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Geometric Filtering Effect of Vertical Vibrations in Railway Vehicles

The paper herein examines the geometric filtering effect coming from the axle base of a railway vehicle upon the vertical vibrations behavior, due to the random irregularities of the track. For this purpose, the complete model of a two-level suspension and flexible carbody vehicle has been taken into account. Following the modal analysis, the movement equations have been treated in an original manner and brought to a structure that points out at the symmetrical and anti-symmetrical decoupled movements of vehicle and their excitation modes. There has been shown that the geometric filtering has a selective behavior in decreasing the level of vibrations, and its contribution is affected by the axle base magnitude, rolling speed and frequency range.

Keywords*: railway vehicle, vertical vibrations, geometric filtering*

1. Introduction

The evaluation of the vibrations behavior in a railway vehicle is one of the issue being looked at even since the designing stage. The lowering of vibrations to an acceptable level in terms of running behaviour, safety, passengers comfort and track fatigue is required by regulations at European level for vehicle homologation and their admission into traffic [1, 2].

The vibrations of railway vehicles mainly come from the interaction with the track. Crosing over a track with irregularities or deviations from the ideal geometry results into vibrations of vehicles, which develop both vertically and horizontally; the two types of vibrations are decoupled, though, due to the construction symmetries (inertial, elastic and geometric) [3]. As for the vertical vibrations, the bounce and pitch vibrations are causal elements for the vehicle dynamic behavior. They can be studied on simple models, one or two degrees of freedom, based on the hypothesis of excitation symmetry, by considering only one mobile base as if the wheelsets had identical moves. This is the reason why the results with these models can be overestimated.

In fact, the vertical wheelset moves are dephased during vehicle rolling on a track with configuration irregularities in dependence with the distance between wheelsets and velocity. The displacement between the wheelsets moves brings in filtering effects of the vertical vibrations [4].

The studies on the vertical vibrations in railway vehicles have proven, with the geometric filtering effect in equation, that an explanation can be found as why the variation of the vertical acceleration with the velocity is not monotonously ascending [4]. Likewise, the geometric filtering effect has been shown to influence the emergence of the vehicle resonant behavior and, thus, it can modify the percentage of influence of various resonance frequencies upon the vehicle vibration behavior [5, 6, 7]. In terms of the issues related to the evaluation of the running behavior based on vertical acceleration or concerning the optimisation of railway vehicles suspension, it has to be considered that the geometric filtering effect has a differentiated contribution along the carbody, in dependence with the rolling behavior [8].

This paper deals, for the first time, with the influence of vehicle wheel bases, along with the running behavior, upon the characteristic of geometric filtering. This influence will be further reflected in the vehicle'e response to the crossing over the rolling track random irregularities and in the magnitude of the vertical accelerations. This is the reason why the complet model of a passenger vehicle has been accounted for, including the carbody bending vibrations. The movement equations have been treated in an original manner and brought to a structure that points out at the symmetrical and anti-symmetrical decoupled movements of vehicle and their excitation modes.

2. The mechanical model and the movement equations

The case of a two-floor suspension railway vehicle that travels on a constant speed V on a track with random irregularities. The mechanical model of the vehicles is shown in figure 1.

Just by considering that the rolling track rigidity is much higher that the one of the suspension floor, and the frequencies of wheelsets on the track are higher than the vehicle, then the hypothesis of rigid track is adopted. Therefore, the vertical wheelset travelling are equal to the corresponding irregularities.

The vehicle carbody of a length L is modelled via an Euler-Bernoulli beam of constant section and uniformly distributed mass, with the bending module EI and mass on length unit m. The carbody structural damping is considered, by the damping coefficient μ. The travelling of a beam section reported to the mobile referential *Oxz,* atached to the rear end of carbody is $w(x,t)$, where t is time. To make the analysis simpler, this model has been adopted that will allow to consider only the vibration modes coming from the carbody bending. In order to study the influence of the complex carbody vibration modes while looking at the constructive

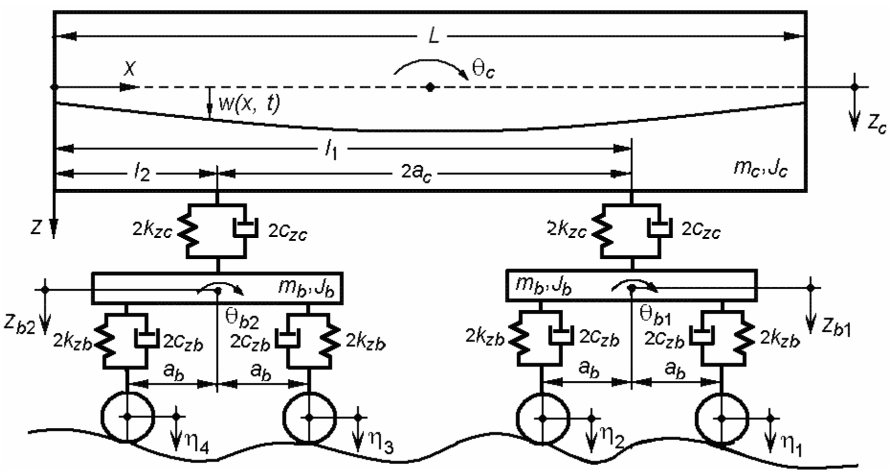

Figure 1. The vehicle mechanical model.

specific features of a certain vehicle, only models based on the finite element method are being used.

