

Experimental study on the performance of a rail damper

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Abstract. This paper is aiming to present the assessment of the performance of an innovative experimental demonstrative rail damper, which mixes viscoelastic properties of rubber with the damping capacity of oil and the inertia of two steel pieces. Nine pairs of rail dampers are mounted every 60 cm on both sides of the rail web along a 5.4 m long rail. The experimental results of assessing the vibration attenuation on the undamped versus damped rail are described. The frequency range of the dampers efficacy is identified and the attenuation of vibration is determined.

1. Introduction

The rolling noise due to the railway vehicles is the most important noise source for a wide range of speed, namely, from 30 to 270 km/h. Rolling noise is generated when the vehicle is running along a straight track, without any local defects in rails or wheels, due to the wheel/rail vibration caused by small amplitude irregularities of the rolling surfaces of both wheel and rail [1].

In terms of the track-generated rolling noise, the rails and sleepers play the main roles, the last ones only in a narrow frequency band, due to the bending waves propagating from the cross-sections of the contact between wheels and rail along the track. Actually, the rolling noise emerged from the rails is dominant at low and middle frequencies up to 1500 Hz because the rail's reacceptance is higher than the one of the wheel.

One way to reduce rolling noise produced by the track is to attenuate the bending waves that propagate along the rail using the rail dampers. Rail dampers are mechanical devices mounted at both sides of the rail web which work as dynamic vibration absorbers [2]. In this way, a shorter length of the rail vibrates effectively and the rolling noise is less intense.

The evaluation of the rail dampers effectiveness is not standardized, hence making the comparison between various types of rail dampers difficult. However, it is considered that the bending waves decay rate in rail is the main parameter to be assessed by either theoretical modelling or measurements [3]. The decay rate may be experimentally determined in field tests on rails with rail dampers but this is difficult and costly. Alternatively, the decay rate may be determined in laboratory measurements using the frequency response function of a freely supported short rail – the short-free rail method [4, 5].

This paper deals with the issue of the assessment of the performance of an innovative experimental demonstrative rail damper which mixes the viscoelastic properties of the rubber with the damping capacity of the oil and the inertia of two steel pieces. The rail dampers performance is assessed in



terms of the vibration attenuation in 1/3 octave frequency intervals. To this end, the short-free rail method is applied.

2. Laboratory measurements

This section describes the laboratory measurements to assess the vibration attenuation for a short-freely rail (5.4 m length) fitted or not with rail dampers.



Figure 1. Rail damper.

Generally, the rail dampers are made of steel plates assembled in rubber cases and clamped on both sides of the rail web [6]. In this particular case, the dampers are composed of two steel pieces connected by rubber elements and embedded in a rubber casing. The damping is given by both the rubber and oil between the two pieces (figure 1). The dampers are fixed to the rail web with adhesive.

The damper is designed for the UIC 49 rail type, commonly used for secondary lines or urban rail transport. The damper's length is 210 mm, and its mass is 3.4 kg, which compared to other dampers used for heavy-rails (i.e. UIC 60 type) is lighter due to the smaller volume available for mounting.

Nine pairs of rail dampers are mounted every 60 cm on both sides of the rail web along the rail section (figure 2). The distance between the free rail ends sections and the vertical axis of the first pair of dampers is 30 cm, and the specific linear mass of the dampers is 11.3 kg/m.

To ensure free vibration conditions, the rail is placed on two supports comprising of 7 rubber plates, such that the frequency of the bounce mode is less than the frequency of the first bending mode.



Figure 2. Short rail with dampers.

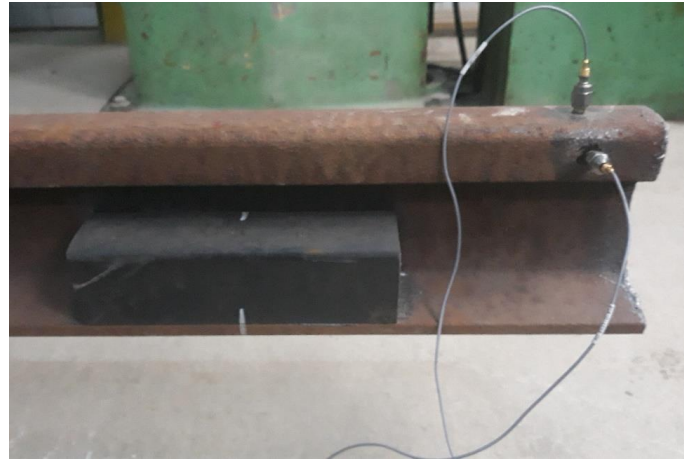


Figure 3. Accelerometers' setup.

