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Estimating the performance of rail dampers using laboratory methods and software predictions

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Summary

Rail dampers are designed to reduce the rail component of rolling noise by increasing the attenuation with distance along the rail (decay rate, DR). There is no standardized method to assess the performance of rail dampers. The method described here, developed during the Franco-German STARDAMP project, uses laboratory tests and computer simulation to avoid the need for expensive and time-consuming field trials. The premise of the method is that the DRs of a damped track can be found from summing the DRs of a short-section of damped 'freely supported' rail and the DRs of an undamped track. Reasonable predictions of the decay rates of a test track have been made using this method. Software has been produced that implements TWINS-like predictions of rolling noise with and without rail dampers to predict the damper effect. The effect of rail pad stiffness on the effectiveness of rail dampers has been considered for track constructions typical in the UK and a regional train travelling at 120 km/h. For track fitted with 'soft' 120 MN/m rail pads, the dampers are predicted to reduce the total level by 2.5 dB(A) while with the 'stiff' 800 MN/m pads a 0.7 dB(A) reduction is expected.

1 Introduction

The noise radiated by the rail is usually the dominant source of rolling noise between 0.5 and 2 kHz and often in terms of overall level [1]. Rail dampers are now commercially available that are designed to reduce the rail component of noise by increasing the attenuation with distance along the rail (decay rate, DR) and hence reduce the radiating length. These dampers, tend to be bolted or clipped onto the rail between sleepers and work on the principle of tuned mass dampers [2-4]. There is no standardized method to assess the performance of rail dampers. Railways are often obliged to undertake line testing which can be expensive and may lead to results which are ambiguous or difficult to generalise.

Two methods for determining damped track DRs were tested in the STARDAMP project. With both methods, the damped track DRs are found by summing the DRs of an undamped track on which the dampers are intended to be fitted and the DRs of a section of freely supported damped rail [5]. With the first method, the damped free-rail DRs are determined for either a 4 m or 6 m length of damped rail at low frequency from the modal properties of the rail, and at high frequencies directly from point and transfer frequency response functions (FRFs) at either end of the rail [2,5]. In the second method, the damped free-rail DRs are determined from FRFs measured at intervals along a longer (e.g. 32 m) rail using a method similar to the track decay rate measurement standard EN15461:2008 [6]. The two methods for determining DRs of damped 'freely supported' rails showed reasonable agreement between 300 Hz and 5 kHz. The modal method for determining DRs on the 'short' rail was restricted to low frequencies (< 300 Hz) and resulted in much lower rates than those measured on the 'long' 32 m rail. With dampers designed for conventional track, below 400 Hz the damper DRs are relatively low and tend to have little influence on overall track DRs. Consequently, the direct short-rail method, yielding plausible measurements down to 300 Hz, was considered to be sufficient for many applications. The method is summarized in Section 2; further details can be found in [5,7].

The in-situ performance of dampers will depend not only on their effect on the track DRs but also on the relative contributions of the wheels and individual track components to the radiated noise. These contributions might be predicted (e.g. using TWINS [8]), however currently available software require a large number of input parameters and considerable expertise of the user. An aim of the Franco-German STARDAMP project was to develop a more user-friendly method to predict the acoustic performance of rail and wheel dampers. The method described in Section 3 uses laboratory tests and computer simulation and avoids the need for expensive and time-consuming field trials. The application to wheel dampers is described in a comparison paper [9]

2 Decay rate measurements

The premise of the 6 m rail method developed within STARDAMP is that the DRs of a damped track can be found by summing the DRs of a damped 'freely supported rail and the DRs of an undamped track. The damped free-rail DRs are derived from the attenuation measured along a 6 m length of rail.

2.1 Damper decay rate test procedure

The proposed damped free rail test procedure is outlined below. To demonstrate the method, example results are given for Schrey and Veit (S&V) rail dampers mounted on UIC 60 rail. Each S&V damper consists of two 7.0 kg laminated rubber and steel construction absorber masses bolted on to the rail web via a solid steel base plate (2.8 kg). The total mass of each damper assembly is 18.6 kg. Test conditions specific to this example are given in parentheses. Other dampers were tested within the project with broadly similar results [7].