The bogies suspended masses are considered two-degree freedom rigid bodies, namely the bounce movement z_{bi} and pitch θ_{bi}, with i = 1, 2. The mass of a bogie is m_b and its inertia moment $J_b = m_b i_b^2$, with i_b – the bogie gyration radius. The suspension levels are modelled via Kelvin-Voigt systems. The elastic constants are k_{zb}, k_{zc}, and the damping ones c_{zb}, c_{zc}.

The movement equations for bending the carbody, bounce of bogies and pitch of bogies are:

$$EI\frac{\partial^4 w(x,t)}{\partial x^4} + \mu I\frac{\partial^5 w(x,t)}{\partial x^4 \partial t} + m\frac{\partial^2 w(x,t)}{\partial t^2} = \sum_{i=1}^{2} F_i \delta(x - l_i)\,, \tag{1}$$

where δ(.) is Dirac's delta function, and F_i represents the force due to the i secondary bogie suspension

$$F_i = -2c_{zc}\left(\frac{\partial w(l_i,t)}{\partial t} - \dot{z}_{bi}\right) - 2k_{zc}\left(w(l_i,t) - z_{bi}\right); \tag{2}$$

that acts at the distance l_i from the carbody end;

$$\begin{aligned} &m_b\ddot{z}_{b1,2} + 2c_{zb}(2\dot{z}_{b1,2} - \dot{\eta}_{1,3} - \dot{\eta}_{2,4}) + 2k_{zb}(2z_{b1,2} - \eta_{1,3} - \eta_{2,4}) + \\ &+ 2c_{zc}\left[\dot{z}_{b1,2} - \frac{\partial w(l_{1,2},t)}{\partial t}\right] + 2k_{zc}[z_{b1,2} - w(l_{1,2},t)] = 0 \end{aligned} \tag{3}$$

$$I_b\ddot{\theta}_{b1,2} + 2c_{zb}a_b(2a_b\dot{\theta}_{b1,2} - \dot{\eta}_{1,3} + \dot{\eta}_{2,4}) + 2k_{zb}a_b(2a_b\theta_{b1,2} - \eta_{1,3} + \eta_{2,4}) = 0\,, \tag{4}$$

where a_b, a_c are the axle bases of bogies and carbody and η_i with i = 1. 4 the irregularities against the wheelset i.

It may be noticed that the pitch movements of bogies are decoupled from the other movements of the vehicle.

The movement equations with partial derivates may be turned into equations with ordinary derivates by implementing the method of model analysis. For this, the rigid and bending modes of carbody are considered as

$$w(x,t) = z_c(t) + \left(\frac{L}{2} - x\right)\theta_c(t) + \sum_{i=2}^{\infty} X_i(x)T_i(t), \tag{5}$$

where $z_c(t)$ and $\theta_c(t)$ represent the carbody vibration rigid modes, namely the bounce and pitch. $T_i(t)$ is time, and $X_i(x)$ is the own function of vibration mode i at bending

$$X_i(x) = \sin\beta_i x + \sinh\beta_i x - \frac{\sin\beta_i L - \sinh\beta_i L}{\cos\beta_i L - \cosh\beta_i L}(\cos\beta_i x - \cosh\beta_i x), \tag{6}$$

with $$\beta_i = \sqrt{\omega_i^2 m/(EI)} \text{ and } \cos\beta_i L \cosh\beta_i L - 1 = 0, \tag{7}$$

where ω_i is the own pulsation of the vibration mode i.

Looking at two own bending modes only, symmetrical and anti-symmetrical, the vehicle vibration is described by a system of 8 ordinary derivates, among which 6 are coupled (bounce and pitch of carbody, the first and second own bending mode and the bogie bounce), and other two are independent (pitch of bogies). An appropriate choice of the coordinates and a correct processing of the equations system will give the decomposition of the system with 6 movement equations, coupled in two independent system of 3 equations each. The two systems describe the carbody symmetrical and anti-symmetrical movements. If we have,

$$\begin{gathered} q_1^+ = z_c\ ;\ \ q_1^- = \theta_c\ ;\ q_2^+ = T_2\ ;\ \ q_2^- = T_3 \\ q_3^+ = \frac{1}{2}(z_{b1} + z_{b2})\ ;\ \ q_3^- = \frac{1}{2}(z_{b1} - z_{b2}) \\ X_2(l_1) = X_2(l_2) = \varepsilon^+;\ X_3(l_1) = -X_3(l_2) = \varepsilon^- \end{gathered} \tag{8}$$

