Effects of railway ballast on the sound radiation from the sleepers

Xianying Zhang, David Thompson and Giacomo Squicciarini

Institute of Sound and Vibration Research, University of Southampton, Southampton SO17 1BJ, UK

Summary

Railway ballast is an important component in the railway system. It is a layer of stones located below and around the sleepers; it supports the track vertically and provides lateral stability. The gaps between the stones make it behave acoustically as a porous material, absorbing noise to some extent. In addition, ballast can itself vibrate and reradiate noise during train pass-by. It is not clear, however, to what extent the ballast contributes to the noise and how much its mechanical and acoustical properties modify the radiation of the sleeper and the rail. This paper focuses on quantifying the influence that ballast has on noise, particularly on the sleeper radiation. A one-fifth scale model of the track is used to conduct acoustic and vibration measurements on ballast. Two different scaling factors (1:√5 and 1:5) are adopted for the stone sizes to reproduce the acoustic properties of the ballast. It is shown that although a scale factor of 1:√5 gives a better scaling of the acoustic properties, the stones scaled at 1:5 also give acceptable results. The effects of the ballast on the noise radiation from a scaled concrete sleeper are obtained experimentally with the ballast located either on a rigid foundation or on a flexible wooden base. The ballast vibration in these configurations has been obtained by using a scanning laser vibrometer. Finally, the effects of ballast vibration on the sleeper radiation are evaluated by means of a prediction based on the Rayleigh integral.

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1. Introduction

In most situations for conventional speeds rolling noise is the dominant source of noise from the railway system. It is radiated by the wheels, the rails and, at low frequencies, the sleepers. Although the TWINS model [1] is widely used and has been extensively validated, less work has been done on the low frequency components of the noise.

The sleepers are supported in the track by the ballast, which is a layer of stones below and around the sleepers. This supports the track vertically and provides lateral stability. The ballast is usually modelled dynamically as a series of springs and dampers under sleepers [2]. Ahlbeck et al. [3] conducted some theoretical work on the ballast vibration, assuming that the load was transmitted within a cone region in the ballast. Zhai et al. [4] proposed a five-parameter ballast vibration model based on the hypothesis that the transmission of the load from the sleepers to the ballast approximately coincides with a cone distribution.

From an acoustical point of view, the gaps between the stones make it behave as a porous material, absorbing noise to some extent. As early as 1940, the absorption coefficient of ballast was measured in a reverberation chamber by Kaye et al. [5]. More recently acoustic measurements were made of flow resistivity, stone density and porosity of ballast by Attenborough et al. [6]. The ballast was considered as an extended reaction medium by Heutschi [7]. The acoustic properties of railway ballast were also investigated recently by Broadbent et al. [8] in terms of absorption coefficient and excess attenuation at the receiver.

Therefore, during a train pass-by, the ballast can vibrate, leading to reradiated noise, while also absorbing a certain amount of the incident sound. It is not clear, however, to what extent these two effects contribute to the noise radiated to the side of the track and how much the mechanical and acoustical properties of the ballast modify the radiation from the sleeper and the rail.

This paper presents the results of measurements performed on a scale model of the track focusing on the influence of the ballast absorption and vibration on the sound radiation from the sleeper. The geometrical scale of the model is chosen as 1:5. Scale model concrete sleepers have been constructed and reduced scale ballast with a suitable range of stone sizes has been obtained by sieving granite ballast with appropriately sized sieves. However a correct acoustic scaling of a porous material requires that interstitial spaces are scaled according to 1:√5 while the overall geometry should be scaled by 1:5 [9]. In contrast, the overall mechanical properties of the ballast would be correctly reproduced by maintaining the 1:5 scale factor for the sieved ballast. To assess the effect of the scaling of the material, the absorption coefficient for both 1:√5 and 1:5 stone sizes is considered in Section 2.

