PROPERTY AND EFFECT ON RAILWAY ROLLING NOISE OF THE TUNED RAIL DAMPER

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A combined railway track model with the TRD (tuned rail damper) is developed to simulate the dynamics. Two criteria are proposed to judge the inherent performance and the practical effect of the TRD on railway rolling noise reduction, respectively. The inherent property of the TRD to reduce the rail vibration and radiation does not vary, but the practical effect may vary with the environmental factors. The model predictions are validated by the measurement in situ. The model developed is useful for analysis of the TRD’s performance on noise reduction. The proposed methodology combining the simulation results with the measurement data can be used to estimate the practical effect on rolling noise reduction of the TRD to be used.

Keywords: tuned rail damper, rolling noise, track dynamics, rail vibration

1. Introduction

The operational noise from the viaduct railway is composed of the structure-borne sound and the rolling noise. The roughness on the wheel and rail treads forms the relative displacement excitation when the wheel and rail come into rolling contact, and causes dynamic interaction and vibration of the wheel, rail and viaduct structure. When the vibration propagates in the wheel, rail and viaduct, they radiate noise. To the overall A-weighted sound the rolling noise dominates, whereas the structure-borne sound is secondary as it is mainly consists of low frequency components, which are dramatically reduced by the A-weighting factors. Moreover, as the operational speed is not very high in the metropolitan transit, the rail radiation is dominant compared with the wheel. Therefore, reducing the rail component is most effective to mitigate the rolling noise in the city metro.

The rail component of rolling noise is mainly in the middle frequency range 500-1500Hz. In this region, the damping introduced by the rail fasteners is low. An alternative way adding extra damping to the rail vibration is the use of the tuned rail damper, which is in fact a vibration absorber and can store and dissipate the rail vibration energy and thus reduce the rail radiation. There are several types of rail damper available and were used or tested in some European railway noise projects, for example, the Corus (now Tata) damper [1], the Schrey & Veit damper [2] and the CDM damper [3], shown in Fig. 1, as well as the Wal damper [4] installed in Hong Kong metro.

Theoretical study on the mechanism of rolling noise reduction using the tuned rail damper (TRD) has been conducted since the last decade. Based on the Corus damper’s prototype, Thompson et al [5] studied the influences of the key parameters of the rail damper on its performance. Wu [6] con-
ducted a comprehensive study on the track dynamics with the tuned rail damper/absorber. Guidelines were given on selection of the types and parameters of the TRD.

![Image of some tuned rail dampers](image1)

Figure 1: Some tuned rail dampers, from left: Corus (Tata), Schrey & Veit and CDM.

As the tuned rail damper functions around its fixed resonance frequency, the practical effect on rolling noise mitigation may vary with the environmental conditions in its use. In some circumstances, a 6-8dB reduction in the overall A-weighted sound level may be achieved by use of the rail damper, whereas in other cases only a 2-3dB reduction is obtained. This will cause inconclusive outcomes for the rail dampers' effectiveness. To judge the inherent performance and the practical effect of the TRD separately, appropriate criteria and methodology should be used.

In this work a new type of the TRD, which is used for reducing the rolling noise of an elevated line in Shanghai metro, is chosen as an example to study. A combined model of the railway track with the rail damper is developed for simulation. Two criteria are used to judge the inherent performance and to predict the real effect on noise reduction, respectively. The predictions are validated by the measurement in situ. The criteria and methodology proposed is useful to assess the performance and effect of the rail damper in the practice of railway noise mitigation.

2. Track modelling with TRD

The un-ballasted slab track is widely used in the city metro in China, where the rails are laid on the concrete bed via the resilient rail fasteners (the rail pad stiffness $\leq 60\text{kN/mm}$). Fig. 2 schematically shows the compound track and rail damper model for vertical vibration. As the component of rail radiation due to the vertical vibration is dominant and usually about 7 dB higher than that due to the lateral vibration [7], the lateral vibration may be ignored for simplicity in studying the dynamic property of the track with the rail damper.

![Diagram of compound track-damper model for vertical vibration](image2)

Figure 2: Compound track-damper model for vertical vibration.

![Diagram of Angel rail damper](image3)

Figure 3: Angel rail damper attached to the rail foot by clips on a slab track.
In Fig. 2 the rail is modelled as an infinite Timoshenko beam on the continuous support composed of the resilient rail pads. The upper parts above the rail represent the discrete rail dampers, which are modelled as a mass-spring layer along the rail. Without losing generality the Angel damper shown in Fig. 3, which is developed and patented by Angel Technical Co Ltd, is chosen as an example to study. Its effect on rolling noise is measured and reported in this work.