then the equations of the symmetrical movements (carbody bounce, the symmetrical bending of two-node carbody and the bogies symmetrical bounce) are obtained

$$m_c\ddot{q}_1^+ + 4c_{zc}(\dot{q}_1^+ + \varepsilon^+\dot{q}_2^+ - \dot{q}_3^+) + 4k_{zc}(q_1^+ + \varepsilon^+ q_2^+ - q_3^+) = 0\,; \tag{9}$$

$$\begin{aligned} &m_{m2}\ddot{q}_2^+ + c_{m2}\dot{q}_2^+ + k_{m2}q_2^+ + 4c_{zc}\varepsilon^+[\dot{q}_1^+ + \varepsilon^+\dot{q}_2^+ - \dot{q}_3^+] + \\ &\quad + 4k_{zc}\varepsilon^+[q_1^+ + \varepsilon^+ q_2^+ - q_3^+] = 0 \end{aligned}\ ; \tag{10}$$

$$\begin{aligned} &m_b\ddot{q}_3^+ + 4c_{zb}(\dot{q}_3^+ - \dot{\eta}^+) + 4k_{zb}(q_3^+ - \eta^+) + \\ &\quad + 4c_{zc}(\dot{q}_3^+ - \dot{q}_1^+ - \varepsilon^+\dot{q}_2^+) + 4k_{zc}(q_3^+ - q_1^+ - \varepsilon^+ q_2^+) = 0; \end{aligned} \tag{11}$$

along with the anti-symmetrical equations (carbody pitch, anti-symmetrical bending of the three-node carbody and bogies anti-symmetrical bounce)

$$J_c\ddot{q}_1^- + 4c_{zc}a_c(a_c\dot{q}_1^- - \varepsilon^-\dot{q}_2^- + \dot{q}_3^-) + 4k_{zc}a_c(a_c q_1^- - \varepsilon^- q_2^- + q_3^-) = 0\,; \tag{12}$$

$$m_{m3}\ddot{q}_2^- + c_{m3}\dot{q}_2^- + k_{m3}q_2^- + \\ + 4c_{zc}\varepsilon^-[-\dot{q}_1^- + \varepsilon^-\dot{q}_2^- - \dot{q}_3^-] + 4k_{zc}\varepsilon^-[-q_1^- + \varepsilon^- q_2^- - q_3^-] = 0; \tag{13}$$

$$m_b\ddot{q}_3^- + 4c_{zb}(\dot{q}_3^- - \dot{\eta}^-) + 4k_{zb}(q_3^- - \eta^-) + \\ + 4c_{zc}(\dot{q}_3^- - a_c\dot{q}_1^- - \varepsilon^-\dot{q}_2^-) + 4k_{zc}(q_3^- - a_c q_1^- - \varepsilon^- q_2^-) = 0; \tag{14}$$

where m_c and $J_c = m_c i_c^2$ are the carbody mass and inertia moment with i_c its gyration radius and $m_{m2,3}$, $c_{m2,3}$ and $k_{m2,3}$ are the masses, dampings and modal rigidities

$$m_{m2,3} = m\int_0^L X_{2,3}^2(x)\mathrm{d}x\,;$$

$$k_{m2,3} = EI\int_0^L \left(\frac{\mathrm{d}^2 X_{2,3}(x)}{\mathrm{d}x^2}\right)^2 \mathrm{d}x\,;\; c_{m2,3} = \mu I\int_0^L \left(\frac{\mathrm{d}^2 X_{2,3}(x)}{\mathrm{d}x^2}\right)^2 \mathrm{d}x\,. \tag{15}$$

The below equations include the excitation mode of the symmetrical and anti-symmetrical movements

$$4\eta_1^+ = \eta_1 + \eta_2 + \eta_3 + \eta_4\,;\quad 4\eta^- = \eta_1 + \eta_2 - \eta_3 - \eta_4\,. \tag{16}$$

The parameters of the carbody bending vibration are the modal pulsation and the damping degree

$$\omega_{m2,3} = \sqrt{\frac{k_{m2,3}}{m_{m2,3}}}\,,\; \zeta_{m2,3} = \frac{c_{m2,3}}{2\sqrt{k_{m2,3}m_{m2,3}}}\,. \tag{17}$$

In order to facilitate the analysis of vibrations, the damping degrees of the suspension floors considered uncoupled, as below

$$\zeta_{b,c} = \frac{4c_{b,c}}{2\sqrt{4k_{b,c}m_{b,c}}}\,. \tag{18}$$

The following section will see the movement equations being used to evaluate the geometric filtering effect due to the vehicle axle base upon the behavior of vertical vibrations.

Thus, the carbody magnification factor is calculated, taking into account the vibration harmonic behavior where the track irregularities have a sinusoidal shape with the wavelength l and the amplitude η_0.

