Structural Engineering and Mechanics, *Vol. 48, No. 4 (2013) 455-466*
DOI: http://dx.doi.org/10.12989/sem.2013.48.4.455

Continous rail absorber design using decay rate calculation in FEM

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(Received May 18, 2013, Revised August 17, 2013, Accepted October 25, 2013)

Abstract. In recent years, many countries have added railway noise to the issues covered by noise regulations. It is known that the rail is the dominant source of rolling noise at frequency range of 500Hz - 2000Hz for the conventional speeds (<160km/h). One of the effective ways to reduce noise from railway track is using a rail vibration absorber. To study the acoustic performance of rail absorber, the decay rates of vibration have long been used by researcher. In this paper, A FE model of a periodic supported rail with infinite element in ABAQUS is developed to study the acoustic performance of the rail absorber. To compute the decay rates, acceleration responses along the rail transferred to MATLAB to obtain response levels in frequency domain and then by processing the response levels, the decay rates obtained for each1/3octav band. Continous rail absorber is represented by a steel layer and an elastomer layer. The decay rates for conventional rail and rail with one-side absorber and also, the rail with two side absorber are obtained and compared. Then, to improve the system of rail absorber, a steel plate with elastomer layer is added to bottom of the rail foot. The vertical decay rate results show that the decay rate of rail vibration along the track is significantly increased around the tuned frequency of the absorber and thus the rail vibration energy is substantially reduced in the corresponding frequency region and also effective in rail noise reduction.

Keywords: continous rail absorber; decay rate; FEM; ABAQUS; noise; UIC60

1. Introduction

Railways than other forms of transport have advantages to environmental factors such as lower power consumption and less pollution. A major drawback, however, is the intense noise produced by railway transportation. In addition, physiological sleep parameters are most severely affected by rail noise rather than Aircraft noise and road traffic noise (Griefahn *et al*. 2006). Therefore, to develop this form of transport to compete with other forms of transport and to meet legal noise limit, considering noise reduction strategies is necessary. For that, in recent years, a number of countries have established noise regulations in response to public concerns. a routine solution to excess noise from roads are noise barriers that already widely used in some countries along the railway lines (Thompson and Jones 2000).However, the acoustic effect of such barriers is limited and insufficient in some European countries in proportion to the expense. Moreover, these have the

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http://www.techno-press.org/?journal=sem&subpage=8 ISSN: 1225-4568 (Print), 1598-6217 (Online)

disadvantage that they are visually intrusive for the line side residents as well as the passengers, so that many noise control approaches at source have been investigated.

For controlling noise at source the first step is to identify the dominant source. Different noise sources in the railway including impact noise, Curve squeal noise, aerodynamic noise and etc., are the dominant source in different conditions. The main source of noise from railway operations on open line is rolling noise, generated by unevenness of the wheel/rail running surfaces. However, impact noise generated by the wheel running over discontinuities at rail joints or crossings is also an important source of noise, particularly in built-up areas close to stations and yards and squealing noise generated in sharp curves. However, the most important one is rolling noise source due to wheel/ rail interaction for speeds from 50 km/h up to 300 km/h or even higher (Thompson and Gautier 2006). Each of the wheel, rail and sleeper vibration contributes to the overall rolling noise. Among them, the rail is a dominant source in the frequency region 500–2000 Hz for the conventional speeds from 100 to 160 km/h (Liu *et al*. 2009). Rail noise reduction, therefore, leads to overall noise reduction for conventional trains.

For the purpose of informing and validating models for rolling noise and also study of rail absorber effectiveness, the decay rates of vibration with distance along the rail have for many years been measured (Jones *et al*. 2006). Due to the rail infinite length, its motion consists in propagating waves that carry vibrations away from excitation point, obviously, wheel/rail contact point (Bracciali and Piccioli). Normally vibrations with propagating waves decade logarithmically with distance that usually given in term of dB/m. Relationship between decay rate and noise was shown through the theoretical models that a doubling of decay rate is equivalent to a 3 dB reduction in radiated noise (Thompson 2009). So the vertical and lateral decay rates are direct factors in the determination of rail noise.

