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TESTING THE FUNCTIONALITY AND PERFORMANCE OF A RAIL DAMPER

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Rezumat

Amortizoarele de şină sunt dispozitive mecanice care lucrează ca absorbitoare dinamice pentru a reduce vibraţia şinei şi zgomotul de rulare. Articolul prezintă rezultatele experimentale de la testarea funcţionalităţii şi performanţei unui amortizor de şină experimental demonstrativ. Atenuarea vibraţiei ia cele mai mari valori între 160 şi 1000 Hz, unde atenuarea este cuprinsă între 6 şi 22 dB.

Cuvinte cheie: şină, amortizor de şină, atenuarea vibraţiei, testarea funcţionalităţii şi performanţelor

Abstract

The rail dampers are mechanical devices which work as dynamic absorbers to reduce the rail vibration and rolling noise. The paper shows the experimental results from the functionality and performance testing of an experimental demonstrative rail damper. The vibration attenuation takes the highest values, namely 6-22 dB, between 160 and 1000 Hz.

Keywords: rail, rail damper, vibration attenuation, functionality and performance testing

1. INTRODUCTION

Trains are significant sources of vibration and noise which affect the travellers' comfort and the environment near to the railway lines, including civil structures and habitants. The main noise emerged from the trains when their speed is ranged between 30 and 270 km/h comes from the rolling gear and rails – the so-called rolling noise. The rolling noise occurs due the structural vibration of the wheels and track components excited by the presence of the rolling surfaces' roughness [1].

Practically, the rolling noise emerges from wheels [2], rails [3] and sleepers [4]. The noise produced by the rails has the highest level of up to 1500 Hz due to the fact that the rail receptance is higher than the wheels. However, there is a short frequency band around of the anti-resonance frequency of the rail when the noise emerged from the sleepers becomes dominant. At high frequency, higher than 1500 Hz, the rails and wheels contribute to the overall noise level in comparable percentages.

To reduce the rolling noise component due to the rails, the rail dampers are being used more and more lately. They are fixed by the rail web or even by the rail foot and work as dynamic vibration absorbers. In this way, the bending waves that propagate along the rails are strongly attenuated and moreover the rolling noise is reduced because a shorter length of the rail vibrates effectively [5].

In this paper, the functionality and performance of an experimental demonstrative rail damper are investigated into a laboratory environment. The rail damper is of innovative conception mixing the viscoelastic properties of the rubber with the oil damping to improve vibration attenuation.

2. RAIL DAMPER DESCRIPTION

The usual rail dampers include steel plates encapsulated in rubber, and they are clamped on the rail web between sleepers (Figure 1).

(a)

(b)

Figure 1. Rail dampers: (a) TATA Steel; (b) Vossloh [6].

Many of these rail dampers work as dynamic absorber with two tuned frequencies, being effective within an enlarged frequency range.

Figure 2. Rail damper

In this case, the rail damper has a cylindrical body placed via rubber rings into a cylindrical shell (Figure 2). A thin oil film is inserted between the body and shell. Two end caps ensure sealing of the rail damper. The compound is inserted into a rubber piece and glued on the rail web.

Dimensions of the rail dampers have been chosen so that it can equip the secondary or urban lines provided with the UIC rail 49. The damper's length is 210 mm, and its mass is 3.4 kg, which means it is lighter compared to other dampers used for heavy-rails (i.e. UIC 60 type) due to the smaller volume available for mounting.

3. FUNCTIONALITY TESTING

Testing of the rail damper functionality was performed on a short length of UIC 49 rail (length of 835 mm) by comparing the dynamic response of the rail with the dampers fitted with that obtained with the undamped rail. In both cases, the excitation was applied in the middle of the coupon and the accelerations were measured both in the middle of the coupon and at one of its ends in relation to the elastic support (Figure 3). The coupon was supported by two sleepers on a metallic frame.

To perform the functionality testing, two accelerometers have been mounted on the rail coupon in the middle and above a sleeper and an impact hammer has been used to put in excitation the rail coupon.

Figure 4 shows the receptance in the middle of the non-damping rail coupon as well as at one end when the impact hammer is applied at the middle of the coupon. The resonance frequency at 1075 Hz is observed, as well as the anti-resonance frequency from approx. 680 Hz in the middle of the rail coupon. The two peaks at 26 and 120 Hz are the impact of the rail coupon support, including rail pads and sleepers.