The size of irregularities against each axle depends on its position. Thus, bearing reference to the track irregularity against the vehicle centre, the defects against the axles are dephased with $2\pi(a_c \pm a_b)/l\Lambda$- opposite the first bogie and with $2\pi(-a_c \pm a_b)/\Lambda$ opposite the second bogie, where Λ is the defect wavelength, $2a_c$ is the vehicle wheelbase and $2a_b$ represents the bogie wheelbase. The defects of the transversal nivelment are overlooked, hence the possibility of exciting the rolling movement is ruled out.

Against each wheelset, the irregularity is dephased corresponding to the vehicle axle base and to the velocity *V* of traveling over the irregularities of rolling track – all these will trigger an imposed movement for the vehicle that becomes a pulsation time function $\omega=2\pi V/\Lambda$

$$\eta_{1,2} = \eta_0 \cos\frac{\omega}{V}(Vt \pm a_b + a_c)\,,\ \eta_{3,4} = \eta_0 \cos\frac{\omega}{V}(Vt \pm a_b - a_c)\,. \qquad (19)$$

The vibrations of carbody are excited by the track irregularities against the wheelsets, and the plan of bogie axles has both a bounce and pitch movement. Since the bogies pitch movement is decoupled from the carbody's, the symmetrical and anti-symmetrical modes of carbody vibration will be determined by the symmetrical and anti-symmetrical axles bounce movements, presented in figure 2.

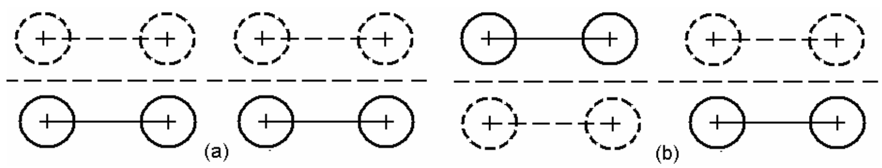

Figure 2. Excitation modes of wheel sets:
(a) symmetric bounce; (b) skew-symmetric bounce

The frequency features of the two excitation modes depend by the axle base in bogie and carbody, as well as by velocity. Their forms for the symmetrical and anti-symmetrical modes are as follows

$$\overline{H}^{+}(\omega) = \cos\frac{a_b\omega}{V}\cos\frac{a_c\omega}{V}\,; \qquad (20)$$

$$\overline{H}^{-}(\omega) = \mathrm{i}\cos\frac{a_b\omega}{V}\sin\frac{a_c\omega}{V}\,, \qquad (21)$$

where ω is the angle frequency and $\mathrm{i}^2 = -1$.

The magnification factor in a point *x* of carbody is as below

$$\overline{H}_c(x,\omega) = \overline{H}_{z_c}(\omega) + \left(\frac{L}{2} - x\right)\overline{H}_{\theta_c}(\omega) + \sum_{i=2}^{3} X_i(x)\overline{H}_{T_i}(\omega)\,, \qquad (22)$$

where $\overline{H}_{z_c}(\omega)$, $\overline{H}_{\theta_c}(\omega)$, $\overline{H}_{T_i}(\omega)$ are the magnification factors corresponding to the vibration modes z_c, θ_c si $T_{2,3}$.

To calculate the carbody vertical acceleration, the power spectrum density of the track random irregularities is as follows [9]

$$S(\Omega) = \frac{A\Omega_c^2}{(\Omega^2 + \Omega_r^2)(\Omega^2 + \Omega_c^2)}\,, \qquad (23)$$

where Ω is the wave number, $\Omega_c = 0{,}8246$ rad/m, $\Omega_r = 0{,}0206$ rad/m, and $A = 4{,}032\cdot10^{-7}$ rad m or $A = 1{,}080\cdot10^{-6}$ rad m, depending on the track quality.

The track irregularities become an excitation factor for a vehicle that travels at speed *V and this is the reason* why the spectral density of the track irregularities need to be expressed as a function of the angle frequency $\omega = V\Omega$, and $G(\omega) = [S(\omega/V)]/V$ respectively. As a result, the rel. (23) will trigger

$$G(\omega) = \frac{A\Omega_c^2 V^3}{[\omega^2 + (V\Omega_c)^2][\omega^2 + (V\Omega_r)^2]}. \tag{24}$$

Starting from the carbody frequency response $\overline{H}_c(x,\omega)$ and from the spectral density of track irregularities, the acceleration spectral density may be calculated at a point locaated at distance x from the carbody referential

$$G_c(x,\omega) = \omega^4 G(\omega) \left| \overline{H}_c(x,\omega) \right|^2. \tag{25}$$

This helps calculate the mean square deviation of acceleration in one point of carbody while the vehicle is crossing over the random irregularities in a track

$$\sigma_c(x) = \sqrt{\frac{1}{\pi}\int_0^\infty G_c(x,\omega)\mathrm{d}\omega}. \tag{26}$$

It is worthwhile mentioning that the mean square deviation is the magnitude of evaluating the rolling quality [1].