Various methods have been taken to reduce railway rolling noise. One of the promising approaches is adding damper / absorber to the wheel and rail (Liu *et al*. 2009). Rail dampers are pre-shaped elements of steel and elastic material that are fixed to side of the rail to increase the decay rate of vibration along the rail. At the resonant frequency the movement of the steel mass relative to the rail is at its greatest, and thus the amount of energy is absorbed as internal friction in the elastomer spring is therefore maximized and the vibrations are minimized, so that leads to reducing the noise (Asmussen *et al*.). In field tests with this system, a reduction of 6 dB in the track component of rolling noise was achieved (Wu and Liu 2009). Besides the direct noise mitigation effect these rail dampers can also indirectly reduce noise emission by reducing the roughness growth rate on the rail (Asmussen *et al*.).There are several different types of rail absorber/damper currently available, which were used or tested in some European railway noise projects. Their performances were reported at the 9th International Workshop on Railway Noise, for example, the Schrey and Veit dampers, and the Corus dampers.

Here, effectiveness of the rail absorber/damper known as continous rail absorber is studied. In this absorbing method, rail is covered by rubber-metal sheet. By using the FE model with infinite element in ABAQUS, effect of continous rail absorber on the decay rate of the rail vibration is considered. Decay rate is an indicator for absorber performance

2. FEM model of rail

The decay rates of vibration along the rail have long been used by researchers as a measurable parameter by which to test and improve the accuracy of prediction models. Moreover, recently, it

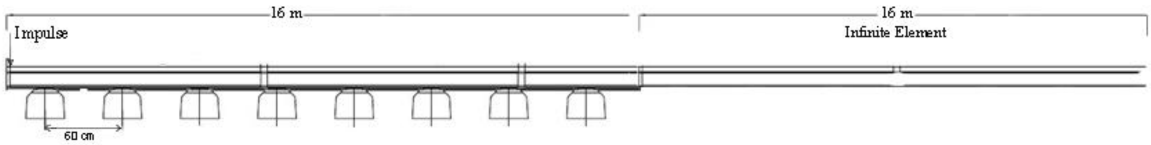

Fig. 1 schematic of rail model and impulse location

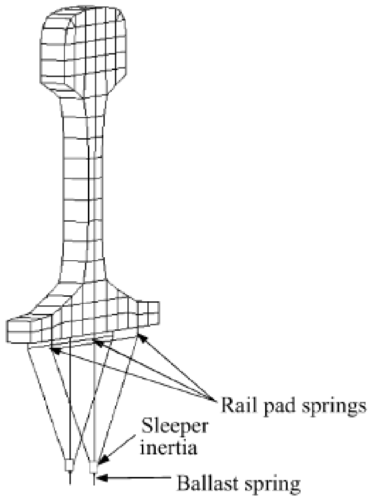

Fig. 2 Rail FE model with rail pad, sleeper and ballast (Brown and Byrn 2005)

has been used as an indicator of the acoustic performance of the track.

The decay rates must be calculated from a set of response, either acceleration or velocity, to different distances along the rail. The necessary response data can be obtained by FE modeling. By this approach the decay rates can be based upon the total response from all waves including near field and propagating. The vertical vibration of the rail is more important than lateral for track with modern soft rail pads since it accounts for a greater proportion of the radiated noise. Here, the train loading is not considered in modeling. However, it has already been shown with more detailed track models that the effects of the train loading can increase the effective decay rate (Jones *et al*. 2006).

To obtain the response data, the 16m rail is modeled by eight-node linear brick elements, in ABAQUS and a force impulse is applied on the rail head at the one end (Fig. 1).

The rail pad, sleeper and the ballast are represented as a series of springs and lumped masses according to the model was used in (Fig. 2). The base of the rail pad is rigidly connected to the sleeper, which is modeled as an inertial element at the centroid of the sleeper cross-section. The ballast is modeled at the centroid of the sleeper cross-section (Fig. 3).

The boundary condition at the end, which is excited by impulse, is symmetry and at the other end to prevent reflection, infinite elements are used. Parameters used in this modeling are presented in Tables 1 and 2.