Figure 3. Set-up of the testing of the rail damper functionality

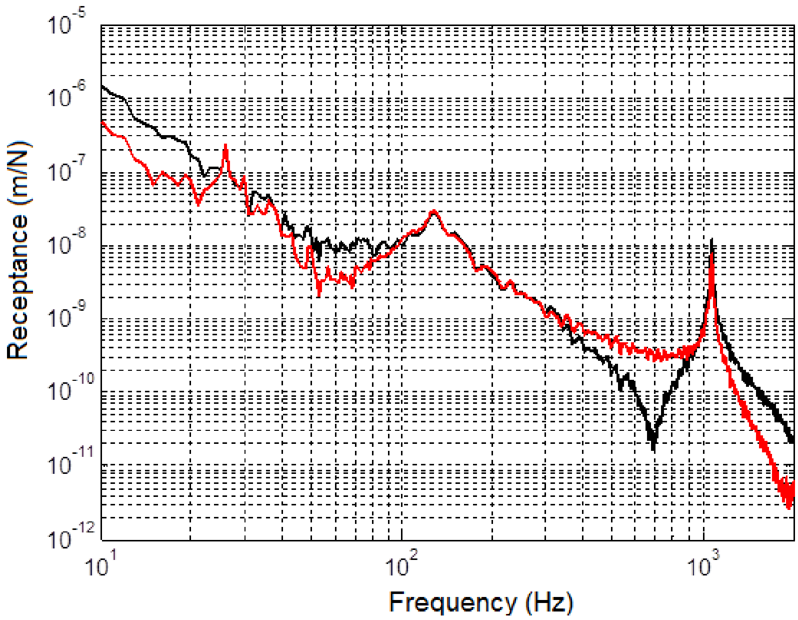
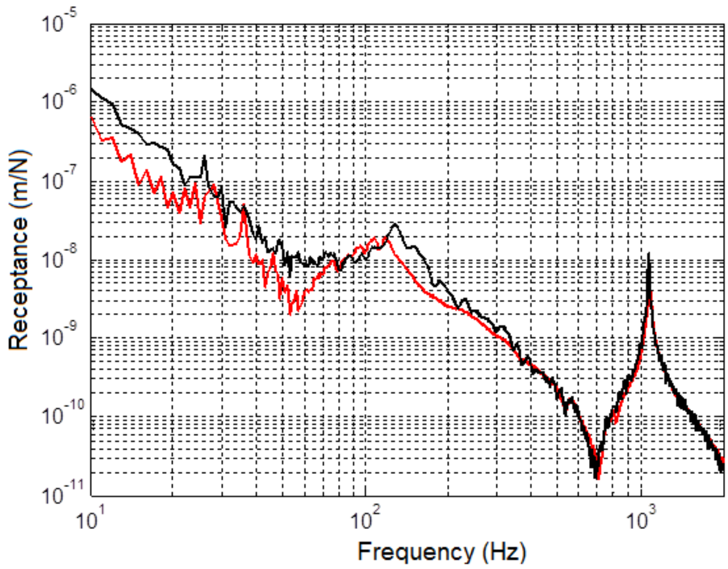
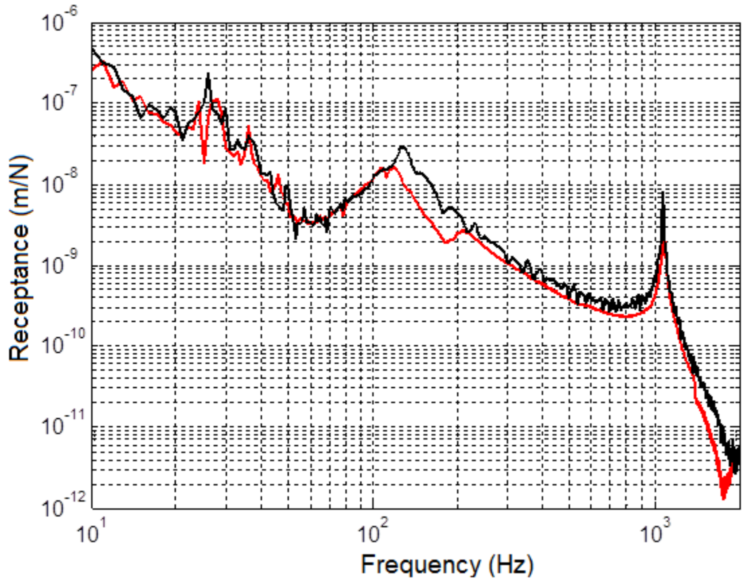


Figure 4. Rail receptance: black line – in the middle, red line – above sleeper.

