Critical points numerical analysis of ride comfort of the flexible railway carbody

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Abstract. Ride comfort is one of the criteria for evaluating the dynamic behaviour in railway vehicles, through which the complex sensation triggered by the vibrations in the railway vehicle carbody upon passengers is being described. The behaviour of vibrations along the vehicle carbody is not uniform and the point where the ride comfort is the least convenient can be considered as the carbody critical point. The paper examines the ride comfort along the carbody and the position of the ride comfort critical points in correlation with the speed, carbody flexibility and suspension damping. To this purpose, there are used the results from numerical simulations regarding the ride comfort index calculated along the carbody and the ride comfort index in three points deemed relevant in terms of evaluating the ride comfort at the center of carbody and above the bogies.

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

In vertical plan, the level of vibrations is not uniform along the vehicle carbody, and the point in which the ride comfort is the least convenient can be seen as the critical point of carbody when it comes to ride comfort [11]. In general, the level of vibrations is lower at the carbody center and higher at its ends. For the flexible carbodies of the high speed vehicles, the level of vibrations at the center of carbody can increase much, due to the first carbody vertical bending mode [12 - 15].
In paper is analyzed the ride comfort along the vehicle carbody and the position of the ride comfort critical points, in correlation with the speed, carbody flexibility and suspension damping. The evaluation of the ride comfort is ruled by the ride comfort index for vertical vibrations, derived from numerical simulations. The vehicle is represented by a rigid-flexible coupled model, which comprises the flexible carbody and six rigid bodies depicting the two bogies and four axles. To establish the carbody critical points, the ride comfort index is calculated in three points deemed relevant in terms of ride comfort evaluation. One of these points is at the carbody center and the other two above the bogies [16, 17].

2. Railway vehicle model and motion equations
The model of the vehicle in figure 1 is a rigid-flexible coupled model, including the flexible carbody, bogies and four axles (figure 1) [11, 19]. The bogies and axles are treated as rigid bodies, since they feature reduced elastic deformations. The vehicle carbody is represented by a free-free equivalent beam, a constant section and uniformly distributed mass, of Euler-Bernoulli type. The parameters of the vehicle model are presented in table 1.

The vehicle is running at constant velocity $V$ on a perfectly rigid track with vertical irregularities. The vertical displacements of the axles $\eta_{1..4}$ are equal with the track vertical irregularities.

The carbory vibrations modes relevant for the ride comfort in a vertical plan are considered: vibration rigid modes - bounce $z_c$, pitch $\theta_c$, and the first bending mode. The vibration rigid modes of the bogies in a vertical plan are bounce $z_{b1,2}$ and pitch $\theta_{b1,2}$.

The vehicle suspension stages are modelled via Kelvin-Voigt type systems. The secondary suspension related to a bogie has three such systems where two are for translation (vertical and longitudinal) and one for rotation; on the other hand, the primary suspension is modelled through one Kelvin-Voigt system for translation in the vertical direction.
Table 1. The parameters of the vehicle model.

| Symbol  | Definition                           | Symbol  | Definition                           |
|---------|-------------------------------------|---------|-------------------------------------|
| $m_c$   | carbody mass                        | $2c_{zc}$ | vertical damping of the secondary suspension* |
| $m_b$   | bogie suspended mass                | $2k_{zc}$ | vertical stiffness of the secondary suspension* |
| $J_c$   | carbody inertia moment              | $2c_{zc}$ | longitudinal damping of the secondary suspension* |
| $J_b$   | bogie inertia moment                | $2k_{zc}$ | longitudinal stiffness of the secondary suspension* |
| $L_c$   | carbody length                      | $2c_{bc}$ | angular damping of the secondary suspension* |
| $L_{bc}$ | carbody wheelbase                   | $2k_{bc}$ | angular stiffness of the secondary suspension* |
| $2a_c$  | carbody wheelbase                   | $2c_{zb}$ | vertical damping of the primary suspension** |
| $I_{1,2}$ | position of the carbody on the suspension | $2k_{zb}$ | vertical stiffness of the primary suspension** |
|         |                                     |         |                                     |

* per bogie; ** per axle

The motion equation of the carbody vehicle has the general form [11, 19]:

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

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; $F_{xci}$, $F_{xci}$ and $M_i$ stand for forces and moments, respectively, due to the secondary suspension of bogie $i$ (with $i = 1, 2$)

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

$$F_{xci} = 2c_{xc} \left( h_{c} \frac{\partial^2 w_c(l_i, t)}{\partial x \partial t} + h_{b} \dot{\theta}_{bi} \right) + 2k_{xc} \left( h_{c} \frac{\partial w_c(l_i, t)}{\partial x} + h_{b} \theta_{bi} \right);$$

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

The vertical displacement of the carbody $w_c(x, t)$ is the result of having overlapped the rigid vibrations modes - bounce and pitch, and the vertical bending [15 - 19],

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

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

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

with

$$\beta = \sqrt{\omega_c^2 \rho_c / (EI)}, \cos \beta L_c \cosh \beta L_c - 1 = 0,$$

where $\omega_c$ is the natural pulsation of the carbody bending.
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 [11, 19],

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

(8)

\[
J_c \ddot{\theta}_c + 2c_{zc} a_c \dot{\theta}_c - (\dot{z}_{b1} - \dot{z}_{b2})] + 2k_{zc} a_c [2a_c \dot{\theta}_c - (\theta_{b1} - \theta_{b2})] + 2c_{xc} h_c [2h_c \dot{\theta}_c + h_c (\dot{\theta}_{b1} + \dot{\theta}_{b2})] + 2k_{xc} h_c [2h_c \dot{\theta}_c + h_c (\theta_{b1} + \theta_{b2})] +
\]

\[
+ 2c_{dc} (\ddot{\theta}_c - (\dot{\theta}_{b1} + \dot{\theta}_{b2})] + 2k_{dc} (\ddot{\theta}_c - (\theta_{b1} + \theta_{b2})] = 0 ;
\]

(9)

\[
m_{mc} \ddot{z}_c + c_{mc} \dot{z}_c + k_{mc} T_c + 2c_{zc} \dot{z}_c + 2 \varepsilon \dot{T}_c - (\dot{z}_{b1} - \dot{z}_{b2})] + 2k_{zc} \dot{z}_c [2z_c + 2 \varepsilon T_c - (z_{b1} + z_{b2})] +
\]

\[
+ 2c_{xc} h_c [2h_c \dot{\theta}_c + h_c (\dot{\theta}_{b1} + \dot{\theta}_{b2})] + 2k_{xc} h_c [2h_c \dot{\theta}_c + h_c (\theta_{b1} + \theta_{b2})] + 2c_{dc} (\ddot{\theta}_c - (\dot{\theta}_{b1} + \dot{\theta}_{b2})] + 2k_{dc} (\ddot{\theta}_c - (\theta_{b1} + \theta_{b2})] = 0 ,
\]

(10)

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

\[
k_{mc} = EI \int_0^L \left( \frac{d^2 X_c}{dx^2} \right)^2 dx ; \quad c_{mc} = \mu I \int_0^L \left( \frac{d^2 X_c}{dx^2} \right)^2 dx ; \quad m_{mc} = \rho_c \int_0^L X_c^2 dx .
\]

(11)

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

\[
X_c(l_1) = X_c(l_2) = \varepsilon ;
\]

\[
\frac{dX_c(l_1)}{dx} = - \frac{dX_c(l_2)}{dx} = \lambda .
\]

(12)

(13)

The equations describing the bounce and pitch motions of the bogies are

\[
m_b \ddot{z}_b1 = \sum_{i=1}^{2} F_{zbi} - F_{x1} ; \quad m_b \ddot{z}_b2 = \sum_{i=3}^{4} F_{zbi} - F_{x2} ,
\]

(14)

\[
J_b \ddot{\theta}_b1 = a_b \sum_{i=1}^{2} \pm F_{zbi} - h_b F_{x1} ; \quad J_b \ddot{\theta}_b2 = a_b \sum_{i=3}^{4} \pm F_{zbi} - h_b F_{x2} ,
\]