The response data are made of the vertical acceleration to vertical excitation every 5×10^{-5}s

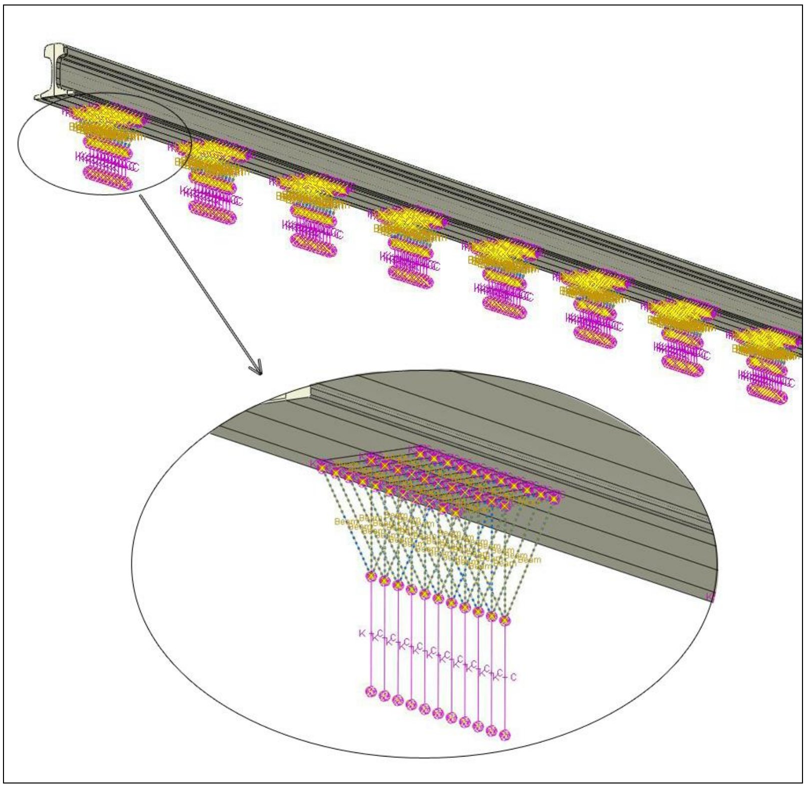

Fig. 3 periodically supported rail model in ABAQUS

Table 1 Ballasted track, mono-bloc sleepers parameters

Rail - UIC60	ρ(kg/m^3)	*N*	*E*(MPa)
	7850	0.3	210
I (kgm^2) of (½ Sleeper (125kg))	Vertical	Lateral	Longitudinal
	1.32	0.64	1.14

Table 2 Properties of spring and damper elements (Jones *et al*. 2006)

Property	pad	Ballast
stiffness(MN/m)	200	150
Vertical damping(Ns/m)	18000	5091

interval in 0.1s period. A ‘grid’ of response location is generally used that was used by the Noemie's project. As Fig. 4 shows responses are obtained from four points per sleeper bay for the first two sleeper bays. For the next few sleeper bays responses are obtained at above-sleeper and mid-sleeper points. Beyond these, measurements are only required at mid-sleeper positions. After obtaining respose data from Abaqus, they transferred to MATLAB to obtain dB levels in one-third octave frequency bands by 1/3 octave filter. Then, the wave decay rates are calculated for one-third octave bands by means of a linear interpolation of band-averaged response in dB versus the distance.

