ROLLING NOISE SIMULATION OF A RAILWAY VEHICLE

Associate Professor PhD. eng. Traian **MAZILU**
Department of Railway Vehicles, University Politehnica of Bucharest
trmazilu@yahoo.com

Abstract *Rolling noise of the railway vehicles is occurred by the wheel/rail vibration excited by the rolling surfaces roughness. This paper presents an acoustic model able to predict the rolling noise level of a railway vehicle. The acoustic model is based on a wheel/rail vibration model which takes into account the structural wheel vibration (Remington model) and the bending vertical waves of the rail. To this, the track model with an infinite Euler Bernoulli beam elastically supported on two layers is applied to simulate the effect of the ballasted track. The influence of the rolling surfaces roughness and vehicle velocity on the rolling noise is investigated.*

Keywords: rolling noise, rail, wheel, roughness

1. INTRODUCTION

Rolling noise and ride comfort are the two main requirements of comfort from the mechanical perspective which have to be accomplished by any modern railway passenger car [1, 2].

The rolling noise is generated by wheel/rail vibration excited at the contact between the wheel and rail by the roughness of rolling surfaces and it represents the most important source of noise from railways.

The railway rolling noise has been extensively studied since the 1970s, mainly by Remington [3, 4] and latter, by Thompson [5–7] who developed a prediction model which has been implemented into the so-called Track-Wheel Interaction Noise Software (TWINS). In this way, the most important aspects related by the rolling noise have become well understand and many solutions for reducing the noise level have been applied, such as wheel and rail dampers, new brake types etc [7].

In this paper, the rolling noise from a particular passenger car with Y 32 bogies is calculated and analyzed in order to point out the main features. The conclusions could be interesting for the rolling stock designers.

2. THE ROLLING NOISE MODEL

In figure 1 the rolling noise model is presented, considering a passenger car on bogies rolling on a ballasted track at constant velocity V. Actually, there are 8 noise sources corresponding to the wheels situated at the R – wheel radius – high above the railhead and also other 8 noise sources for the rails. The distances between the noise sources are determined by the distances between the bogie wheels, $2a$, and between bogies, $2a_v$, and also, the distance between the two rails of the track, $2e$. It supposes that the all noise sources are incoherent.

The noise is generated by the wheel/rail vibration induced by the irregularities of the rolling surfaces and the wheel/rail vibration model is presented in figure 2. The wheel model suggested by Remington and the rail model with two viscoelastic layers for the ballasted track are considered [6]. In fact, the coupling between the wheels via the vehicle suspension is neglected because the vibration (noise) frequency is much higher than the natural frequencies of the vehicle. Also, the coupling between the wheels via the bending waves of the rails is disregarded due to the attenuation effect and the long distance between the wheel/rail contact points.

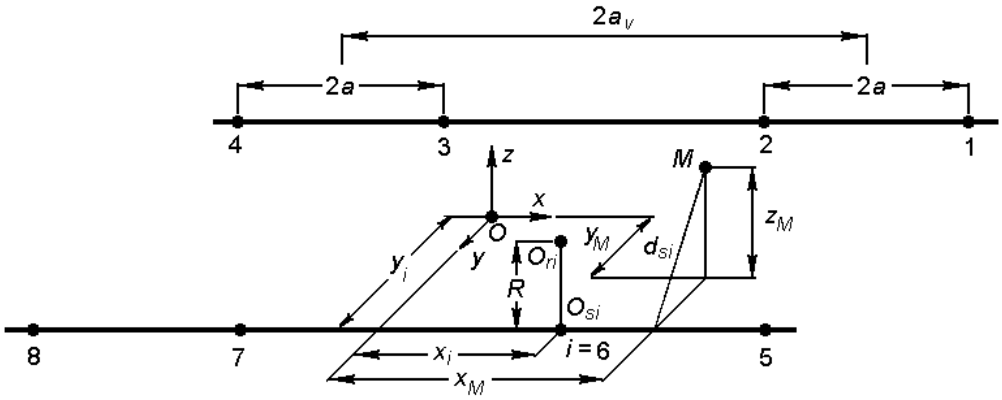

Fig. 1. Acoustic model of the vehicle.