Table 1

Carbody mass	m_c = 34320 kg
Bending module	$EI = 3.2 \cdot 10^9$ Nm^2
Carbody length	L = 26.4 m
Carbody axle base	$2a_c$ = 19 m
Carbody gyration radius	i_c = 7.6 m
Carbody modal damping ratio	$\zeta_{m2,3}$ = 0.015
Vertical rigidity of secondary suspension	$4k_{zc}$ = 2.4 MN/m
Vertical damping of secondary suspension	$4c_{zc}$ = 68.88 kNs/m
Bogie mass	m_b = 3200 kg
Bogie gyration radius	i_b = 0.8 m
Bogie axle base	$2a_b$ = 2.56 m
Vertical rigidity of primary suspension	$4k_{zb}$ = 4.4 MN/m
Vertical damping of primary suspension	$4c_{zb}$ = 52.21 kNs/m

3. The analysis of the geometric filtering effect

This section presents the results of the numerical simulations, derived from the model and method above, concerning the frequency characteristics of the vehicle excitation modes, the influence of the geometric filtering upon the frequency response of a passenger vehicle carbody, as well as upon the running behavior evaluated by the mean square acceleration of the vehicle carbody vertical vibrations. The model parameters are shown in table 1.

The excitation modes will bring a series of maximum and minimum values that are conditional upon the velocity and may smoothen or not the resonance peaks of the carbody frequency response. For instance, for the frequencies $f = (2n+1)V/4a_b$ with $n = 0, 1, ...$, the wheelsets' bounce (symmetrical and anti-symmetrical) is null and, thus, the carbody bounce and pitch are not excited at these frequencies – the geometric filtering effect given by the axle base. The distance between bogies will derive an effect of geometric effect for the symmetrical bounce of wheelsets at frequencies $f = (2n+1)V/4a_c$ and for the anti-symmetrical bounce mode at frequencies $f = nV/2a_c$.

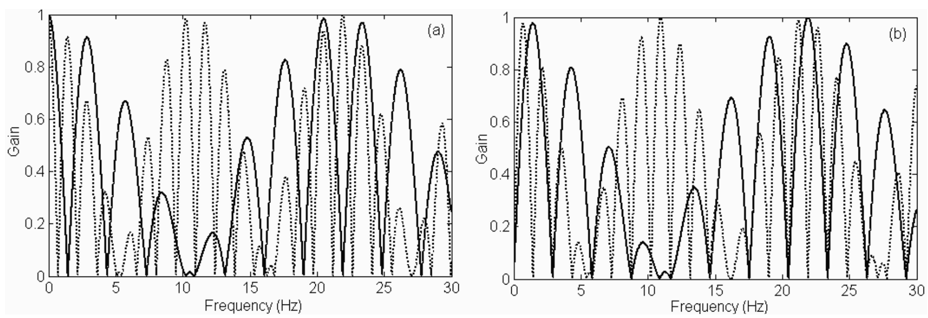

Figure 3. Influence of velocity upon the frequency characteristics of the vehicle excitation modes: (a) symmetrical mode; (b) skew-symmetrical mode: · · ·, V = 100 km/h; ——, V = 200 km/h.

Figure 3 shows the frequency feature of the two excitation modes at velocities of 100 and 200 km/h, for $2a_c$ = 19 m and $2a_b$ = 2.56 m. Since the distance between bogies is much longer than the one between the wheelsets in a bogie, the frequency features will display itself in the shape of a carrying wavelength, corresponding to the geometric filtering effect of the distance between bogies and modulated in amplitude by the effect of geometric filtering of the distance between wheelsets. The frequency of the geometric filtering effect increases along with the velocity of vehicle. For instance, the first frequency of geometric filtering due to the bogie axle base is of 5.43 Hz at 100 km/h, and the next one at 16.28 Hz. For V = 200 km/h, the interval between two consecutive minimum values is 21.7 Hz, and the first minimum occurs at 10.85 Hz. The filtering due to the carbody axle base introduces the first frequency minimum at 0.73 Hz for the symmetrical bounce mode of axle bases at 100 km/h and at 1.46 for 200 km/h. For the skew-symmetrical bounce condition, the first minimum occurs at 1.46 Hz if speed is 100 km/h and at 2.92 Hz for 200 km/h.

Should one of the geometric filtering frequencies coincides with one of the own vehicle frequencies, at a certain velocity, then the intensity of the vibration behavior lowers. Still, at high velocities, the filtering frequency of the axles bounce exceeds the range of the own vehicle frequencies and this filtering effect exerts a much smaller impact. Indeed, the frequency of geometric filtering effect of wheel-set bounce is 10.85 Hz, a value superior to the frequencies of the main natural vibration conditions of the vehicle under discussion. For this reason, it is expected to have a vibration behavior more intensified at velocities higher than 200 km/h, as seen further herein.