Two Bruel&Kjaer 4514 type accelerometers capable of capturing vibrations between 1.0 and 10000 Hz (figure 3) were placed on the rail head and on its flank at the two ends of the rail.

The data acquisition system consists of NI cDAQTM-9174 chassis and NI 9234 serial input/output module. Data processing is performed at 64 kHz sampling frequency. The rail is excited at one end (named “active end”) by an instrumented (impact) hammer.

The vibration attenuation is determined by processing the accelerations measured at the two ends of the rail coupon when the instrumented hammer is applied to the active end. Basically, the frequency response functions are calculated using the Fourier integral for every 1/3 octave band and then the attenuation is computed.

Attenuation corresponding to the k 1/3 octave interval having the limits between the angular frequencies ω_{kmin} and ω_{kmax} can be calculated using the following equation (1):

$$A_k = 10 \lg \frac{\sum_{i=1}^{N_k} |\alpha_{ud}(\omega_i)|^2}{\sum_{i=1}^{N_k} |\alpha_d(\omega_i)|^2} [\text{dB}] \quad (1)$$

where $\alpha_{ud}(\omega_i)$ and $\alpha_d(\omega_i)$ are the receptances for the undamped and damped rail for the angular frequency ω_i , ($\omega_{kmin} < \omega_i < \omega_{kmax}$), and N_k are the angular frequency points within the k 1/3 octave interval.

Using above equation, attenuation has positive sign as long as the receptance of the undamped rail is higher than the receptance of the damped rail and the rail dampers work.

3. Experimental results

This section describes the results with regards to vibration attenuation obtained by using the above procedure.

Figure 4 shows the rail receptance measured at both ends in the 10 – 2000 Hz frequency range for undamped rail. The receptance shows a characteristic shape at the active end due to the alternation of resonance and anti-resonance frequencies. The anti-resonance frequencies no longer appear at the passive end due to the reflection of the bending waves. Note that both symmetrical and anti-symmetrical bending modes are excited.

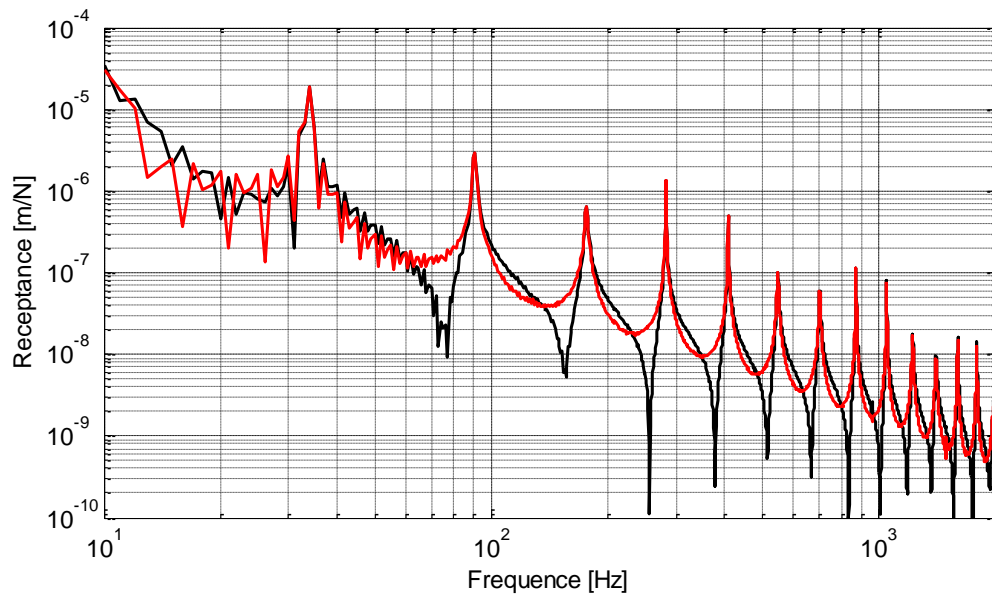


Figure 4. Receptance of undamped rail: black – at the active end, red – at the passive end.