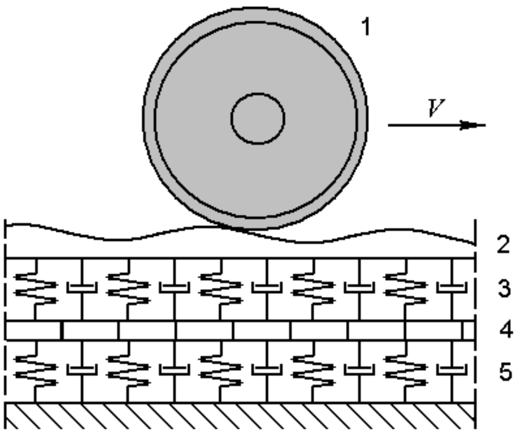

Fig. 2. Wheel/rail vibration model: 1. wheel; 2. rail; 3. rail pad; 4. sleeper; 5. ballast.

Coming back to the figure 1, it has to be mentioned that the sound pressure will be calculated in the point M of coordinates (x_M, y_M, z_M) in respect to the reference frame *Oxyz*. The wheel/rail contact point O_{ri} with $i = 1 \div 8$, has the coordinates (x_i, $y_i = \pm e$, $z_i = 0$), where

$$x_{1,5} = -x_{4,8} = a_v + a\,;\ x_{2,6} = -x_{3,7} = a_v - a\,. \qquad (1)$$

For a particular one-third octave band with the center angular frequency of ω_c, the sound pressure level in the point *M* due the rails is

$$L_r(\omega_c) = 10\lg \frac{1}{p_0^2} \sum_{i=1}^{8} \Phi_{ri}(\omega_c)\,, \tag{2}$$

where $\Phi_{ri}(\omega_c)$ is the mean spectral power of the sound pressure occurred by rail under wheel i, and $p_0 = 2 \cdot 10^{-5}$ Pa.

The mean spectral power $\Phi_{ri}(\omega_c)$ can be calculated using the relation

$$\Phi_{ri}(\omega_c) = \frac{\sigma_r}{\pi} z_0^2 \frac{a_r + a_f}{d_{ri}} \Gamma_i \Phi_{vr}(\omega_c)\,, \tag{3}$$

where σ_r is the acoustic effectiveness of the rail, z_0 – characteristic acoustic impedance of the air, $2a_{r,f}$ – railhead and respectively rail foot width, d_{ri} – the M point–rail distance, $\Phi_{vr}(\omega_c)$ – mean spectral power of the rail velocity in the contact point, Γ_i – factor witch introduces the influence of the exponential decrease of the velocity along the rail.

Mean spectral power of the rail velocity $\Phi_{vr}(\omega_c)$ is calculated taking into account the limits of the one-third octave band, i.e. ω_l and ω_h

$$\Phi_{vr}(\omega_c) = \int_{\omega_l}^{\omega_h} \frac{\omega^2}{V} |H_c(\omega)|^2 |H_{vr}(\omega)|^2 \left[G_{rr}(\omega) + G_{rw}(\omega)\right] \mathrm{d}\omega\,, \tag{4}$$

where $H_c(\omega)$ is the wheel/rail contact filter, H_{vr} – frequency-domain rail's response in terms of rail velocity in the contact point and $G_{rr,rw}$ – spectral power density of the roughness.