(15)

where \( F_{zbi} \), \( F_{xbi} \) stands for the forces due to the primary suspension, given in the relations below

\[
F_{zbi,2} = -2c_{zb} (z_{b1} + a_b \dot{\theta}_{b1} - \dot{\eta}_{1,2}) - 2k_{zb} (z_{b1} + a_b \dot{\theta}_{b1} - \eta_{1,2}) ;
\]

(16)

\[
F_{zbi,4} = -2c_{zb} (z_{b2} - a_b \dot{\theta}_{b2} - \dot{\eta}_{3,4}) - 2k_{zb} (z_{b2} - a_b \dot{\theta}_{b2} - \eta_{3,4}) ;
\]

(17)

The motion equations of the bogies write as such [14, 19]:

\[
m_b \ddot{z}_b1 + 2c_{zb} [2 \dot{z}_b1 - (\dot{\eta}_{1} + \dot{\eta}_{2})] + 2k_{zb} [2z_{b1} - (\eta_{1} + \eta_{2})] +
\]

\[
+ 2c_{xe} (z_{b1} - z_c - a_e \dot{\theta}_c + \varepsilon \dot{T}_c) + 2k_{xe} (z_{b1} - z_c - a_e \dot{\theta}_c + \varepsilon \dot{T}_c) = 0 ;
\]

(18)

\[
J_c \ddot{\theta}_c + 2c_{zc} a_c [2a_c \dot{\theta}_c - (\eta_{1} - \eta_{2})] + 2k_{zc} a_c [2a_c \dot{\theta}_c - (\eta_{1} - \eta_{2})] +
\]

\[
+ 2c_{xc} h_c \dot{h}_c (\dot{\theta}_{b1} + \dot{h}_c (\dot{\theta}_c + \varepsilon \dot{T}_c)) + 2k_{xc} h_c \dot{h}_c (\dot{\theta}_{b1} + \dot{h}_c (\dot{\theta}_c + \varepsilon \dot{T}_c)) = 0
\]

(19)
The motion equations for the carbody and bogies (8-10 and 18-21) make up a 7-equation system with ordinary derivatives, which can be solved numerically using MATLAB code. The system can be matrix-like written

\[
\mathbf{M}\ddot{\mathbf{p}} + \mathbf{C}\dot{\mathbf{p}} + \mathbf{K}\mathbf{p} = \mathbf{P}\eta + \mathbf{R}\eta,
\]

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

### 3. Calculation of the ride comfort index

To evaluate the ride comfort in the vertical vibrations, the partial ride comfort index is calculated [16]

\[
N_{MV} = 6aW_{ab},
\]

where

\[
a(x) = \sqrt{\frac{1}{\pi} \int_0^\infty G_{w}(x, \omega) d\omega},
\]

is the root mean square of the vertical acceleration and \(W_{ab} = W_aW_b\) represents the weight filter of the accelerations [16, 17]. The filter \(W_a\) is a pass band filter and the weighting filter \(W_b\) takes into account the high human sensitivity to the vertical vibrations. The root mean square of the vertical acceleration is calculated according to the power spectral density of the carbody acceleration

\[
G_{w}(x, \omega) = G(\omega)\omega^2\left[\Pi_{z_c}(\omega) + \left(\frac{L_c}{2} - x\right)\Pi_{\theta_c}(\omega) + X_c(x)\Pi_{T_c}(\omega)\right]^2,
\]

where \(\Pi_{z_c}(\omega), \Pi_{\theta_c}(\omega), \Pi_{T_c}(\omega)\) are the frequency response functions corresponding to the bounce and pitch and to the vertical bending of the carbody, and \(G(\omega)\) is the power spectral density of the vertical track irregularities,

\[
G(\omega) = \frac{A\Omega^2V^3}{(\omega^2 + (V\Omega_c)^2)(\omega^2 + (V\Omega_p)^2)},
\]

where \(\Omega_c = 0.8246\) rad/m, \(\Omega_p = 0.0206\) rad/m, and \(A = 4.032 \times 10^{-7}\) radm - for a high level quality track, \(A = 1.080 \times 10^{-6}\) radm - for a low level quality track [18].