Table 1 contains the resonance and anti-resonance frequencies. These values are close to the theoretical values obtained using the free-free Timoshenko beam model. On the other hand, the bounce frequency is lower than the first bending frequency.

Table 1. Resonance and anti-resonance frequencies.

	Frequency (Hz)												
Resonance frequency	34	91	176	282	408	549	703	872	1046	1221	1404	1597	1787
Anti-resonance frequency	75	155	256	378	516	668	835	1007	1182	1366	1556	1744	1931

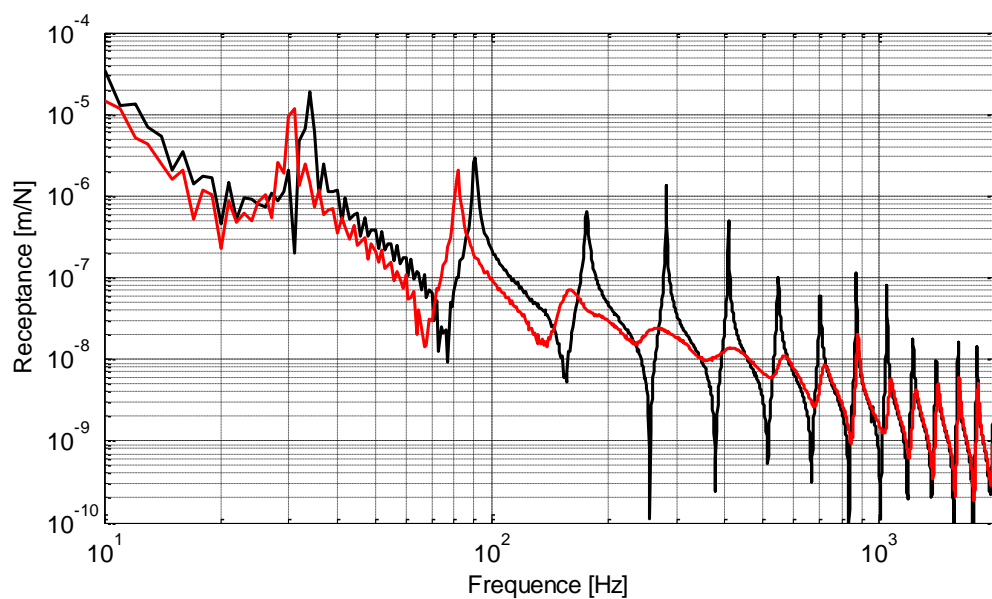


Figure 5. Receptance at the active end: black – undamped rail, red – damped rail.