With the method, dampers are installed symmetrically over the whole length of a 6 m rail (UIC 60) at a centre-to-centre spacing representative of the intended track installation (see Fig. 1). The rail should be 'freely suspended' at either end on a foundation that is soft enough so that the bounce mode has a natural frequency less than 30 Hz (12 rubber rail pads were used at either end of the rail, giving a bounce mode ≈ 20 Hz). Miniature accelerometers are rigidly attached (using a thin layer of beeswax) as close as possible to either end of the rail (5 mm), attached either at the centre of the rail head for vertical measurements or on the side of the rail head for lateral measurements. A small instrumented hammer, with a hard (titanium) tip, is used to excite the rail with a force of approximately 400 N. This was adequate to ensure that the force spectrum is flat up to high frequencies, dropping by less than 20 dB by 7 kHz.

For both lateral and vertical measurements, a point FRF at one end and a transfer FRF to the other end is measured. The rail temperature should be controlled between 18 and 25°C during the tests. Further measurements are recommended at temperatures encompassing the in-situ temperature range. It is also recommended to measure more than one sample of rail fitted with a given type of rail damper in order to check variability.

In each one-third octave band, the DR is determined as the decibel difference of the transfer FRF to the point FRF divided by the rail length. With low DRs, the % error in the DR for a given dB error in the FRFs is large and therefore in practice the lower threshold for reliable measurements is found to be ~1.0 dB/m.

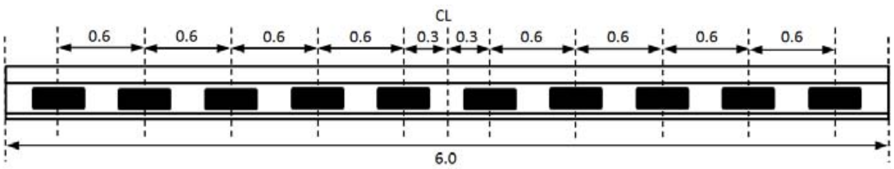

Fig. 1. Example of damper installation with 0.6 m spacing. Dimensions in metres.

2.2 Track decay rates measurements

To demonstrate the method, track DRs of an undamped track were measured on a 32 m test track at the University of Southampton using a procedure based on EN 15461:2008 [6]. In practice these measurements would be made on a circulated track. The test track has UIC 60 rail, 51 concrete monobloc sleepers with a mean spacing of 0.63 m (s.d. = 0.03 m), Pandrol Fastclips, Pandrol 10 mm studded natural rubber pads (effective stiffness approx. 120 MN/m), and granite ballast to depth of 0.3 m.

For the prediction of the damped track DRs, measurements were made of the DRs of the undamped test track. Additionally, for validation, damped track DRs were measured directly using the same EN 15461:2008 procedure, with the dampers bolted on at mid span along the full length of the rail, except at inter-sleeper positions 18 and 37 where rail welds prevented their attachment.

Vertical and lateral DRs were measured with the method. A measurement grid was marked up from a reference point 10 sleeper spans (5.96 m) from the rail end. Measurements were made at ¼-sleeper intervals from this point up to the 16th sleeper span, then at mid-span positions 17, 18, 20, 22, 26, 30, 34, 38, 42 and 46.

An instrumented hammer was used to excite the rail at each of the measurement points in turn. The response was measured with an accelerometer mounted at the reference point.

DRs in each ⅓ octave band up to 5 kHz were calculated in dB/m from the point frequency response function (FRF) at the reference point, $A(x_0)$, and the transfer FRFs, $A(x_n)$, between the reference position and the other points on the measurement grid, x_n, using:

$$DR = 4.343 \bigg/ \sum_{x=0}^{x_{\max}} \frac{|A(x_n)|^2}{|A(x_0)|^2} \Delta x_n \tag{2.1}$$

The derivation of this equation can be found in [5].

2.3 Decay rate results

Vertical DRs for the undamped track, a free 6 m rail fitted with the dampers and the damped track are shown in Fig 2. For the undamped track, at low frequencies, there is high attenuation because of the stiffness of the foundation. At around 250 Hz there is a broad peak associated with the sleeper and rail pad acting as a 'dynamic absorber'. Above around 500 Hz, waves begin to propagate freely in the rail and the DR decreases, before increasing again to a peak at around 5 kHz, caused by a flapping mode of the rail foot [1]. Measurements in the lateral direction showed similar trends (Fig. 3). One difference was that the undamped lateral track DRs were, at most frequencies, much lower than in the vertical direction. The lower lateral rates explain why, while the excitation is generally lower in the lateral direction, its contribution to overall noise levels can be of significance. In both directions, the damped 'free' rail DRs show that the dampers introduced high attenuation in the region 0.5 to 3 kHz.

Damped track DRs have been predicted by summing the damped 'free' rail DRs with those of the undamped track. These show reasonable agreement with the directly measured DRs of the damped track. Some of the inaccuracies in the predicted DRs are likely to have been caused by temperature variations between conditions affecting the pad and damper properties and end effects due to the finite rail lengths e.g. [5,7,10].

