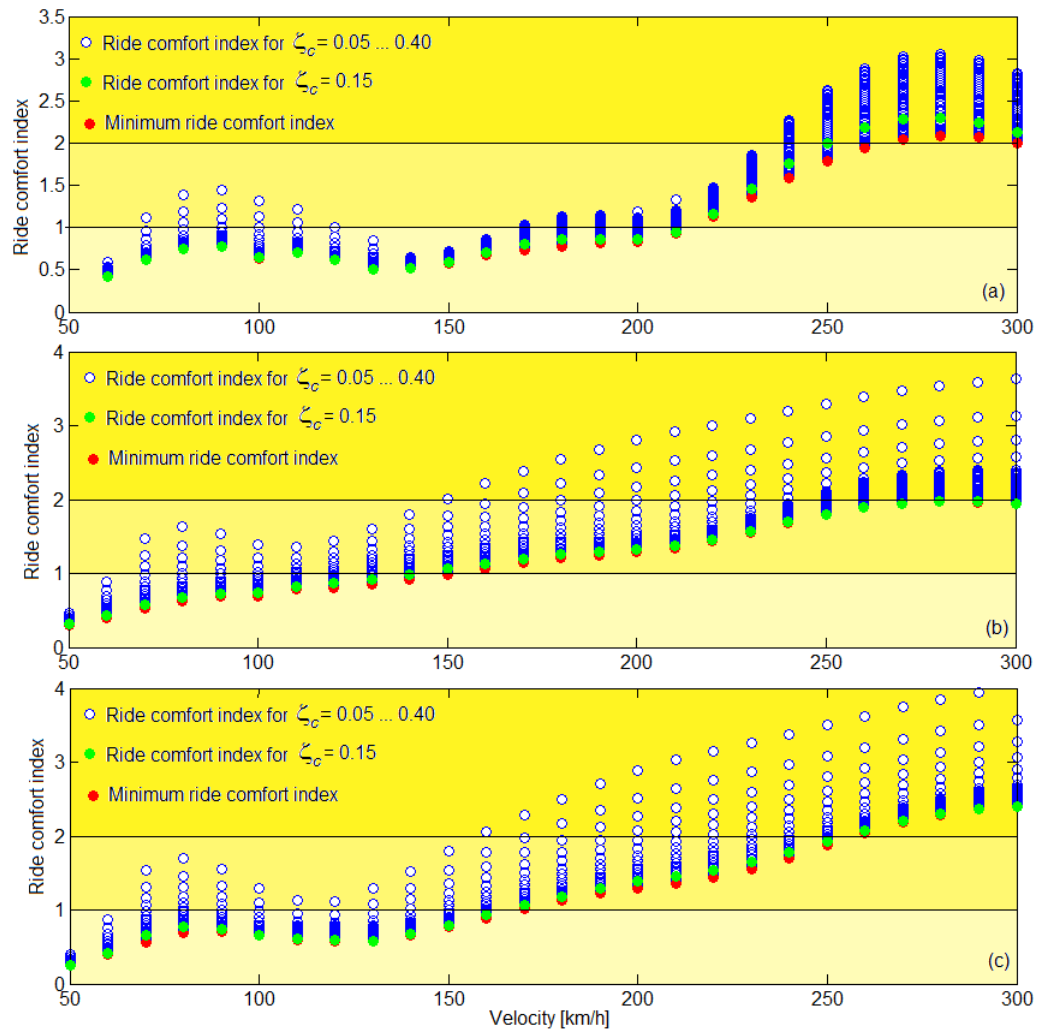
Fig. 7. Ride comfort index: (a) at the carbody centre;
(b) above the front bogie; (c) above the rear bogie.

There are very large differences between the damping ratio of the secondary suspension required to minimize the ride comfort index at the carbody centre and the damping ratio of the secondary suspension to minimize the ride comfort index above the bogies. Generally speaking, a very low damping ratio of the secondary suspension is necessary to minimize the ride comfort index at the carbody centre, which leads to a deterioration of the ride comfort in the points above the bogies.

The conclusion is that it is rather difficult to establish a value for the damping ratio of the secondary suspension to simultaneously minimize the ride comfort index in all three points of the carbody that are relevant to the ride

comfort. Under such circumstances, the selection of the best damping of the secondary suspension needs to underlie certain criteria, such as the velocity range and the points where the ride comfort is the least favourable. In the end, the choice of the best damping of the secondary suspension relies on a compromise to lead to the best performance of ride comfort in all three points of the carbody.

# R E F E R E N C E S

[1]     *M. Dumitriu*, Confortul vibratoriu la vehiculele feroviare (Ride comfort of railway vehicles), Editura Matrix Rom, Bucureşti, 2019.

[2]     *N. J. Mansfield*, Human response to vibration, CRC Press LLC, 2005.