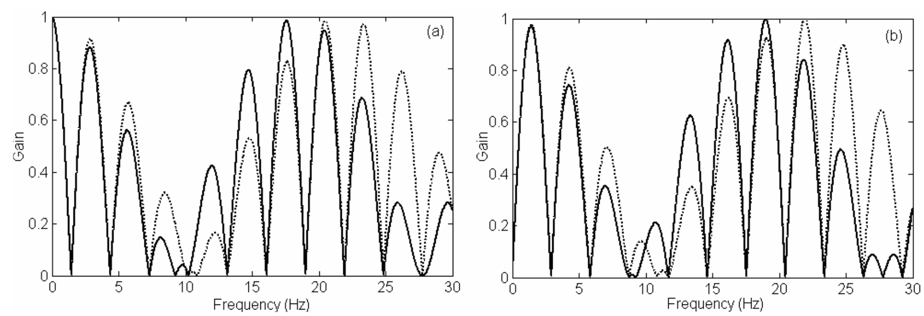

Figure 4. Influence of bogie axle upon the frequency characteristics of vehicle excitation modes: (a) symmetrical; (b) skew-symmetrical: · · · ·, $2a_b$ = 2.56 m; ——, $2a_b$ = 3 m.

The frequency of filtering effect decreases along the increase of bogie axle base – see figure 4, which features the filtering characteristics of the two excitation modes for the axle base reference value ($2a_b$ = 2.56 m) and for $2a_b$ = 3 m. The velocity is V = 200 km/h, and the reference value for the carbody axle base is presented in the table 1. The first frequency of the geometric filtering given by the 3-meter axle base lowers to 9.26 Hz, compared to the previous situation – the 2.56-meter axle base (see figure 2), where the frequency was shown to be 10.85 Hz. Another mention to be made is related to the change in the filtering lobes height for the two situations above. The closer their height is to 1, the lower the geometric filtering effect. It is evident that an increase in the bogie axle base results into a higher or a lower geometric filtering effect, depending on the frequency range. At 200 km/h, within the frequency interval of up to circa 10 Hz, which includes the frequencies of main natural vibration conditions of vehicle – the higher the bogie axle base, the higher the geometric filtering effect. For frequencies between 10 and 19 Hz, the effect derived from the increase of axle base upon the filtering effect is opposite.

Figure 5 shows the influence of the carbody axle base upon the frequency characteristics of the two excitation modes at velocity of 200 km/h. A first observation would be related to the height lowering for the filtering lobes at frequencies below circa 11 Hz, for a 17-meter carbody axle base. Such observations are general in nature, since it is evident that, for the frequency range above, a smaller carbody axle base may trigger a lower geometric filtering effect, than for a 19-meter axle base and viceversa.

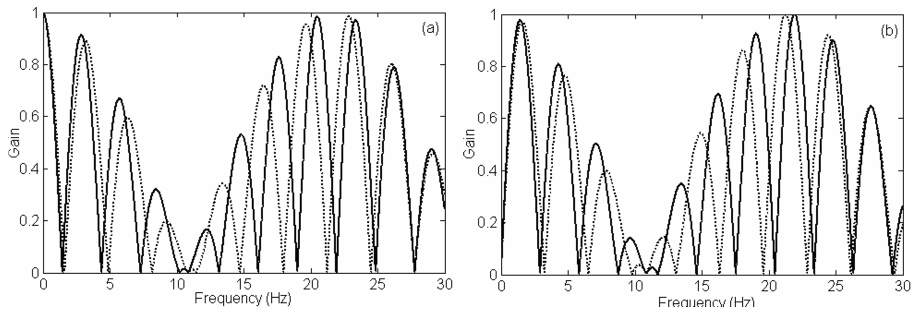

Figure 5. Influence of carbody axle base upon the frequency characteristics of the vehicle excitation modes: (a) symmetrical; (b) skew-symmetrical: · · · ·, $2a_c$ = 17 m; ——, $2a_c$ = 19 m.

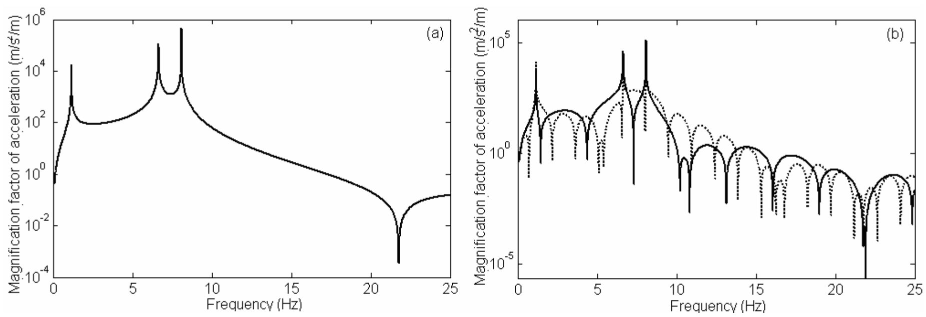

Figure 6. The magnification factors of the vertical acceleration: (a) non-filtered; (b) filtered, · · · ·, V = 100 km/h; ——, V = 200 km/h.