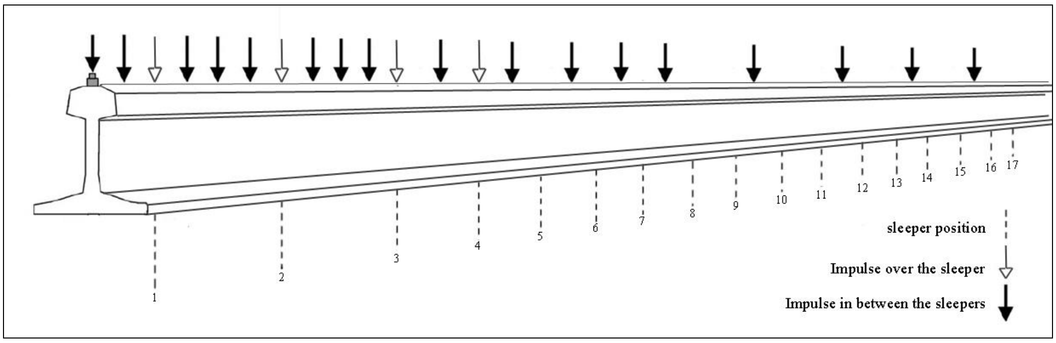

Fig. 4 location of the response derivation points

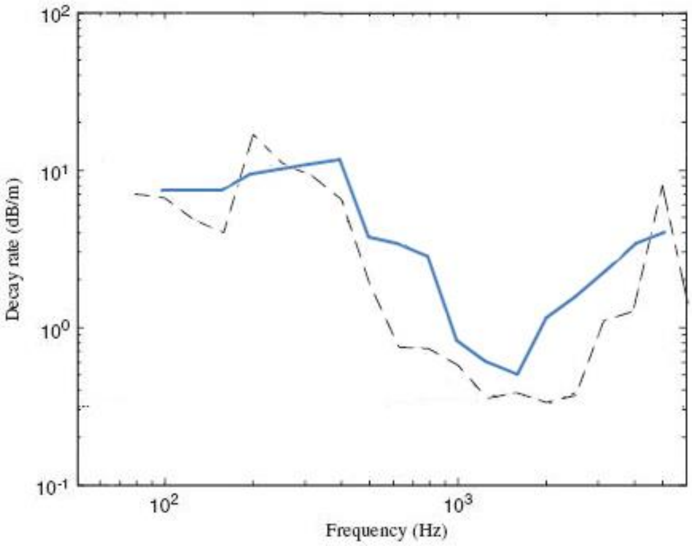

Fig. 5 decay rate: —, FEM model; - - -, Measured response (Jones *et al*. 2006)

To validate the FE model of untreated rail a comparison is made between calculated decay rate and measured decay rate for the ballasted track with mono-bloc sleepers by Jones *et al*. as shown in Fig. 5. The greatest difference exists in the frequency range just above the cut-on of propagation in the rail where the decay drops sharply. Although, the infinite elements at the end are used in FEM, still some wave reflections occur in the rail. So that, there is this discrepancy between the measured decay rate and calculated decay rates from FEM. Nevertheless, it follows the same trend and the FEM decay rate can be used for making comparison between different rail absorber and as an indicator of their performance.

Furthermore, the response data are plotted together versus distance, for each one-third octave frequency band (Fig. 6). Near-field effects can be seen throughout but especially in the low frequency bands up to 315 Hz. In the 1250 Hz-1600Hz band, the relatively high response of the near-field wave and low decay rate of the propagating wave leads to a clearly higher estimate of the decay rate than that of the propagating wave alone. The effect of the pinned–pinned mode can be seen