The effects of the ballast on the sound radiation from the sleeper are quantified by measurements in terms of the sleeper radiation ratio, again obtained on the 1:5 scale model. The case of a single sleeper embedded in ballast is considered (Section 3). The ballast vibration per unit force is also measured in this arrangement by means of a scanning laser vibrometer (Section 4). Finally, the effects of the ballast vibration on the sleeper radiation are evaluated in Section 5.

2. Measurements of ballast absorption

The gaps between the ballast particles mean that the ballast can be viewed as a porous material, which is acoustically absorbing to some extent. The ballast absorption will have an influence on the rail and sleeper radiation which needs to be quantified.

According to the scaling law presented in [9], although the frequency scales properly with the sample thickness, it is the 'number of interstices' per area rather than the pore diameter that provides the correct scaling of the absorption. This scaling law would suggest that tests should be performed with stone dimensions scaled at 1:√5 while the thickness is scaled by 1:5.

In order to obtain the ballast absorption over a broad frequency range, the reverberation chamber method is used. Although the measurement standard [10] requires the volume of the room to be at least 150 m^3, due to the small area of the scale ballast used and the higher frequency range, a small reverberation chamber has been used. This has a volume of 15.6 m^3 (2.5 m × 2.4 m × 2.6 m) and a corresponding Schroeder frequency of 800 Hz for the empty chamber. The sound sources, which include a high frequency horn and a low frequency hemispherical speaker, were positioned in separate corners of the room. The results were averaged over two source and three receiver positions.

The absorption of 1:√5 scale ballast measured in the reverberation chamber is shown in Figure 1. The results are compared with the absorption measured for the corresponding thicknesses of full scale ballast from [8]. Two ballast thicknesses were tested: 30 mm and 60 mm, which correspond to thicknesses of 150 mm and 300 mm at full scale. The frequency range of the full scale measurements has been increased by a factor of 5 in this plot. Generally good agreement can be seen. The ballast absorption increases with thickness, and similar peaks and dips are found in the reduced scale and full scale results. The absorption coefficient for the thicker layers exhibits a peak at 1250 Hz, which is where the thickness is roughly equal to a quarter of the acoustic wavelength [8]. There is a corresponding dip at 2500 Hz, where the thickness is approximately equal to half the acoustic wavelength. Similarly for the thinner layers a peak occur at 2500 Hz and a dip at 5 kHz. In summary Figure 1 shows that, although there is not a perfect match, the absorption coefficients for full scale and reduced scale ballast follow a similar trend and the main features of the curves (maxima and minima) are correctly reproduced.

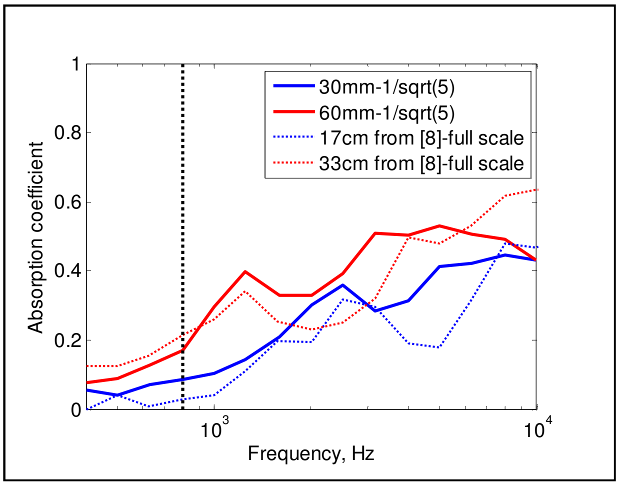

Figure 1. Absorption coefficient for the 1:√5 scale and full scale ballast in reverberation chamber (results at full scale shifted in frequency). Vertical line corresponds to the Schroeder frequency of the room.

The absorption of 1:5 scale ballast was also measured. Figure 2 presents a comparison of the absorption coefficient of the two scale ballasts. The difference between these two results is not very significant and it is on average smaller than the difference between the 1:√5 scale and the full scale ballast. The difference between the results for the two different scale factors is smaller for the larger thickness than for the smaller thickness. It was therefore considered sufficient to use the 1:5 scale ballast to assess both acoustic and mechanical properties in the following sections.