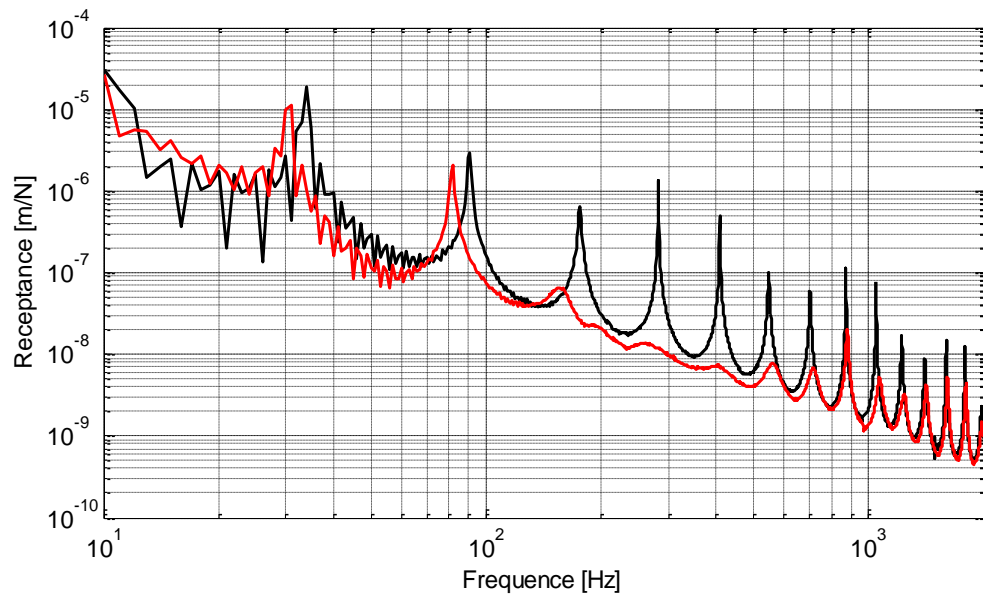


Figure 6. Receptance at the passive end: black – undamped rail, red – damped rail.

Figures 5 and 6 show the measured receptances for both undamped and damped rail (active end - figure 5, and passive end - figure 6, respectively).

There is a decrease of the resonance frequencies elicited by mounting the dampers on the rail and consequently an increase of the modal masses of the vibration eigenmodes.

The damping effect is especially effective for frequencies in the range of 100-1000 Hz.

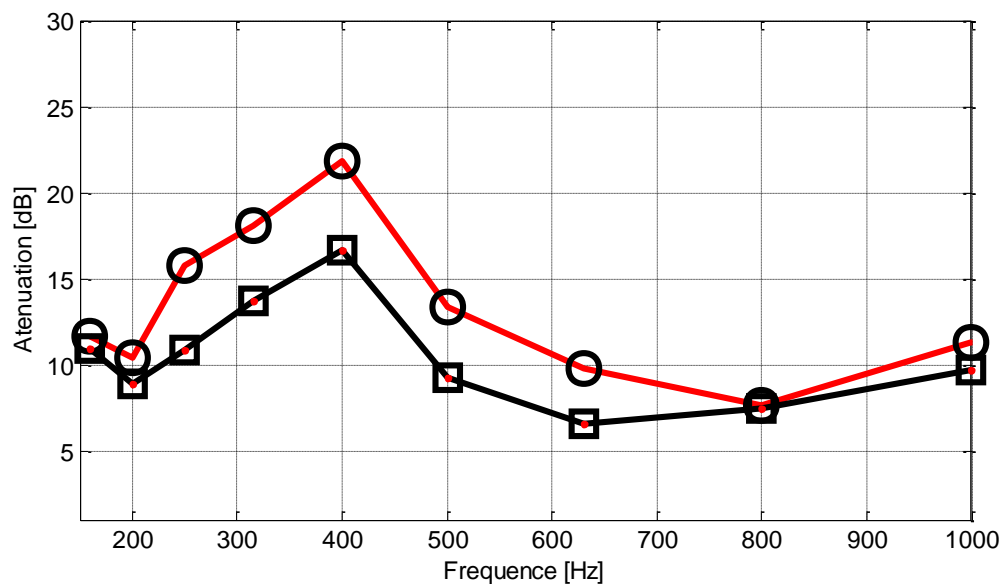


Figure 7. Attenuation: black – at the active end, red – at the passive end.

The vibration attenuation at both ends for the undamped versus damped rail is quantified and shown in figure 7 for 1/3-octave intervals with frequencies ranging from 160 to 1000 Hz. The attenuation is greater at the passive end when the rail dampers are embedded.

4. Conclusions

In this paper, the performance of an innovative experimental demonstrative rail damper which mixes the viscoelastic properties of the rubber with the damping capacity of the oil and the inertia of two steel pieces which works as a dynamic absorber with two tuning frequencies, has been evaluated experimentally. To this end, the short-free rail method was used for a 5.4 m UIC 49 rail to experimentally determine the rail's response like receptances.

The results showed that the effectiveness of the rail dampers is between 160 and 800-1000 Hz and the vibration attenuation (undamped/damped rail) is ranged between 6 and 22 dB.

Based on the results obtained, the future research should be focused on modelling damped rail to identify the constructive parameters on which the rail dampers performance depends.

Acknowledgments

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5. References

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