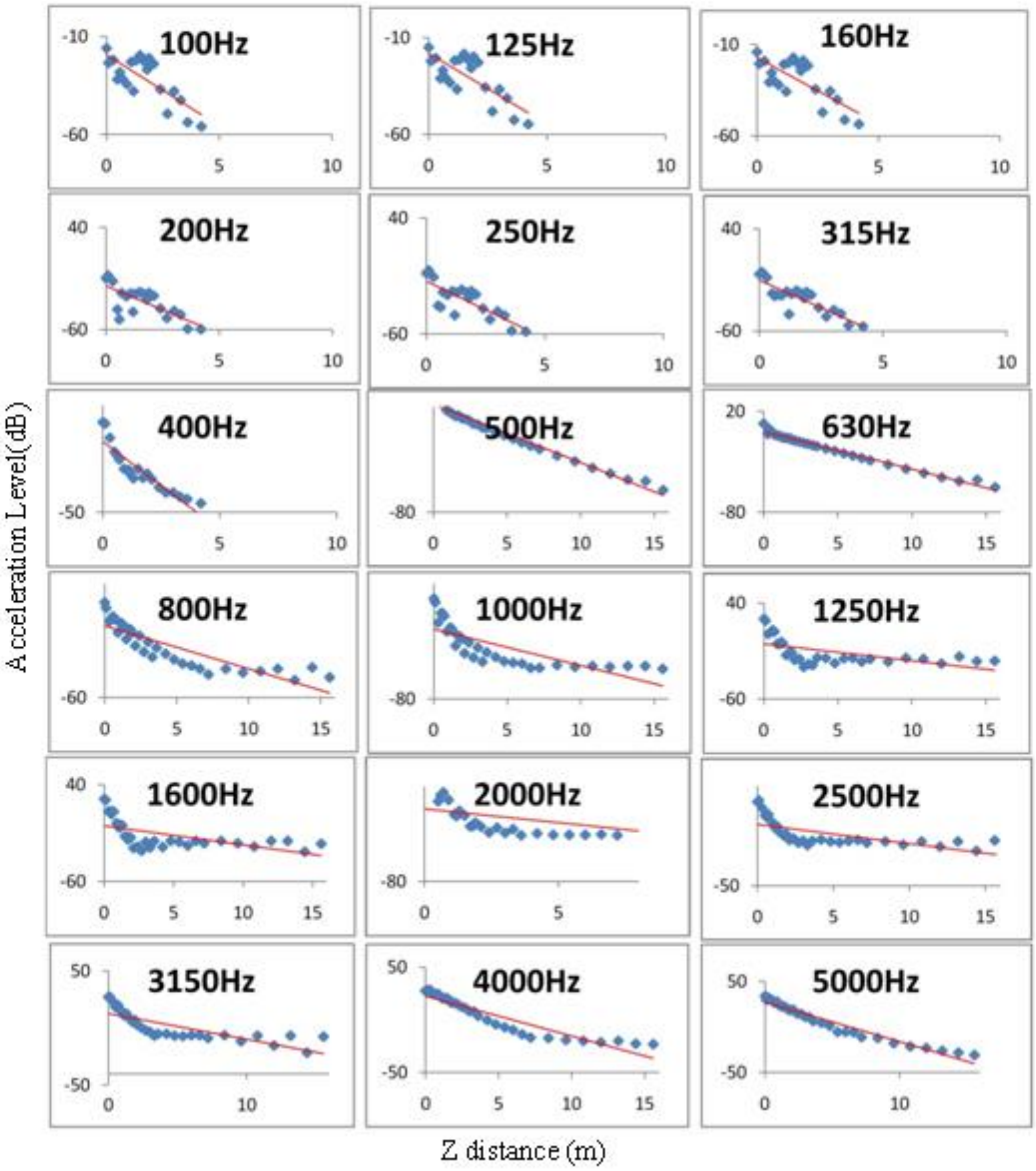

Fig. 6 Calculated vertical response from FE model, from 0 to 16m for each 1/3 Octave-band

particularly in the 800Hz and 1 kHz band where midpoint responses are greater than those at above-sleeper positions.

3. Decay rate with rail absorber

The rail absorber studied consists of steel layer and elastomeric material layer. It is known that the dynamic properties of the absorber will vary with temperature, so that the higher the loss factor, the bigger change in the stiffness over a range of temperature and vice versa (Ahmad 2009). Therefore, the balance between two effects over the range of temperature required. In seeking to achieve the desired material properties over a suitable working temperature range Thompson *et al*. developed the parameters of an elastomeric material that should ideally have the loss factor more than 0.35 and 14 MPa for Young's modulus. So, here these dynamic properties are used for elastomer layer. (Table 3)

To model the rail absorber in ABAQUS, Shell elements are used by means of multi-layer Skin (Fig. 8). The procedure for obtaining response and then calculating decay rates is same as the rail without absorber. Here, firstly, the rail with absorber just on one side is studied. Secondly, the rail with both side absorbers is modeled and then a comparison between their decay rate results has been made. (Fig. 9)