This also avoids a particular problem with the 1:√5 scale ballast, which is that the stone size is quite large compared with the layer thickness.

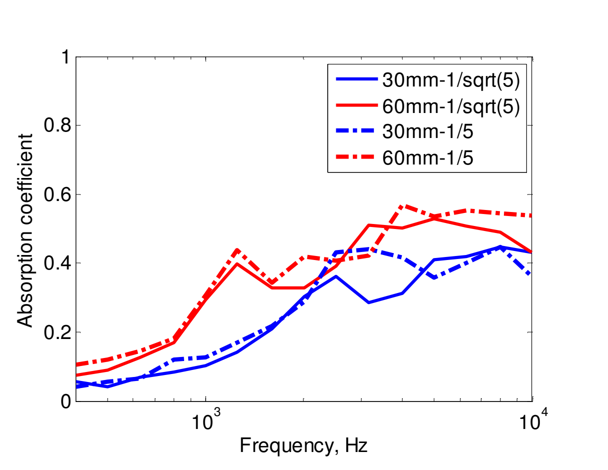

Figure 2. Comparison of the absorption for the 1:5 and 1:√5 scale ballast obtained in reverberation chamber.

3. Effects of ballast on the sleeper radiation

In order to give reliable predictions of the sound radiation from the sleepers, it is necessary to account for the influence of the ballast. The sound radiation of the sleeper can be expressed in terms of its radiation ratio which can be written as

$$\sigma = \frac{W/\overline{F^2}}{\rho_0 c_0 \sum_i \left|\frac{v_i}{F}\right|^2 dS_i} \tag{1}$$

where $W/\overline{F^2}$ is the sound power normalised by the mean-square force and $|v_i/F|^2$ is the squared transfer mobility (complex ratio between velocity and force) from the force position to response position i. dS_i is the surface area of the rail associated with position i.

Following equation (1), the radiation ratio can be estimated from two measurements. One is used to measure the transfer mobility, which has been carried out using an impact hammer. The other is used to obtain the sound power for a given force. This has been determined reciprocally by measuring the acceleration response to a measured sound pressure field in a reverberation chamber [11]. Note that the mobility measurements should be carried out for the same configuration as the sound power tests.

A 1:5 scale model of a sleeper has been measured initially in a freely suspended condition. The radiation ratio for this is compared with a boundary element (BE) prediction in Figure 3. The BE prediction is based on an assumption of rigid body motion of the sleeper although in practice a number of bending modes occur in the frequency range considered, the first occurring at around 530 Hz. Nevertheless, generally good agreement is found apart from close to this resonance frequency.

The sleeper has next been fully embedded in a layer of ballast. The ballast had a total thickness of 100 mm, with 60 mm beneath the sleeper and the top surface flush with the top of the sleeper. Two cases are considered: in one the ballast was resting on the floor of the chamber, which can be considered rigid, while in the other case it was contained in a wooden box.

The results are shown in Figure 3. The sound radiation for the sleeper embedded in ballast is higher than that of the sleeper in free space, especially at low frequency. The two different foundations for the ballast have different effects on the sleeper radiation. In particular the radiation ratio is higher when the ballast is resting in the wooden box than on rigid ground. There are various possible reasons for the higher radiation ratio of the embedded sleeper compared with the case of a sleeper in free space. Firstly the sleeper tends to have the characteristics of a monopole when it is embedded in ballast instead of a dipole when in free space. This changes the slope of the radiation ratio curve at low frequencies. Secondly the vibration of the ballast itself can increase the radiated noise and this may be enhanced by a softer support such as the wooden box. Finally the wooden box itself may contribute to reradiated noise.