The equations of motion for the combined system of the track and rail damper are given in the frequency domain. For the track, it is modelled as a continuously supported Timoshenko beam, 

\[ -\rho A \omega^2 X + GA \kappa (\phi' - X') + k_p X + k_d (X - X_d) = F_r \delta (z), \quad (1) \]

\[ -\rho I \omega^2 \phi + GA \kappa (\phi - X') - EI \phi'' = 0, \quad (2) \]

and for the rail damper, it is modelled as a continuous mass-spring layer without bending stiffness, 

\[ -m_d \omega^2 X_d - k_d (X - X_d) = 0, \quad (3) \]

where \( X \) is the vertical displacement of the rail, \( X_d \) is the displacement of the rail damper, \( \phi \) is the rotation of the cross-section of the beam due to bending, and \( \phi' \) indicates the derivative with respect to \( z \), the longitudinal position of the beam. The material properties of the rail are represented by \( E \), the Young’s modulus, \( G \), the shear modulus and \( \rho \), the density. The geometric properties of the rail cross-section are characterized by \( A \), the cross-sectional area, \( I \), the area moment of inertia and \( \kappa \), the shear coefficient. For the rail support, \( k_p \) is the stiffness of the rail pad per unit length, and for the rail damper, \( m_d \) and \( k_d \) are the mass and stiffness per unit length, respectively. Damping is treated by adding loss factor \( \eta \) to the stiffness in the complex form \( k(1+i\eta) \). The harmonic excitation force \( F_r \) is assumed to apply at \( z = 0 \), the middle of the beam.

3. Dynamic property of the track with TRD

The performance of the rail damper on rolling noise is related to the decay rate of rail vibration [8]. The tuned rail damper is used to increase the decay rate of rail vibration propagation and to decrease the energy and radiation of rail vibration. Decay rate is calculated using the combined slab track and rail damper model in Fig. 2. The parameters adopted in calculation are from CHN60 rail that is similar to UIC60 rail. The rail pad stiffness is chosen to be both 60kN/mm and 20kN/mm, which are often used in Shanghai metro with span length 0.6m. Damping is added via loss factor \( \eta_p=0.15 \) to the rail pad. The rail damper is installed by clips onto the rail foot, as shown in Fig. 3.

Figure 4: Wheel-rail interaction model.

To calculate the rail vibration and sound radiation due to the wheel-rail interaction, the wheel-rail interaction force (to the rail) \( F_r \) is calculated by the following formula, refer to Fig. 4,

\[ F_r = \frac{-R}{\alpha_w + \alpha_c + \alpha_r}, \quad (4) \]

where \( R \) is the roughness excitation spectrum, \( \alpha_w \) is the wheel receptance and \( \alpha_w = -1/(m_w \omega^2) \), where \( m_w \) is the wheel mass (unsprung mass). \( \alpha_c \) is the receptance of the linearized wheel-rail contact spring, \( k_c \), and \( \alpha_c = 1/k_c \). \( \alpha_r \) is the rail receptance, calculated by Eqs. (1-3).

Figure 5 shows the calculation results of the decay rate of rail vibration. The decay rate can be seen to be high at low frequencies as the displacement of rail vibration is large and so is the deformation of the rail pad. At high frequencies, the decay rate is low as the displacement of the rail and thus the pad deformation are small, and therefore, the rail pad dissipates little energy. The decay
rate reaches the highest level around the rail damper’s resonant frequency, and there the decay rate of rail vibration is almost the same for the both tracks with the 20kN/mm and 60kN/mm rail pad.

![Decay rate vs Frequency](image1)

![Rail vibration energy vs Frequency](image2)

Figure 5: Comparison of decay rate of rail vibration. Figure 6: Energy of rail vertical vibration within one car length $L_c=22.8\,\text{m}$ due to $1\,\mu\text{m}$ roughness.

The vertical vibration energy of the rail, $E_v$, which is related to the sound radiation, can be obtained from Eqs. (1-3) if the wheel-rail force is known, and is calculated by the following formula

$$E_v = h_p \Delta z \sum_{L_c} v_r^2,$$

where $v_r$ is the velocity amplitude of rail vibration, $\Delta z$ is the length of a small section of the rail, and $h_p$ is the total projected width of the rail section in the vertical direction, allowing for the top and bottom of the rail head and foot. If the Angel rail damper is installed, a corresponding part should be deducted from $h_p$, as some area at the rail bottom is covered by the rail damper and do not radiate sound. The summation is performed within one car length $L_c$.

The rail vertical vibration energy due to $1\,\mu\text{m}$ roughness excitation is calculated and shown in Fig. 6 for comparison between the two tracks with and without the rail damper. It can be seen that the rail vibration energy is dramatically reduced in the frequency region 500-1200Hz. In addition, the energy reduction is almost the same for the both tracks with 20kN/mm and 60kN/mm rail pad.