The sound pressure level in the point M due the wheels can be calculated using

$$L_w(\omega_c) = 10\lg \frac{1}{p_0^2} \sum_{i=1}^{8} \Phi_{wi}(\omega_c)\,, \tag{5}$$

where $\Phi_{wi}(\omega_c)$ is the mean spectral power of the sound pressure occurred by the wheel i, which is given as

$$\Phi_{wi}(\omega_c) = \frac{\sigma_w}{2} z_0^2 \frac{R^2}{d_{wi}^2} \Lambda \Phi_{vw}(\omega_c)\,, \tag{6}$$

where $\sigma_w = 2$ is the acoustic effectiveness of the wheel, $d_{wi} = MO_{wi}$, $\Phi_{vw}(\omega_c)$ – the mean spectral power of the wheel velocity in the contact point, and the Λ factor which takes into account the velocity distribution on the wheel tread.

The mean spectral power $\Phi_{vw}(\omega_c)$ can be calculated using

$$\Phi_{vw}(\omega_c) = \int_{\omega_l}^{\omega_h} \frac{\omega^2}{V} |H_c(\omega)|^2 |H_{vw}(\omega)|^2 \left[G_{rr}(\omega) + G_{rw}(\omega)\right] \mathrm{d}\omega\,, \tag{7}$$

where $H_{vw}(\omega)$ is the frequency-domain wheel response in terms of velocity at contact point.

It has to be underlined that the frequency–domain response of the wheel and the rail in terms of velocity at contact point can be calculated based on the wheel/rail vibration model. In fact this response represents the wheel or rail velocity at contact point due to a unit roughness input.

Taking into account the contact stiffness

$$k_H = \frac{3}{2}\sqrt[3]{Q_o C_H^2}\ , \tag{8}$$

where Q_o is the static load and C_H is the Hertzian constant, which depends on the rolling profiles curvatures, the frequency–domain response of the wheel and rail can be written as

$$H_{vw}(\omega) = \frac{1}{1 + \frac{\overline{Z}_w}{\overline{Z}_r} + \mathrm{i}\omega\frac{\overline{Z}_w}{k_c}}\ ,\ H_{vr}(\omega) = -\frac{1}{1 + \frac{\overline{Z}_r}{\overline{Z}_w} + \mathrm{i}\omega\frac{\overline{Z}_r}{k_c}}\ , \tag{9}$$

where Z_w and Z_r are the impedances of the wheel and rail at contact point.

The sound pressure level of the rolling noise occurred by the vehicle in a particular one-third octave band can be calculated

$$L(\omega_c) = 10\lg 10^{L_w(\omega_c)/10} + 10^{L_r(\omega_c)/10}\ . \tag{10}$$

Finally, the sound pressure level of the rolling noise is obtained

$$L_V = 10\lg\sum_{j=1}^{n_b} 10^{L(\omega_{cj})/10}\ , \tag{11}$$

where n_b are the number of one-third octave bands and ω_{cj} is the centre angular frequency of the j one-third octave band.

3. NUMERICAL APPLICATION

In this section, the numerical results derived from de model presented previous for a particular passenger car are presented and discussed. Figure 3 shows the roughness spectra from the vehicles wheels fitted with cast–iron block brakes and from those without (e.g. disk braked). The frequency corresponds to the velocity of 160 km/h. The two kinds of wheel are named braked wheel and respectively, non-braked wheel. It appears that the roughness is higher for a braked wheel due to the wear caused by the friction between the brake blocks and the wheel tread.

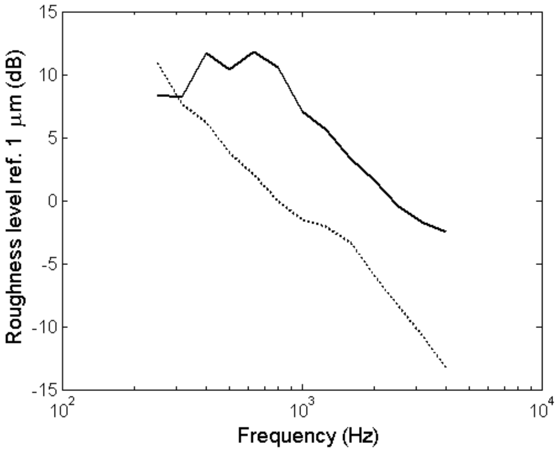

Figure 3. Roughness level of the wheel:
─, braked wheel; -----, non-braked wheel.