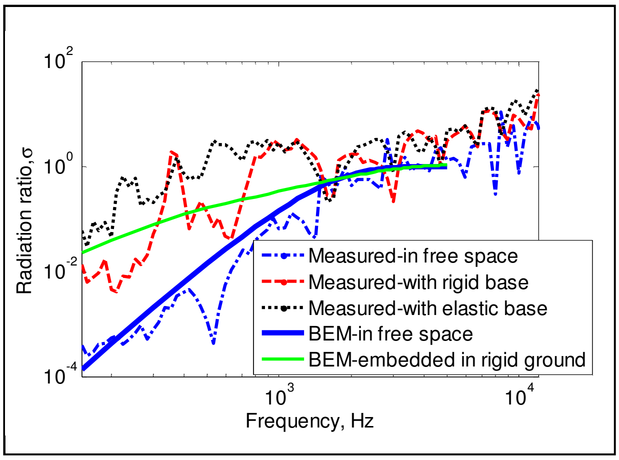

Figure 3. Measured radiation ratio for sleeper embedded in ballast and in free space compared with numerical predictions.

4. Measurements of ballast vibration

It has been seen in Section 3 that more noise is radiated when the sleeper is embedded in ballast. One possible reason is that the ballast vibrates, induced by the sleeper vibration, and this radiates sound. The ballast vibration is explored here. A scanning laser vibrometer has been used to measure the vibration of the ballast when the sleeper was excited using an instrumented hammer. The excitation was applied at the point at which the rail would be connected to the sleeper. The ballast surface was scanned to one side of the sleeper to obtain transfer mobilities between the force on the sleeper and the vibration of the ballast. Figure 4 shows the grid of measurement points. Approximately 100 points were used to capture the ballast vibration, while 7 points were used for sleeper.

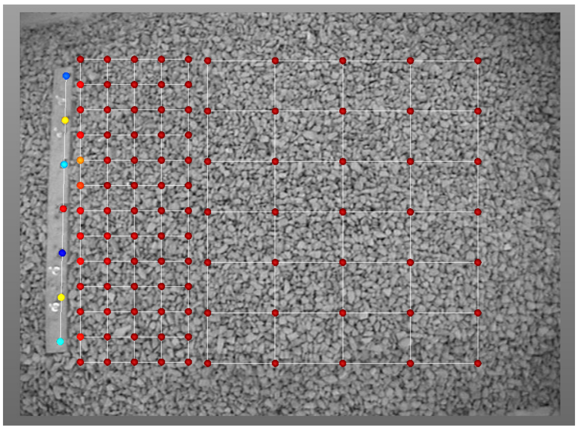

Figure 4. Measurement grid for ballast and sleeper vibration for sleeper embedded in ballast.

Figure 5 shows the averaged mean square mobility of the sleeper and the ballast at different distances away from the sleeper when the ballast is located on the rigid ground and in the wooden box. Each line corresponds to the average over the positions at a certain distance from the centre of the sleeper. Some results are omitted around 1 kHz as the data was found to be contaminated. As can be seen, for both cases, the vibration level of the sleeper is much higher than that of the ballast. Generally, the vibration level of the ballast becomes smaller as the distance increases. The peak in the sleeper vibration at around 1.64 kHz corresponds to the second vibration mode of the sleeper (the first is not excited as the excitation point is close to a nodal point). The ballast vibration at a certain distance away from the sleeper is less when the ballast is located on the rigid ground than in the box. It is also can be seen that, for the rigid ground, the ballast vibration drops dramatically below 300 Hz compared with that of the sleeper whereas in the box, below 200 Hz, the ballast vibration drops very little beyond 0.1 m. At low frequencies, some influence of vibration modes of the wooden box can be seen. Clearly at low frequencies the ballast vibration is strongly affected by the stiffness of the foundation.