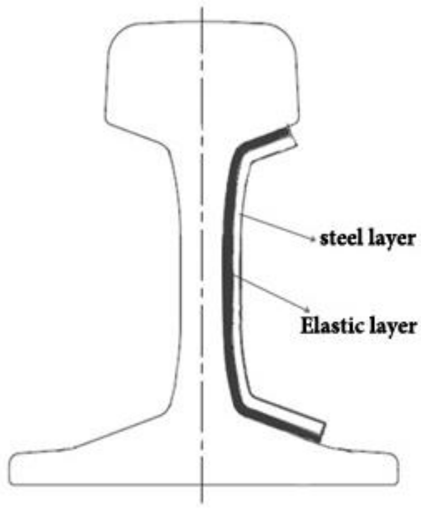

Fig. 7 Continous rail absorber in one side

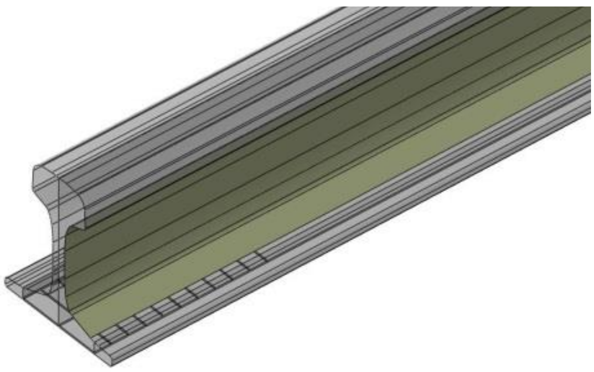

Fig. 8 Rail model with absorber (multi-layer skin) in ABAQUS

Table 3 Rail absorber parameters

	Loss factor damping	Thickness(mm)	E(MPa)
Elastomer layer	.35	1.93	14
Steel layer	0.02	2	$207*10^3$

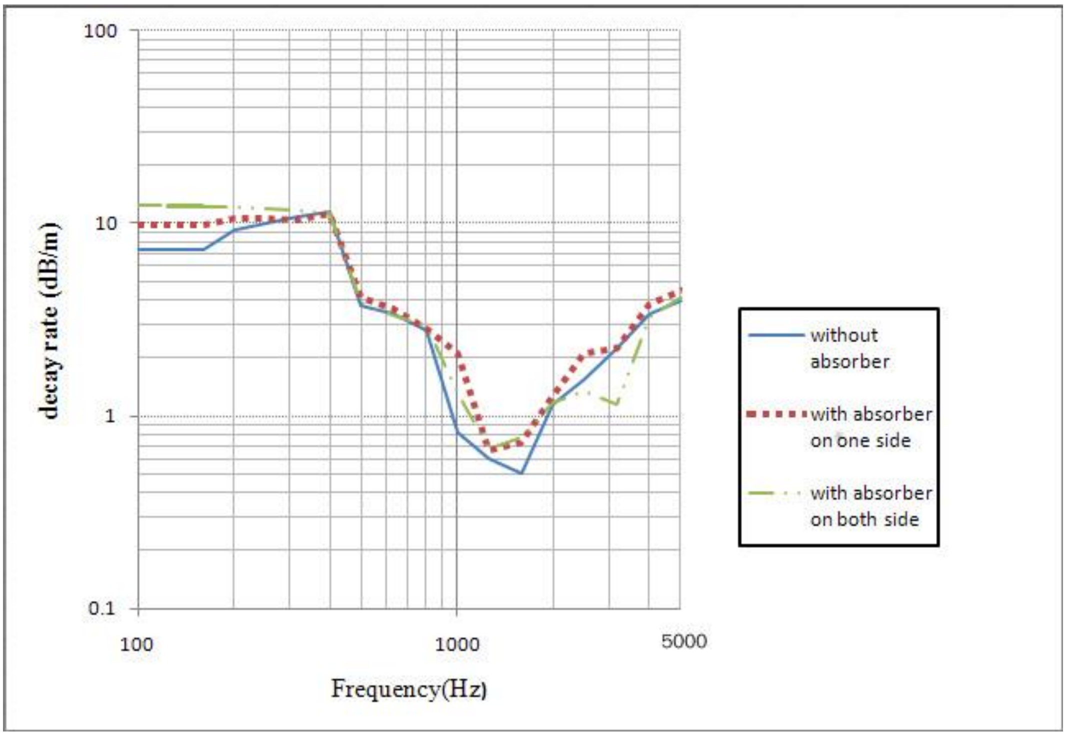

Fig. 9 vertical decay rates from FEM : —the rail with absorber on one side, - - -on both side, _._ the rail without absorber

It is shown in Fig. 9 that the rail with absorber just on one side of the rail increases the decay rate by about 2 dB at low frequency. In case, the rail with both side absorbers, the decay rate increases by 4 dB at low frequency. It can be seen that both, one side and both side rail absorber are particularly effective for low frequency. Moreover, different thickness of elastomer layer and steel layer up to 5mm have been studied but it does not make particular different in results. So this system of absorber even on both side of rail is not effective in noise reduction of the rail. The wave decay rate in the 500-2500 Hz frequency range has to be improved.