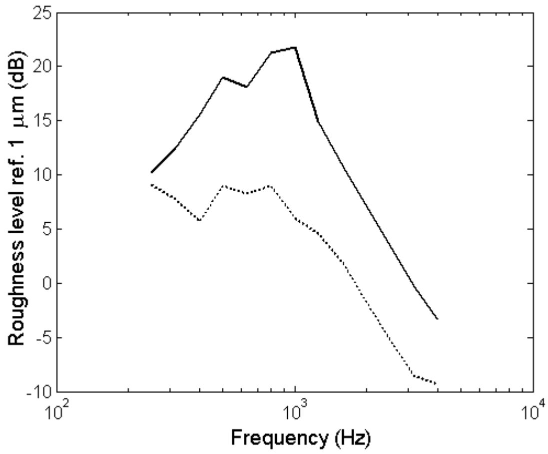

Figure 4. Roughness level of the rail;
─, corrugated rail; -----, smooth rail.

Figure 4 presents the roughness spectra from a corrugated rail and respectively from a smooth rail. General speaking, the corrugated rail is an important rail defect which can lead to an increase in the noise emission between 10 and 20 dB. The peaks appear against the particular frequencies determined by the wavelength of the corrugation and velocity. Obvious, the rolling noise has two main sources, the wheel and the rail and, to achieve its reduction, it is of paramount importance to know whether rolling noise is caused by wheel or rail.

The sound pressure level at middle of the bogie and 0.9 m from the rail head for smooth rail and non-braked wheels at 160 km/h is presented in figure 5. In fact, the sound pressure levels due to the wheels and rails are shown separately and the total noise level.

The rail noise is more intense, particular at low and middle frequencies. Indeed, the rail impedance is smaller than the wheel impedance in the range of these frequencies. The wheel impedance decreases as the frequency increases due to the resonant behavior of the wheel tread. In this way, the wheel contribution to the total noise level becomes higher.

Figure 6 presents the influence of the rolling surfaces roughness on the sound pressure level at 80 and 160 km/h. Three cases of rolling surfaces roughness are simulated: non – braked wheel and smooth rail, braked wheel and smooth rail and non-braked wheel and corrugated rail.

The most noisy roughness combination involves the corrugation of the rail and less noisy is produced when the wheel is non-braked and rail is smooth. Indeed, for instance, at 80 km/h the sound pressure level is 99.3 dB for the non-braked wheel – smooth rail, 102.5 dB for the braked wheel – smooth rail and 109.2 dB for non-braked wheel – corrugated rail. It can be seen that the sound pressure level increases with 3 dB due to the braked wheel and 10 dB due to the corrugation of the rail. These results are consistent with those presented in other papers [8].

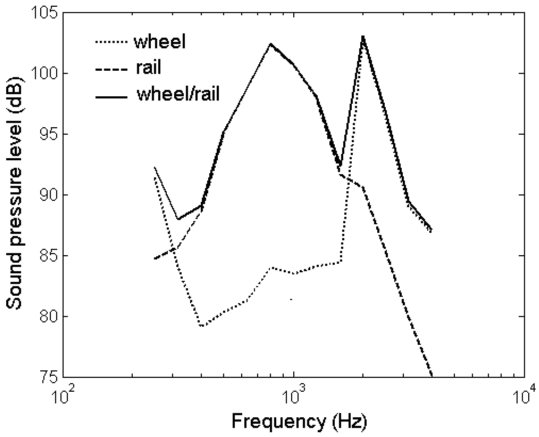

Fig. 5. Sound pressure level due to the rail and wheel.