The rail radiated sound power, $W_r$, can be estimated if the radiation ratio of rail vibration is known, given by

$$W_r = \rho c \sigma S \langle \overline{v_r^2} \rangle = \rho c \sigma E_v,$$

where $\rho$ is the density of air, $c$ is the speed of sound in the air, $\sigma$ is the radiation ratio of the rail structure, $S$ is the surface area of radiation, $\langle \overline{v_r^2} \rangle$ is the surface-averaged mean-square normal velocity, $\langle \overline{v_r^2} \rangle = E_v / S$, and $E_v$ is calculated by Eq. (5).

### 4. Inherent performance and practical effect of the TRD

The practical effect of the rail damper on rolling noise mitigation in terms of the overall A-weighted sound pressure level may vary with the excitation and environmental conditions, whereas the inherent performance of the rail damper should not vary with these factors. It is therefore necessary to propose a criterion for judgment of the inherent performance of the rail damper on rail vibration and sound radiation.

The proposed criterion here is based on the generation procedure of the rail component of rolling noise, refer to Fig. 7. The original excitation for rolling noise generation is the wheel-rail dynamic interaction caused by the roughness on the wheel and rail treads. According to the diagram in Fig. 7 the generation procedure of the rail radiated noise can be described by the following formula

$$N_r(\omega) = R(\omega) T_{inc}(\omega) T_{rail}(\omega) \rho c S \sigma(\omega) T_{trans}(\omega),$$

(7)
where $N_r$ is the sound pressure spectrum at the receiver, $R$ is the roughness excitation, $T_{\text{inter}}$ represents the transfer function of the wheel-rail interaction force to the roughness excitation, $T_{\text{rail}}$ represents the transfer function of the surface-averaged mean-square normal velocity to the wheel-rail force, and $T_{\text{trans}}$ represents the transfer function of sound propagation from the source to receiver.

$$\frac{N_r}{N_{rd}} = \frac{RT_{\text{inter}} T_{\text{rail}} \rho c \sigma T_{\text{trans}}}{R_{d} T_{\text{inter}} T_{\text{rad}} \rho c \sigma_{d} T_{\text{trans}}},$$

where the terms with subscript $d$ are for the track with the rail damper. The terms $R$ and $R_{d}$, $T_{\text{trans}}$ and $T_{\text{trans}d}$ are the same for the compared tracks, respectively, and thus vanish. The ratio of $N_r$ to $N_{rd}$ also represents the sound power ratio of rail radiation, $W_r$ to $W_{rd}$. As the ratio is only related to the dynamic and radiation property of the track and does not vary with the external factors (roughness input and propagation path), it can be used as a criterion to judge the inherent performance of the rail damper. Taking logarithm of the sound power ratio gives

$$L_{\text{reduct}} = L_{N_r} - L_{N_{rd}} = L_{W_r} - L_{W_{rd}} = 10 \log_{10} \left( \frac{W_r}{W_{rd}} \right),$$

where $L_{\text{reduct}}$, which is a function of the frequency, e.g. in the third octave band, represents the inherent performance of the rail damper on reduction of the rail radiated noise in dB.

The ratio of sound power may be further simplified for the Angel rail damper shown in Fig. 3. As the radiation ratio of the rail with and without the rail damper is very close, and the surface area of rail radiation is partly covered by the rail damper installed, the sound power ratio can be simplified to

$$\frac{W_r}{W_{rd}} = \frac{T_{\text{inter}} T_{\text{rad}} S}{T_{\text{inter}} T_{\text{rad}} S (S - A_d)},$$

where $S$ is the radiation surface area of the rail and $A_d$ is the surface area covered by the rail damper.
calculated using Eqs. (1) and (6). The track parameters used in the calculation are given in section 3. Two rail pad stiffness is considered, 60kN/mm and 20kN/mm. The wheel mass (unsprung mass) is chosen to be \(m_w = 600\)kg and the linearized wheel-rail contact stiffness \(k_c = 1.14\)MN/mm. The rail radiation ratio used is from [9] for UIC60 rail and shown in Fig. 9, which is similar to CHN60 rail. The running speed of the train is chosen to be 65km/h, i.e. about 18m/s.

The calculation results are shown in Fig. 10 for the two roughness inputs in terms of the A-weighted sound power spectra radiated by vertical rail vibration in the 1/3 octave frequency band. For the corrugated rail the sound power can be seen to reach a peak at 800Hz corresponding to the peak at roughness wavelength 25mm in the 1/3 octave wavelength band. By use of the TRD the peak at 800Hz is dramatically suppressed. Consequently, as the contribution of the 800Hz component dominates, the overall A-weighted sound power level is effectively reduced by 8.6dB. For the roughness input of the ISO limit, the overall A-weighted sound power level is only reduced by 4.3dB. This is because the contribution from other bands above 1kHz becomes dominant, although the 800Hz component is also dramatically reduced by the TRD. The calculation results demonstrate why the practical effect on noise reduction may vary with the environmental conditions in terms of the overall A-weighted sound power level.