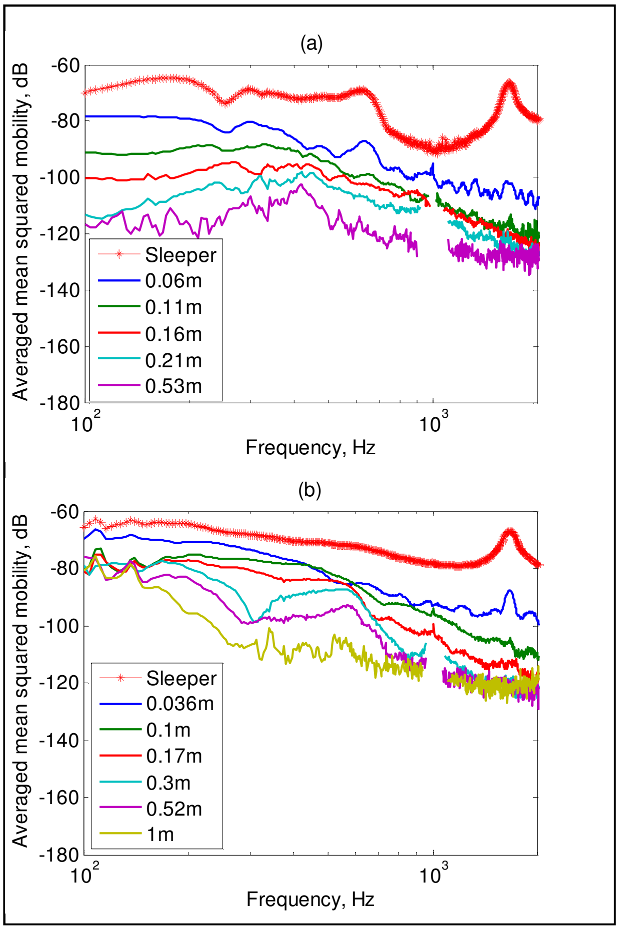

Figure 5. Comparison of sleeper vibration and ballast vibration at different distances for sleeper embedded in ballast. (a) On rigid ground; (b) In wooden box.

Figure 6 presents these results as a decay with distance of the ballast vibration at different frequencies. The average sleeper vibration is taken as the reference value. As can be seen, for both cases, the ballast vibration decays with distance over the whole frequency range, but the decay is smaller at low frequency. Also, at a given frequency, the decay of the ballast vibration with distance when the ballast is contained in the wooden box is lower than on the rigid ground.

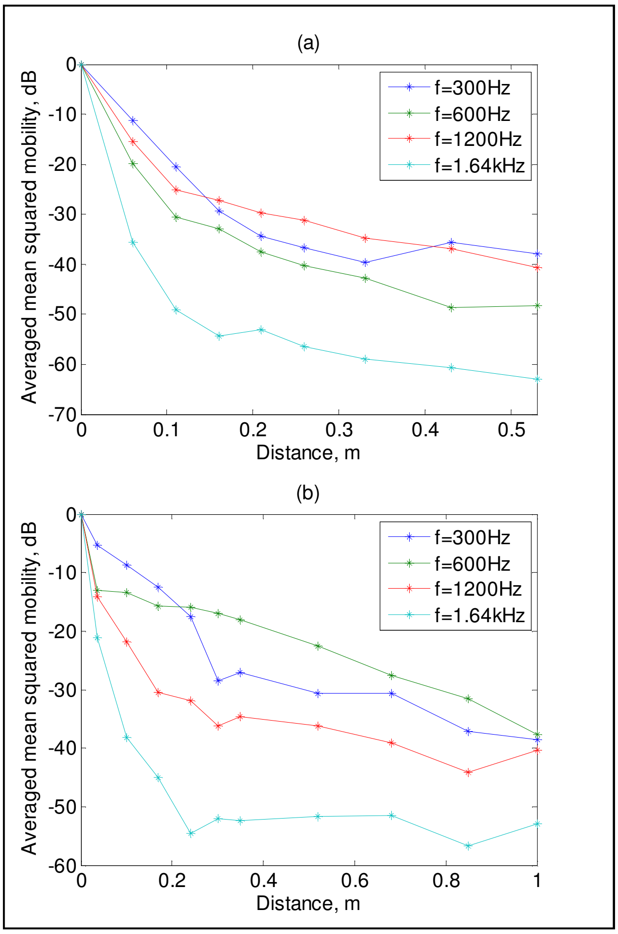

Figure 6. Vibration decay with distance for sleeper embedded in ballast. (a) On rigid ground; (b) in wooden box.