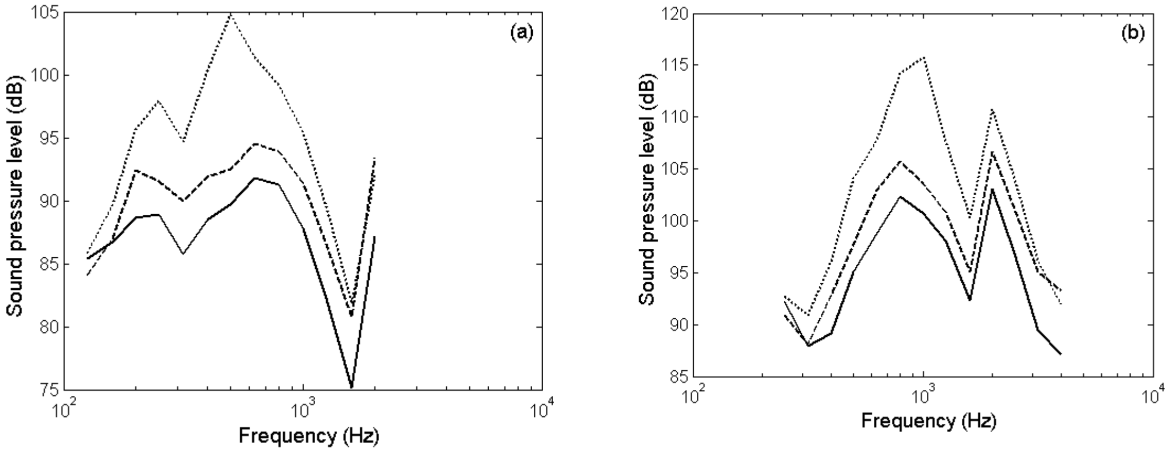

Figure 6. Influence of the roughness on the sound pressure level:
(a) at 80 km/h; (b) at 160 km/h; ─, non – braked wheel and smooth rail; -----, braked wheel and smooth rail; ▪▪▪▪, non-braked wheel and corrugated rail.

On the other hand, figure 6 also shows the influence of the velocity on the sound pressure level. Comparing the sound pressure level calculated at 80 km/h and at 160 km/h, it can be shown that the sound pressure level increases with 9.6 dB for the non-braked wheel – smooth rail, 9.8 dB for braked wheel – smooth rail and 10.6 for the non-braked wheel – corrugated rail.

These values are very close to those reported by the experiments which show that the rolling noise increases with 9 dB for a doubling of speed [8].

4. CONCLUSIONS

Numerical simulation of the rolling noise produced by a railway vehicle is a theoretical tool that can serve to investigate the wheel/rail parameters influence on the noise level. This is useful since it helps to identify the ways and the means to control the rolling noise.

Numerical simulation presented here has shown that the noise level depend strongly on the rolling surfaces roughness and vehicle velocity.

REFERENCES

[1] **Mazilu, T.,** Confortul la materialul rulant (Comfort at rolling stock), MatrixRom, Bucharest, 2003.

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

[3] **Remington, P. J.,** *Wheel/rail noise, I, characterization of the wheel/rail dynamics system*, Journal of Sound and Vibration 46 1976, pp. 381-394.

[4] **Remington, P. J**., *Wheel/rail rolling noise: what do we know? What don't we know? Where do we go from here?* Journal of Sound and Vibration, 120, 1988, pp. 203–226.

[5] **Thompson, D. J.**, *Wheel-rail noise: theoretical modeling of the generation of vibration*, Ph. D. thesis, University of Southampton 1990.

[6] **Thompson, D. J**., **Hemsworth, B., Vincent, N.,** *Experimental validation of the TWINS prediction program, Part 1: Description of the model and method*. Journal of Sound and Vibration, 193, 1996, pp. 123–136.

[7] **Thompson, D. J**., **Fodiman, P., Mahe, H.,** *Experimental validation of the TWINS prediction program for rolling noise, Part I1: Results*, Journal of Sound and Vibration, 193, 1996, pp. 123–135.

[8] **Thompson, D. J.**, *Railway Noise and Vibration: Mechanisms, Modelling and Means of Control*, Elsevier, 2009.

Editura "Academica Brâncuşi" , Târgu Jiu, ISSN 1844 – 640X