Figure 5 shows the comparison between the free-damper rail receptance and the receptance of the rail with dampers. It is noted that in the middle of the rail coupon the dampers diminish the dynamic response in the low and medium frequencies, while at one end, the dampers influence the response in the medium and high frequency range. It can be concluded that by the measurements made, the functionality of the rail dampers has been demonstrated.



(a)



(b)

Figure 5. Rail receptance: (a) at middle; (b) above sleeper; black line – without dampers, red line – with dampers

4. PERFORMANCE TESTING

The dynamic response of the rail was determined using a 5.4 m long rail coupon placed on two elastic supports formed by each of 7 rubber plates (Figure 6) to ensure the free bend conditions at the ends. In other words, the resilient supports have to provide the bounce natural frequency of the rail coupon lower than the lowest bending frequency of the rail coupon. Accelerometers were fitted at the two ends of the rail coupon (Figure 7). For ease of exposure, we call the 'active end' the end where the force of excitation was applied by means of the impact hammer, and the 'passive end', the other end.

Figure 6. Rail coupon with dampers on resilient supports.

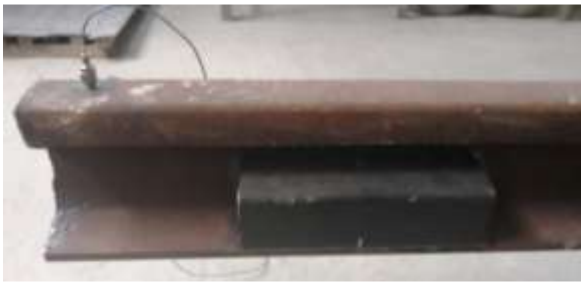


Figure 7. Accelerometers' set-up

The rail response was therefore determined at both ends for both the undamped rail and the damped rail. It is noted that 9 pairs of rail dampers were mounted equidistant at 60 cm intervals. The distance between the rail ends and the first pairs of dampers is 30 cm. Considering that the mass of a shock absorber is 3.4 kg, an equivalent linear mass of 11.3 kg / m results.

Vibration attenuation is calculated starting from the rail frequency response determined via the Fourier integral for each 1/3 octave interval

$$A_k = 10\lg \frac{\sum_{i=1}^{N_k} \left|\alpha_{ud}(\omega_i)\right|^2}{\sum_{i=1}^{N_k} \left|\alpha_d(\omega_i)\right|^2}, \qquad (1)$$

where $\alpha_{ud}(\omega_i)$ and $\alpha_d(\omega_i)$ are the receptance for the undamped and damped rail corresponding to the ω_i circular frequency. It notes that the attenuation is calculated considering N_k components for the k 1/3 octave interval.

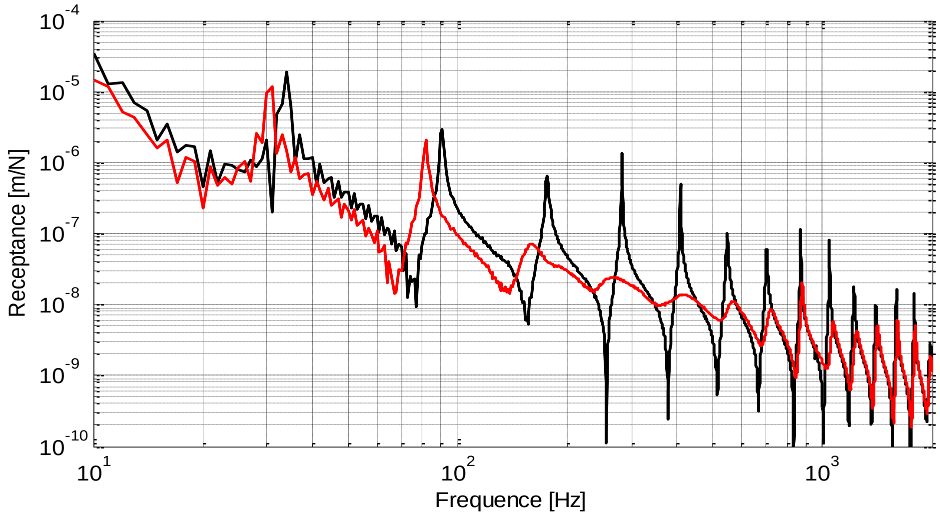


Figure 8. Receptance at the active end: black line – undamped rail, red line – damped rail.