When adopting the hypothesis that the vertical accelerations have a Gaussian distribution with the null mean value, we will have the following relation to calculate the partial ride comfort index [19]

\[
N_{MV}(x) = 6\Phi^{-1}(0.95)\sqrt{\frac{1}{\pi} \int_0^\infty G_{w}(x, \omega)H_{ab}(\omega)^2 d\omega},
\]

where \(\Phi^{-1}(0.95)\) represents the quantile of the standard Gaussian distribution with the probability of 95%, and \(H_{ab}(\omega) = H_a(\omega)H_b(\omega)\), where \(H_a(\omega)\) and \(H_b(\omega)\) are the transfer functions corresponding to the filters \(W_a\) and \(W_b\) [16, 17].
The position of the critical points in terms of the ride comfort, in correlation with the root mean square of vertical acceleration on of the critical points of the ride comfort, three important carbody points of the ride comfort index has an asymmetrical distribution along the carbody versus its center.

The results of the numerical simulations

This section features the numerical simulations results relating to the ride comfort index along the carbody and the position of the critical points in terms of the ride comfort, in correlation with the speed, carbody flexibility and suspension damping. The numerical simulation applications are developed in MATLAB numerical computing environment.

To examine the position of the critical points of the ride comfort, three important carbody points are taken into account, from the viewpoint of ride comfort evaluation – one at the carbody center and two points above the bogies.

The damping ratios corresponding to the two levels of suspension will be brought in, as below

\[
\zeta_c = \frac{4c_{zc}}{2\sqrt{4k_{zc}m_c}}; \quad \zeta_b = \frac{4c_{zb}}{2\sqrt{4k_{zb}m_b}}. \tag{30}
\]

The numerical simulation parameters (reference parameters of vehicle) are presented in table 2. According to these parameters, the natural frequencies of the carbody bending \( f_b \) is 8 Hz and the damping ratios of suspension are \( \zeta_c = 0.12 \) – for the secondary suspension and \( \zeta_b = 0.22 \) – for the primary suspension.

| Parameter       | Value                      |
|-----------------|----------------------------|
| \( m_c \)       | 34.0·10^3 kg               |
| \( L_c \)       | 26 m                       |
| \( k_{zc} \)    | 0.60 MN/m                  |
| \( m_b \)       | 3.20·10^3 kg               |
| \( h_c \)       | 1.30 m                     |
| \( k_{zc} \)    | 2.00MN/m                   |
| \( m_{mc} \)    | 35.2·10^3 kg               |
| \( h_b \)       | 0.20 m                     |
| \( k_{zc} \)    | 128 kN/m                   |
| \( k_{mc} \)    | 89.0 MN/m                  |
| \( a_c \)       | 19.0 m                     |
| \( c_{zc} \)    | 17.15 kNs/m                |
| \( a_{mc} \)    | 53.1 kNm/s                 |
| \( c_{zb} \)    | 2.56 m                     |
| \( J_s \)       | 1.96·10^6 kg·m^2           |
| \( I_1 \)       | 3.5 m                      |
| \( c_{zb} \)    | 128 kN/m                   |
| \( J_b \)       | 2.05·10^3 kg·m^2           |
| \( l_2 \)       | 22.5 m                     |
| \( c_{zb} \)    | 1000 Hz, \( f_2 = 100 \) Hz |
| \( J_b \)       | 2.05·10^3 kg·m^2           |
| \( l_2 \)       | 22.5 m                     |
| \( c_{zb} \)    | 1.10 MN/m                  |
| \( EI \)        | 3.02·10^9Nm^2              |
| \( A \)         | 1.080·10^{-6} radm         |

Figure 2 shows the root mean square (rms) of vertical acceleration along the vehicle carbody for more velocities. First, the level of carbody vibrations is noticed to increase with the velocity. Second, the behaviour of vibrations along the carbody is not symmetrical compared to its center. In general, the root mean square of vertical acceleration is lower at the center and raises at the carbody ends.