5. Estimation of effects of ballast vibration on sound radiation

Although the ballast has a lower vibration level than the sleeper, it may still contribute to the noise due to its larger radiating area. Therefore, the sound power from the ballast vibration is estimated here.

The sound radiated by a vibrating structure mounted in a rigid baffle can be calculated by using the Rayleigh integral [12]. The pressure at a point $\mathbf{r}=(r,\theta,\phi)$ can be expressed in terms of the structure surface complex velocity $v(x,y)$ as

$$p(\mathbf{r})=\frac{jk\rho c}{2\pi}\int_S v(x,y)\frac{e^{-jkr'}}{r'}\mathrm{d}x\mathrm{d}y \qquad (2)$$

where k is the acoustic wavenumber and $r'=|\mathbf{r}-\mathbf{x}|$ represents the distance between a point $\mathbf{x}$ on the structure surface.

The sound power can be obtained by integrating the acoustic intensity over a hemisphere in the far field as

$$W=\int_0^{2\pi}\int_0^{\pi/2}\frac{|p(\mathbf{r})|^2}{2\rho c}r^2\sin\theta\mathrm{d}\theta\mathrm{d}\phi \qquad (3)$$

where ρ is the air density, c is the speed of the sound and $p(\mathbf{r})$ is the complex acoustic pressure amplitude at $\mathbf{r}=(r,\theta,\phi)$.

The effect of the ballast vibration on the sleeper radiation has been evaluated based on the above equations. The radiated power is calculated including an increasing area of the ballast in the Rayleigh integral. The measured ballast vibration levels per unit force obtained using the scanning laser vibrometer are used in this calculation although phase is ignored for simplicity. This will yield an upper bound for the radiated sound.

The results are shown in

Figure 7. The values in the legend indicate the width of the ballast on one side of the sleeper included in the calculation; symmetry is adopted to consider the ballast on the non-measured side of the sleeper as well. As can be seen, for both cases, the ballast vibration has an impact on the sound power of the system below about 1500 Hz (300 Hz at full scale). Above 500 Hz (100 Hz at full scale), only the ballast vibration close to the sleeper (0.1 m) contributes significantly to the sound power and the increase in noise is less than 3 dB. Below this frequency an increase of around 7 dB is found due to the ballast.

When the ballast is located on the rigid ground, the increase in the sound power due to the ballast is smaller than when it is located in the wooden box. The ballast radiates more noise in the latter case because of the compliance of the elastic base so that a greater area of ballast is involved. This suggests that the effect of the ballast vibration on the sleeper radiation depends strongly on the stiffness of the foundation. Further research is needed to quantify this in more detail for the real situation. Moreover, it can be expected that the vibration of multiple sleepers in a track will modify this behavior.