4. Improved rail absorber

Here, this absorber system has been improved by adding a tuned absorber bottom of the rail foot between sleepers in length of sleeper bay. The necessary reaction force to reduce the amplitude of the vibrating rail at the two pinned–pinned frequencies is mainly generated by the steel plate. The bending mode shapes are the Eigen modes that give the highest reaction force.

The thickness of steel plate is designed to have two specific eigen frequencies, with bending mode shapes that coincide with the pinned–pinned frequencies of the rail. With the same parameters for plate and its elastomer, the appropriate thickness for plate has been achieved by 5mm with 1.93mm elastomer.

It is known that For UIC60 rails with 0.6 m sleeper bay, the first pinned–pinned vibration Eigen mode occurs at approximately 950 Hz (Maes and Sol 2003). It is called pinned-pinned frequency because at this resonance or Eigen frequency of the railway track, the nodes of the rail's mode shape coincide with the positions of the sleepers. There is zero amplitude of the rail at the sleepers. A second pinned–pinned frequency can be found at approximately 2200 Hz. At this frequency the same phenomenon occurs for a higher harmonic.

The first resonant frequency of the bottom plate separately is tuned to a frequency of 928 Hz

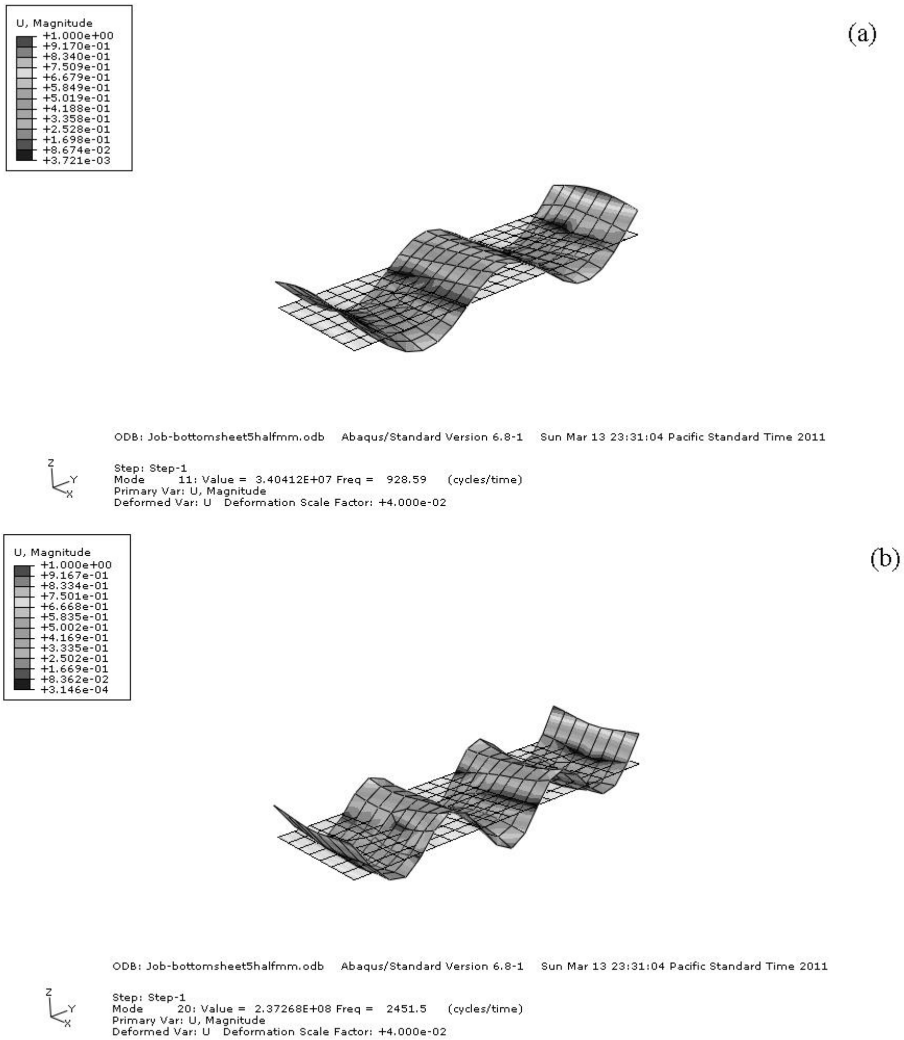

Fig. 10 The bending mode shapes of bottom plate: (a) at 928Hz (b) at 2451Hz

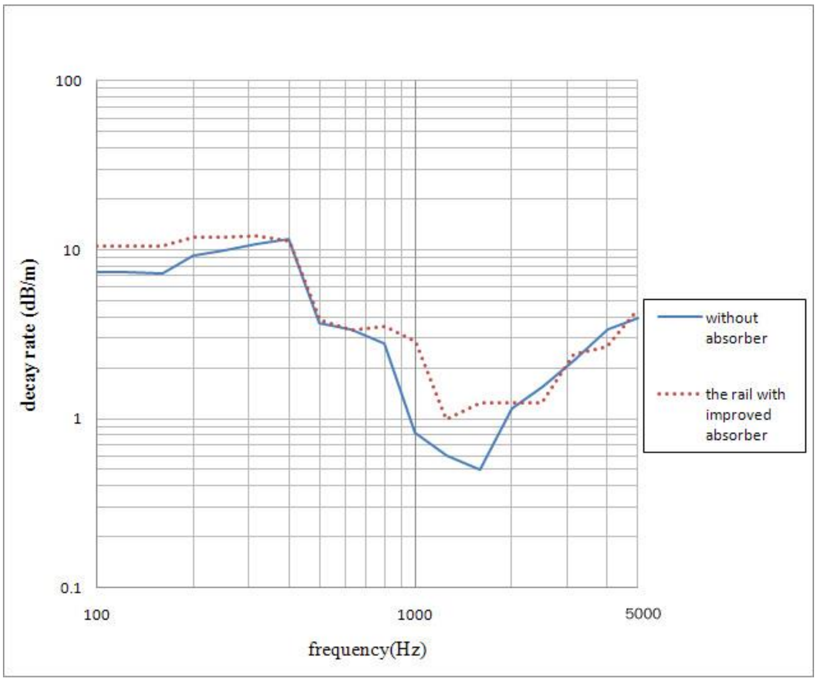

Fig. 11 Vertical decay rates calculated from FEM: —the rail with improved absorber, - - - the rail without absorber

and is associated with a bending mode shape. The mode shape is similar to Fig. 10(a). At 2451 Hz, the second bending mode appears. The Eigen mode, seen in Fig. 10(b), is the bending mode as calculated by FE model.

This bottom sheet is added to the rail with both side absorbers to improve its performance in noise reduction. The vertical decay rates of the treated rail and the rail without absorber are shown in Fig. 11.