Figure 3 shows that the ride comfort index increases with the velocity, which means a worse ride comfort. For velocities up to 200 km/h, the ride comfort index is seen to be lower at the carbody center and higher towards its ends. Should velocity goes up (\( V = 250 \) km/h or \( V = 300 \) km/h), a significant increase of the ride comfort index occurs at the carbody center. Also, the ride comfort index has an asymmetrical distribution along the carbody versus its center.
In diagram (b), the critical points of the ride comfort are located above one of the two bogies, and their position changes upon velocity. At velocity of 100 km/h and 150 km/h, the critical point of the ride comfort is placed above the front bogie, while this point moves above the rear bogie for higher velocities. It is important to notice that the ride comfort index at the carbody center is higher than this index above the front bogie, for velocities of 250 km/h and 300 km/h.

![Figure 2. Root mean square of vertical acceleration along the carbody.](image)

![Figure 3. Ride comfort index – influence of the velocity: (a) along the carbody; (b) at the carbody center and above the bogies.](image)

The ride comfort is strongly influenced by the carbody flexibility, as shown in figure 4. A higher bending modulus means a lower comfort index along the entire carbody. A decrease in the comfort index is more visible at the carbody center. Nevertheless, significant improvement of the ride comfort is not acquired beyond a certain value of the bending modulus \((EI = 4.71 \times 10^9 \text{ Nm}^2)\). For all the cases analyzed, the critical point of the ride comfort is found above the rear bogie (see diagram (b)).
Figure 4. Ride comfort index of 250 km/h – influence of the carbody bending modulus: (a) along the carbody at velocity; (b) at the carbody center and above the bogies.

Figure 5. Ride comfort index of 250 km/h – influence of the damping ratio of the secondary suspension: (a) along the carbody at velocity; (b) at the carbody center and above the bogies.
Figure 5 shows the influence that the damping ratio of the secondary suspension has upon the ride comfort index. The increase of the damping ratio of the secondary suspension has contrary effects upon the ride comfort. While the ride comfort becomes better at the carbody ends this it degrades considerably at the carbody center. Such observations are much clearer in diagram (b). The same diagram underlies the determination of the critical points in the ride comfort. For small values of the damping ratio of the secondary suspension ($\zeta < 0.1$), the critical point in the ride comfort is noted to be above the rear bogie. When the damping ratio of the secondary suspension raises, the critical point in the ride comfort moves to the carbody center.

![Diagram of ride comfort index influenced by secondary suspension damping ratio.](image)

**Figure 6.** Ride comfort index of 250 km/h – influence on the damping ratio of the primary suspension: 
(a) along the carbody at velocity; (b) at the carbody center and above the bogies.

The influence of the damping ratio of the primary suspension upon the ride comfort index is shown in the diagrams of figure 6. An increased damping ratio of the primary suspension results into a better ride comfort along the entire carbody, yet the most important results are derived at the carbody center. The damping ratio of the primary suspension also affects the position of the critical points in the ride comfort, as seen in diagram (b). For $\zeta_p \leq 0.1$, the critical point in the ride comfort is at the carbody center; this point moves above the rear bogie when the damping ratio of the primary suspension has higher values.

5. **Conclusions**

The paper herein submits an analysis of the ride comfort at vertical vibrations along the railway vehicle carbody and the position of the critical points in the ride comfort. To this end, the results to the ride comfort index derived from the numerical simulations are used. Numerical simulations are developed by a rigid-flexible coupled model type of vehicle, where the carbody vibration modes relevant for the ride comfort in a vertical plan are taken into account - bounce, pitch and the first bending modulus.

The observations made have pointed out at the fact that the ride comfort index rises along with velocity, plus the asymmetrical distribution of the ride comfort index reported to the carbody center. The ride comfort index is generally lower at the carbody center and rises towards its ends. Under
certain circumstances – flexible carbody, high velocities, high damping ratio of the secondary suspension or low damping ratio of the primary suspension, the ride comfort index rises a great deal at the carbody center.

Considering three important points of the carbody from the ride comfort evaluation perspective – a point located at the carbody center and two points above the bogies, it turned out that any of these points can represent the critical point of the carbody in terms of ride comfort. The position of the critical point of the ride comfort changes depending on the vehicle velocity, carbody flexibility and the suspension damping. The points above the bogies are normally critical points of the ride comfort. However, the critical point in the ride comfort is placed at the carbody center, for a high damping ratio of the secondary suspension or a low damping ratio of the primary suspension.

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