Figure 6 shows the frequency response of the acceleration in the center of carbody, both without the filtering effect due to the vehicle axle bases (a) and with it in the equation (b). The reference velocities: 100 and 200 km/h. In the center of carbody, the frequency response is only due to the vibration symmetrical modes, namely the symmetrical bounce and pitch. In order to point out at the system resonance frequencies, the non-damped case has been admitted, when the rest of the parameters are in Table 1. The peaks of resonance frequencies of low bounce at 1.17 Hz and of high bounce at 6.65 Hz, as well as of the symmetrical bending's (two-node) at 8.1 Hz.

The geometric filtering effect helps conclude, on the one hand, that the magnification factor is smaller and, on the other hand, the succession of maximum and minimum values derived from it. The maximum values correspond to the situation where the geometric filtering effect is minimum, which is similar in the position on the filtering characteristic against the center of a lobe.

For the frequencies of filtering effect, the vehicle response frequency has a minimum value, an anti-resonance. Should the filtering effect frequency coincides with the dominant frequency of the vehicle response, then the response to the random irregularities in the rolling track is weaker. As already shown, the filtering effect makes that the system frequency response be dependent on the velocity, which is quite natural for vehicles. It can also be noted that a visible change of the magnification factor in dependence with velocity occurs, and that the filtering effect is mainly visible at a 100 km/h speed within the range of low frequencies (as thus anticipated)

The magnitude of the vehicle axle bases, whose influence upon the frequency characteristics of the vehicle excitation modes was shown above and will therefore determine changes in the vehicle frequency response.

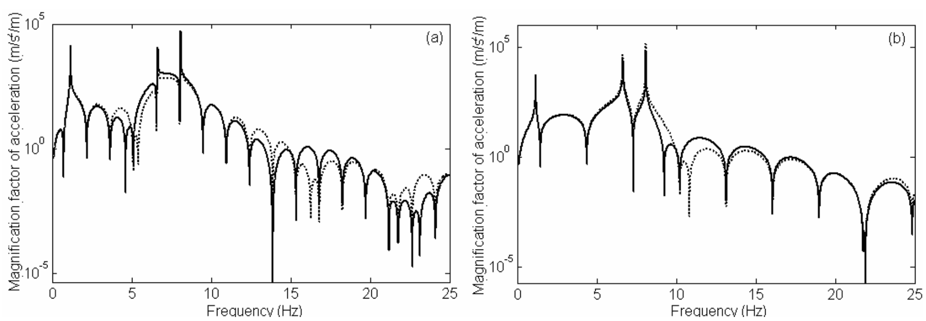

Figure 7. Influence of bogie axle base upon magnification factor of the vertical acceleration: (a) V = 100 km/h; (b) V = 200 km/h: · · · ·, $2a_b$ = 2.56 m; ——, $2a_b$ = 3 m.

Figure 7 shows the vehicle frequency response at 100 and 200 km/h, for the values of the bogie axle base included in the legend. The analysis of the results needs to be done in correlation with the observations made for figure 4. As already presented, an increase in the bogie axle base means a higher or a lower geometric filtering effect, dependent on the frequency range under study.

The observation made for V = 200 km/h and frequencies lower than 11 Hz, the higher the bogie axle, the more intense the geometric filtering effect, an evident fact in the vehicle response. Moreover, for low speeds, the frequency intervals between anti-resonants become smaller. Consequently, a change in the bogie axle base will make the filtering effect of vertical vibration be selective in nature, depending on the frequency range and velocity.

The lowering of the carbody axle base will generally result into a weaker magnification factor of the vertical acceleration, for frequencies of up to 10 Hz.

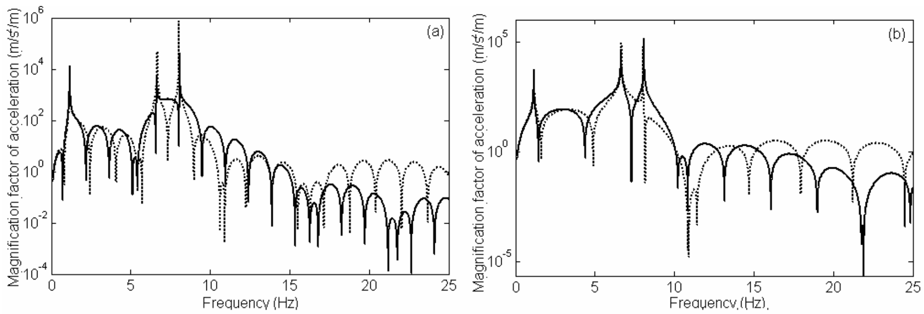

Figure 8. Influence of carbody axle base upon the magnification factor of vertical acceleration: (a) V = 100 km/h; (b) V = 200 km/h:
· · · ·, $2a_c$ = 17 m; ——, $2a_c$ = 19 m.