On the other hand, the logarithm of the sound power ratio of rail radiation in Eq. (9) will not vary with the environmental factors in each frequency band, and thus it can be used to represent the inherent performance of the TRD on noise reduction. The calculation results for \(L_{\text{reduct}}\) are shown in Fig. 11 and are identically obtained from the curves either in (a) or in (b) in Fig. 10. It can be seen that the ability of the TRD to reduce the rail component of rolling noise reaches the highest around 600-800Hz, but away from these bands the reduction comes down quickly. In addition, the performance on noise reduction of the TRD is similar for the both tracks with 60kN/mm and 20kN/mm rail pad, even identical in the frequency region above 400Hz.

Figure 10: A-weighted sound power level radiated due to vertical vibration for (a) corrugated rail and (b) ISO limit:
- 60kN/mm no TRD, - 60kN/mm with TRD, - 20kN/mm no TRD, - 20kN/mm with TRD

Figure 11: Sound power reduction \(L_{\text{reduct}}\) of rail radiation by TRD.
5. Measurement results

The Angel rail damper has been installed at a curved viaduct section for reducing the operational rolling noise in Shanghai metro line 11, which is terminated at the Disneyland in Pudong district. The section installing the rail damper is about 300 meters long, refer to Fig. 12. The rail damper is only applied to the low rail, shown in Fig. 3, because the short pitch corrugation is developed at the low rail, see in Fig. 13, and it rarely happens at the high rail. The wavelength spectrum of the rail corrugation developed is given in Fig. 8, which is used in the predictions in section 4.

![Image](image1.png)  
**Figure 12:** Scene of measurement at a curve with TRD.  
**Figure 13:** Corrugated rail at the curved section.

The passing noise was measured during the train operation by two steps: before and after installation of the TRD. The monitor point of the noise is about 140m (open field) away from the viaduct, refer to Fig. 12, and the train speed is about 65km/h. The rail pad stiffness of the track there is about 40kN/mm and there is a sound barrier set on the viaduct wall along the track towards the monitor side. The measurement results are shown in Fig. 14 in terms of the A-weighted sound pressure level in the 1/3 octave band.

![Image](image2.png)  
**Figure 14:** A-weighted sound pressure level of passing noise in 1/3 octave band at train speed 65km/h: — measured before TRD installed, --- measured after TRD installed, --- estimated with TRD applied.

It can be found from Fig. 14 that the components in the 500-1000Hz bands are dominant before installation of the TRD, and mainly consist of the rolling noise. The components below 250Hz and above 2500Hz are secondary and their contribution to the overall A-weighted sound level may be neglected. The noise peak appears around 630-800Hz, corresponding to the 25mm wavelength corrugation and the train speed 18m/s (65km/h). The peak can be seen to be effectively suppressed by use of the TRD. A significant 8dB reduction at the monitor point (140m away the noise source) is achieved by the TRD via reducing the rail component of rolling noise. However, if the rail is not severely corrugated, the rolling noise reduction will not be as effective as in this case, refer to predictions in section 4.
Another curve in Fig. 14 (dotted line) is obtained by subtracting the data of the performance curve in Fig. 11 from the data of the sound pressure level measured without the TRD. This curve can be used as a prediction of the practical effect on noise reduction for the TRD. The estimated effect on noise reduction is about 9.4dB. Though the reduction is over-estimated by about 1.4dB compared with the measurement result 8dB, it is acceptable and can be used as an approximate prediction for the practical effect of the TRD to be used on noise reduction.

6. Conclusions

The practical effect on rolling noise reduction of the TRD may vary with the environmental factors in terms of the overall A-weighted sound pressure level, whereas the inherent performance of the TRD does not change. A track model combined with the TRD is developed to analyse its performance and to predict its practical effect on noise reduction. The inherent performance of the TRD is obtained by comparison between the tracks with and without the TRD in terms of the rail radiated sound power. The TRD’s effects on noise reduction are predicted by model simulation. The simulation results demonstrate that the effects are inconsistent under different roughness excitations. Measurement in situ is conducted to testify the results of model simulation. The measurement data show that the Angel rail damper can effectively reduce the passing noise at a curved viaduct section by about 8dB for the corrugated rail, which agrees with the simulation results of 8.6dB reduction in the rail radiated sound power. Finally, a methodology is proposed to estimate the practical effect of the TRD on noise reduction by combining the measurement data of the passing noise with the simulation results of the rail damper’s performance.

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