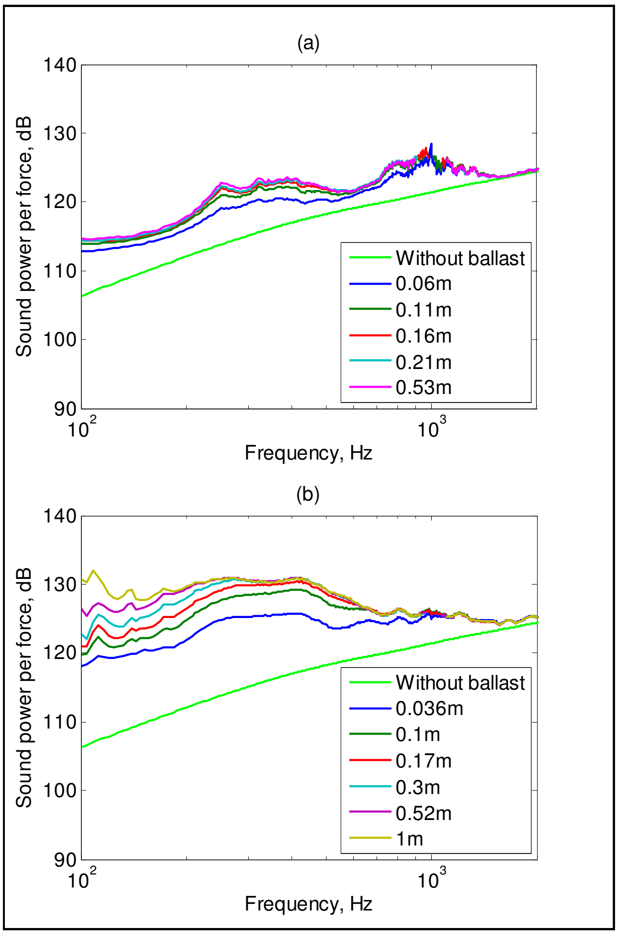

Figure 7. Sound power of the sleeper embedded in ballast. (a) On rigid ground; (b) in wooden box.

6. Conclusions

The effects of railway ballast on the sound radiation from the sleeper have been quantified. A one-fifth scale model of the track has been used to carry out acoustic and vibration measurements. Similar ballast absorption has been found when the stones are scaled at 1:√5 and 1:5 and good agreement is found with full scale ballast. It is also shown that vibration of the ballast affects the sleeper radiation below 1500 Hz for the scale model (300 Hz at full scale). Above 500 Hz (100 Hz at full scale), only the region of ballast close to the sleeper (within 0.1 m) contributes significantly to the sound power and the increase in noise is less than about 3 dB. At lower frequencies the sound power has been estimated to increase by around 7 dB for the ballast with a rigid base, and 12 dB for the ballast on an elastic base. Further research is needed to quantify this in more detail for the real situation.

Acknowledgement

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References

[1] D.J. Thompson, B. Hemsworth and N. Vincent. Experimental validation of the TWINS prediction program for rolling noise, part 1: description of the model and method. Journal of Sound and Vibration, 193, 123-135, 1996.

[2] K. Knothe, S. Grassie. Modelling of railway track and vehicle/track interaction at high frequencies. Vehicle System Dynamics, 22, 209-262, 1993.

[3] D. R. Ahlbeck, H. C. Meacham, R. H. Prause. The development of analytical models for railroad track dynamics. Railroad Track Mechanics & Technology, Pergamon Press, Oxford, 1978.

[4] W. M. Zhai, K. Y. Wang, and J. H. Lin. Modelling and experiment of railway ballast vibration. Journal of Sound and Vibration, 270, 673-683, 2004.

[5] G. W. C. Kaye, E. J. Evans. The sound absorbing properties of some common outdoor materials. Proceedings of the Physical Society, 52, 371-379, 1940.

[6] K. Attenborough, P. Boulanger, Q. Qin, and R. Jones. Predicted influence of ballast and porous concrete on rail noise. Internoise , Brazil, 2005.

[7] K. Heutschi. Sound propagation over ballast surfaces. Acta acustica united with acustica, 95, 1006-1012, 2009.

[8] R. A. Broadbent, D. J. Thompson, and C. J. C. Jones. The acoustic properties of railway ballast. EURONOISE, Edinburgh, 2009.

[9] M. C. Junger. Model scaling laws for sound absorptive boundaries. Journal of the Acoustical Society of America, 62, 209-211, 1977.

[10] BS EN ISO 354:2003. Acoustics-Measurement of sound absorption in a reverberation room. British Standards Institution, 2003.

[11] G. Squicciarini, et al. Use of a reciprocity technique to measure the radiation efficiency of a vibrating structure. Applied Acoustics 89, 107-121, 2015.

[12] F. Fahy, P. Gardonio. Sound and structural vibration, radiation, transmission and response. Second version, Elsevier, 2007.