The expected reduction in noise from the rail in each ⅓ octave band, from installing dampers, ΔL, can be calculated from the undamped track decay rate, DR_u and the damped track decay rate, DR_d according to

$$\Delta L = 10\log_{10}(DR_u / DR_d) \tag{2.2}$$

To calculate improvements to the overall sound level, predictions are required of the contributions of the individual track components, with and without the dampers, for which the software described in the next section is intended.

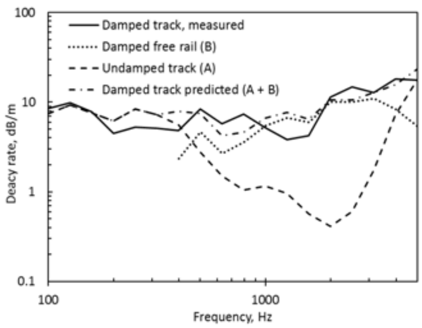

Fig. 2. Vertical decay rates

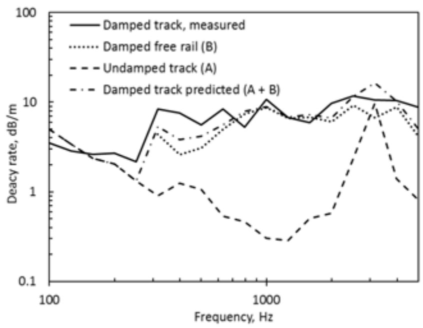

Fig. 3. Lateral decay rates

3 Stardamp software

3.1 Description of software

The software tool, developed within the STARDAMP project, is based on the same theoretical models used in the TWINS software [8]. It implements an analytical description of the wheel-rail interaction where the contact forces are calculated as the ratio between the wheel-rail roughness spectrum and the sum of rail, wheel and contact mobilities. Both vertical and lateral degrees of freedom at the contact are considered. From the contact forces, wheel, rail and sleeper responses are calculated and the sound power levels estimated through radiation efficiencies. If rail dampers are to be included their effect is accounted by replacing analytically calculated rail wavenumbers with measured DRs. Finally a simple model for acoustic propagation above a partially absorptive flat ground gives the sound pressure levels at specific field positions.

Vertical and lateral rail mobilities are calculated by a model of a Timoshenko beam [11] on a double layer continuous elastic support, which accounts for pads, sleepers and ballast. Coupling between vertical and lateral motion is empirically modelled by a constant factor (normally between -7 dB and -12 dB). To define the track, several combinations of track types, sleeper types, rail types and pad stiffness and damping values can be selected. Most importantly, the track can be ballasted or slab-track, in this second case the continuous elastic support has a single layer only. For ballasted track the sleeper can be monobloc (concrete or wooden) which are modelled as beams or bibloc which are modelled as masses. The software can determine DRs analytically from the track response or use measured values. When measured DRs are used, all the other wheel and track-related quantities (e.g. mobilities and contact forces) are retained from analytical calculations and are assumed not to be modified by the presence of dampers.

The wheel is described in terms of a Finite Element (FE) model. This is used to compute natural frequencies and mode shapes at the contact point and at a limited number of positions on the external face. This information is stored in an external text file (modal parameters file) which is loaded in the software; wheel mobilities are then calculated through modal summation and modal damping ratios can be added either adopting standard values or after measurements. Modal models of three typical undamped wheels of freight, regional and high-speed trains are implemented in the software. The user can also include their own.

Typical roughness spectra corresponding to wheels with cast-iron brake blocks, K-block brakes and disc brakes are supplied; again measured values can be loaded by the user. Generally, the number of accessible options is reduced with respect to TWINS in order to permit the use by non-expert users through a simple Graphical User Interface. Lastly, to increase reliability, the final results shown are an average over three contact positions: the nominal one (70 mm from flange back) and ± 10 mm from this.

The software permits the direct assessment of rail dampers, wheel dampers, or a combination of both. In this paper only the application of rail dampers is discussed; wheel dampers are discussed in [9]. When the software is used for assessing dampers, it first computes pass-by noise levels for a baseline model without dampers then it estimates noise levels considering the dampers. The effectiveness can be then visualised by comparing damped versus non-damped sound pressure spectra and overall levels.