[3]     *M.J. Griffin*, Handbook of Human Vibration, London, Academic Press, 1990.

[4]     *G.L. Richards*, On the psychology of passenger comfort, In Human factors in transport research, **vol. 2**, Academic Press, New York, 1980.

[5]     *H. Suzuki*, Effects of the range and frequency of vibrations on the momentary riding comfort evaluation of a railway vehicle, Japanese Psychological Research, **vol. 40**, issue 3, 1998, pp. 156-165.

[6]     *E. Kardas-Cinal,* Ride comfort for various passenger positions in a railway vehicle – simulation study, The Archives of Transport, **vol. 22**, issue 2, 2010, pp. 189-199.

[7]     *Y. G. Kim, S. Choi, S.W. Kim, Y. M. Kim, T. W. Park*, An experimental study on the ride comfort of the Korean high-speed train, Techniques, **vol. 33**, issue 6, 2009, pp. 30 – 37.

[8]     *Y. G. Kim, H. B. Kwon, S. W. Kim, C. K. Park, T. W. Park*, Correlation of ride comfort evaluation methods for railway vehicles, Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit, **vol. 217**, 2003, pp. 73-88.

[9]     *X. Wei, Y. Liu, Q. Sun, L., Y. Wang*, Rail vehicle ride comfort prediction based on bogie acceleration measurements, 25th Chinese Control and Decision Conference (CCDC), 2013, pp. 3810-3815.

[10]   *S. Iwnicki*, Handbook of railway vehicle dynamics, Taylor & Francis Group, LLC, 2006.

[11]   *W. Sun, J. Zhou, D. Gong, T. You*, Analysis of modal frequency optimization of railway vehicle car body, Advances in Mechanical Engineering, **vol. 8**, issue 4, 2016, pp. 1–12.

[12]   *M. Kozek, C. Benatzky, A. Schirrer, A. Stribersky*, Vibration damping of a flexible car body structure using piezo-stack actuators, Proceedings of the 17th World Congress The International Federation of Automatic Control, Seoul, Korea, 2008, pp. 8284-8292.

[13]   *A. Schirrer, M. Kozek*, Co-simulation as effective method for flexible structure vibration control design validation and optimization, 16th Mediterranean Conference on Control and Automation Congress Centre, Ajaccio, France, June 25-27, 2008, pp. 481-486.

[14]   *A. Orvanäs*, Active secondary suspension in train. A literature survey of concepts and previous work, KTH Engineering Sciences Department of Aeronautical and Vehicle Engineering, Division of Rail Vehicles, Sweden, 2008.

[15]   *M. Dumitriu, C. Cruceanu*, Influences of carbody vertical flexibility on ride comfort of railway vehicles, Archive of Mechanical Engineering, **vol. LXIV**, issue 2, 2017, pp. 119-238

[16]   *M. Dumitriu,* Evaluation of the comfort index in railway vehicles depending on the vertical suspension, Annals of Faculty Engineering Hunedoara – International Journal of Engineering, **tome XI**, fascicule 4, 2013, pp. 23-32.

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

[18]  *M. Dumitriu,* Ride comfort enhancement in railway vehicle by the reduction of the carbody structural flexural vibration, IOP Conference Series: Materials Science and Engineering, **vol. 227**, 2017, Article number 012042.

[19]  *J. Zhou, R. Goodall, L. Ren, H. Zhang*, Influences of car body vertical flexibility on ride quality of passenger railway vehicles, Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit, **vol. 223**, 2009, pp. 461-471.

[20]  *X. Wei, Y. Liu, Q. Sun, L. Jia, Wang Y.*, Rail vehicle ride comfort prediction based on bogie acceleration measurements, 25th Chinese Control and Decision Conference (CCDC), 2013, pp. 3810 - 3815.

[21]  *D. Gong, J.S. Zhou, W.J. Sun*, On the resonant vibration of a flexible railway car body and its suppression with a dynamic vibration absorber, Journal of Vibration and Control, **vol. 19**, issue 5, 2013, pp. 649–657.

[22]  *I. Sebeşan, M. Dumitriu*, Validation of the theoretical model for the study of dynamic behavior on vertical direction for railway vehicles, Annals of Faculty Engineering Hunedoara – International Journal of Engineering, **tome XII**, fascicule 2, 2014, pp. 153-160.

[23]  *M. Dumitriu,* On the critical points of vertical vibration in a railway vehicle, Archive of Mechanical Engineering, **vol. LXI**, issue 4, 2014, pp. 115-140.

[24]  *D. I. Stănică, M. Dumitriu*, Critical points numerical analysis of ride comfort of the flexible railway carbody, IOP Conf. Series: Materials Science and Engineering, **vol. 682**, 2019, Article number 012004.

[25]  ENV 12299, Railway applications ride comfort for passengers measurement and evaluation, 1997.

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

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