On the contrary, the vibration behavior will intensify at high frequencies (see figure 8). At 100 km/h, the decrease of the carbody axle base at 17 meters, compared to the value adopted as reference, will trigger the decrease of the magnification factor of the vertical acceleration at high bounce resonance frequencies and symmetrical bending – whereas at 200 km/h, the same result is only obtained at the high bounce resonance frequency. Such observations are supported, on the one hand, by making the correlation with the mentions of figure 5, where it was evident that the geometric filtering effect can be higher or lower for one or the other end between the carbody axle bases, depending on the frequency and height of the filtering lobes.

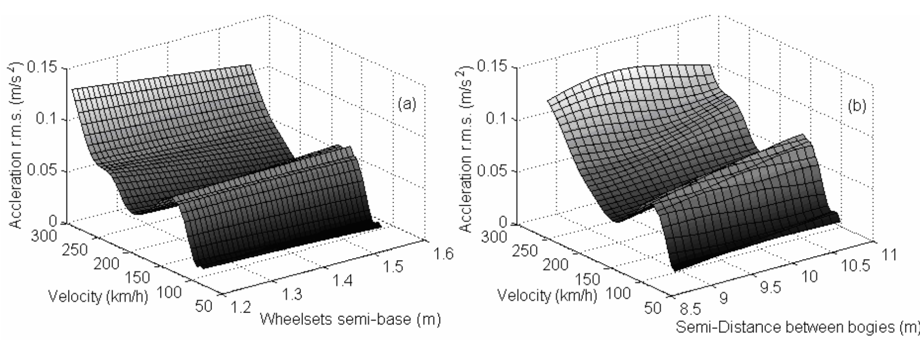

Figure 9. The r.m.s. acceleration in the center of carbody:
(a) influence of bogie axle base; (b) influence of vehicle axle base.

Figure 9 displays the r.m.s. acceleration in the middle/center of the carbody for velocities between 40 and 250 km/h. The influence of the vehicle axle base upon this magnitude is pointed out at, upon which the running behavior of a railway vehicle is evaluated. It needs to be mentioned that the acceleration shows a succession of maximum and minimum values for a speed of circa 160 km/h, due to the geometric filtering effect. At higher speeds, the filtering frequency of the wheelset bounce exceeds the range of natural vehicle frequencies and the filtering effect has a smaller influence. Starting with this speed up to the superior limit (250 km/h), acceleration rises along with velocity, the maximum and minimum values derived from the filtering effect have a lower frequency and the acceleration variations are also smaller.

In terms of the influence of the bogie axle base upon the vertical acceleration, it can be noticed that its increase will result into different effects upon the acceleration magnitude, velocity-dependent. Generally speaking, the level of the vertical vibration tends to lower – except for the speed interval of 80 – 150 km/h, where the acceleration is higher if the axle base does the same. Should the carbody axle base increases, the vertical vibration behavior intensifies, which is visible by a higher acceleration. There are exceptions, though, for small intervals of speeds, 50 – 80 km/h and 230 – 250 km/h, where acceleration shows a series of maximum and minimum values, triggered by the geometric filtering effect.

4. Conclusions

The vertical vibrations behavior during the railway vehicle rolling on a track with nivelment irregularities is atenuated by the geometric filtering effect from the vehicle axle bases. While neglecting this phenomenon in the study of the vertical vibrations, the results can be overestimated and the vehicle resonant condition incorrectly evaluated.

To make the analysis simpler, the model of a two-level suspension passenger railway vehicle has been considered – where the carbody, modelled via an Euler-Bernoulli beam, is linked by Kelvin-Voigt systems to the two (considered) rigid bogies. Following the modal analysis, the symmetrical and anti-symmetrical de-coupled movements of vehicle and their excitation modes have been highlighted.

The geometric filtering effects in the vehicle response to the random irregularities in the rolling track are analyzed in correlation with the filtering characteristics and the influence of vehicle axle base. The geometric filtering is shown being able to lower the level of vertical vibrations and the vehicle axle bases play an important role here. Nevertheless, the filtering effect has a selective nature, a function of the frequency range and velocity. At higher speeds, the filtering has a smaller influence, easily explained by the fact that the filtering frequency of wheelset bounce exceeds the interval of the vehicle natural frequencies.

The r.m.s. acceleration, which is the evaluation factor of the running behavior at a railway vehicle, has on its turn a string of maximum and minimum values in dependence of velocity, due to the geometric filtering effect. The intensification of the vibration behavior for certain velocities are influenced by the magnitude of vehicle axle base.

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