Using the above equation, attenuation has a positive sign if the undamped rail reception is greater than that of the shock absorber rail, which demonstrates the effectiveness of the dampers.

The receptance at the two ends for the undamped and damped rail is shown in Figure 8 for the active end and Figure 9 for the passive end.

It is noted that due to the mounting of the dampers, the resonance frequencies are reduced and this aspect is related to the increase of the modal mass of the bending modes. Note that the damping is effective in the frequency range 100-1000 Hz.

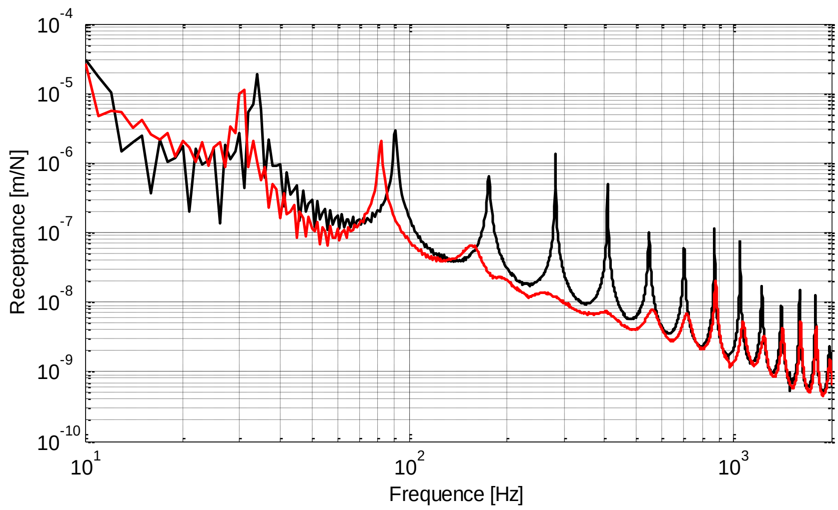


Figure 9. Receptance at the passive end:
black line – undamped rail, red line – damped rail.

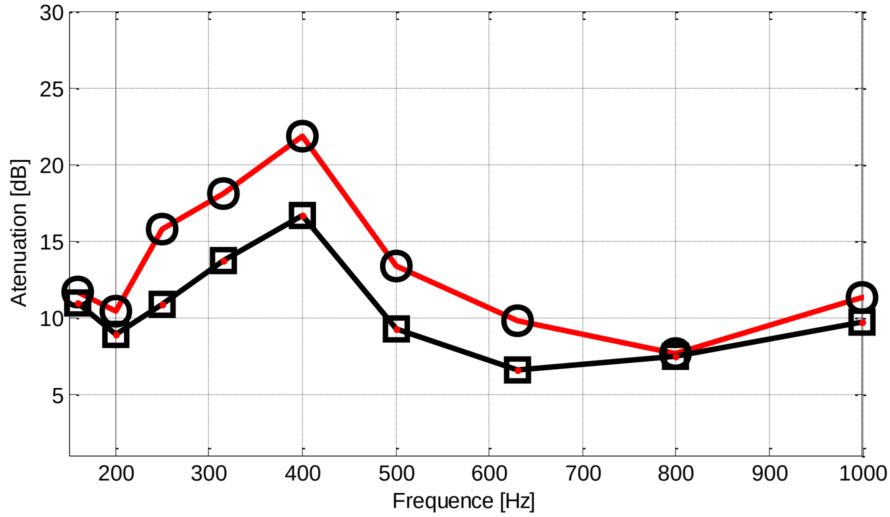


Figure 10. Attenuation: black line – at the active end, red line – at the passive end.

The vibration attenuation at both ends of the undamped - damped rail is shown in Figure 10 for 1/3 octave ranges from 160 to 1000 Hz. Attenuation is lower at the active end comparing to the one at the passive end. The highest attenuation is at 400 Hz: 16 dB at the active end and 22 dB at the passive end. The lowest attenuation can be observed between 630 and 800 Hz, around 6-8 dB.

5. CONCLUSIONS

In this paper, the functionality and performance of an innovative experimental demonstrative rail damper has been tested in a laboratory environment. To this purpose, two rail coupons of 0.835 and 5.4 m have been used to determine the rail receptance by means of the impact hammer method.

The results show that the rail dampers provide an attenuation of 6-22 dB in the frequency range of 160 – 1000 Hz.

Acknowledgments

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