Fig. 11 shows that the decay rate increases at pinned-pinned frequency and at low frequency by about 3dB in proportion to the decay rate without absorber. So, a significant improvement of the results is attained in comparison with the rail without absorber.

4. Conclusions

Through the theoretical models and field measurements for rolling noise in the past, it has been shown that the rate of decay of vibration along the rail is closely linked to the noise performance of the track. Therefore, in this paper, the effectiveness of a kind of absorber has been studied by comparing the decay rate results from FEM and then the rail absorber system has improved to be effective in rail noise reduction.

To obtain the response, a periodically supported UIC60 rail FE model using infinite element has been developed for the calculation of decay rates so that, the acoustic performance of different rail absorbers can be studied. The initial design of rail absorber, either on one side of the rail or both side, has been shown that acts as a constrained layer that is just effective at low frequency due to the long structural wavelengths in rails and small surface strains. In second step, the initial rail absorber has been improved by adding a bottom steel sheet with elastomer layer. The Eigen frequencies of the bottom sheet should be tuned to coincide with the two pinned–pinned frequencies considered so that, the vertical reaction force of the bottom sheet was maximal. Therefore, this improved rail absorber increased the vertical decay rate by about 3dB in the 500–2500 Hz frequency range and can be effective in rail noise mitigation.

Acknowledgments

The authors are grateful to Prof. Hecht and Prof. Thompson for their generous support of this work and their guidance and their comments.

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