3.2 Example predictions

To illustrate the Stardamp software, the effect of dampers on noise from a train pass-by has been predicted for two different track conditions typical in the UK. For the first case, ‘soft’ 120 MN/m rail pads are assumed, while in the second case, stiffer 800 MN/m pads are assumed. Other track parameters were selected to be consistent with the test track (see Section 2.2). For both cases, a regional train travelling at 120 km/h with roughness representative for disc brakes has been assumed. The decay rates measured on the short rail (Figs 2 and 3) have been used as input to the software, along with measured track decay rates applicable to each pad stiffness.

Fig. 4 gives the predicted noise levels for a receiver at 7.5 m from the centre of the track fitted with soft rail pads. It can be seen that the noise contribution of the rail is dominant in the mid frequency region, wheel noise is the main source at high frequency while the contribution of the sleepers is at a much lower level. There is a substantial reduction in the rail contribution after introducing the rail dampers (solid lines), giving an overall reduction of 6 dB(A) in this component. There is also some reduction in the sleeper noise but this component is relatively low compared to the others and has minimal effect on the overall level. There is no reduction predicted in the wheel component of noise. This is a consequence of the fact that the contact forces in the model are not modified by the introduction of dampers on the track. The overall noise is reduced by about 2.5 dB(A).

Fig. 5 gives the predicted noise levels for the track fitted with stiff pads. The higher stiffness of the rail pads decreases the rail component of rolling noise but conversely increases the noise radiated from the sleeper due to the increased coupling (compare Fig. 4 and Fig. 5). As the decay rates are initially higher, the damper only reduces the rail component by about 3.5 dB(A). As a result of the lower rail contribution, the wheel noise dominates the overall noise level and hence the effect of the damper on the overall noise is relatively small at 0.7 dB(A).

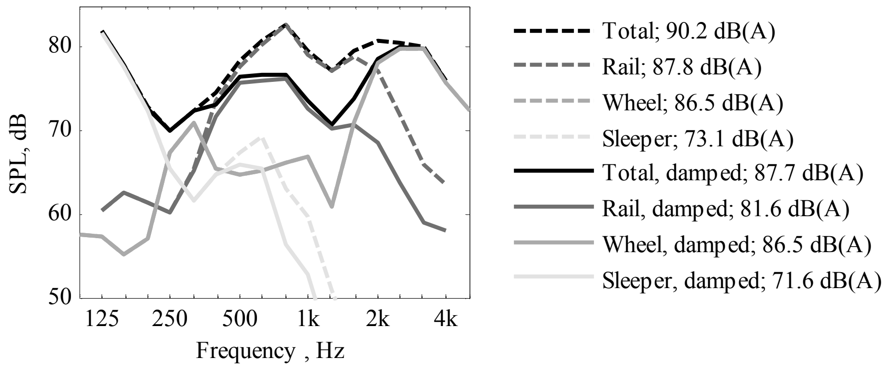

Fig. 4. Sound pressure levels determined at 7.5 m from track fitted with soft rail pads.

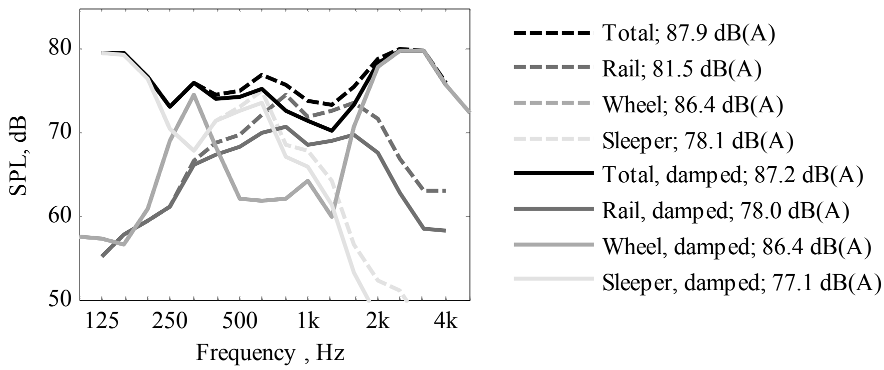

Fig. 5. Sound pressure levels determined at 7.5 m from track fitted with stiff rail pads

4 Conclusions

A combined experimental-numerical procedure for determining rail damper effectiveness without the need to mount them on the track has been proposed and demonstrated. The method consists of measuring the DRs of a short section of freely supported rail equipped with dampers and the DRs of the real track where the dampers are intended to be fitted. The DRs are then used as inputs in rolling noise prediction software which compares noise radiated from the wheel and track, with and without rail dampers. Reasonable predictions of the damped track DRs of a test track have been obtained using the method. Predictions demonstrate that fitting dampers to track with soft pads is likely to be more effective at controlling noise than fitting them on a track with stiff pads.

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