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Experimental procedures for testing the performance of rail dampers

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A B S T R A C T

Rail dampers work by increasing the attenuation with distance of vibration transmitted along the rail, a quantity known as the track decay rate. Currently, there are no standardized procedures to measure their effectiveness in reducing rolling noise without the need for in-track installation and time-consuming tests. This paper describes and evaluates experimental procedures for assessing rail dampers. Instead of field measurements it is proposed to use laboratory measurements of vertical and lateral decay rates on a free rail equipped with dampers. These are combined with in-situ measurements on an undamped track. The decay rates of a damped track can be approximated by adding the results of the damped free rail to those of the undamped track.

Three different methods are studied to measure the decay rates of damped free rails: (i) using a long rail, in the present work 32 m long, from frequency response functions measured at intervals along the rail; (ii) using a short rail, in the present work 6 m long, from the modal properties of the rail; and (iii) directly from the point and transfer frequency response functions at both ends of the short rail. The latter two are complementary: the modal method is more suited to low frequencies while the direct method is more suited to high frequencies. These methods are evaluated theoretically and by comparison with experimental results.

Good agreement is found between the various methods, for vibration in both vertical and lateral directions, between 300 Hz and 5 kHz. In practice, the direct short-rail method is likely to be sufficient for most applications. The limitations of the methods are identified and corrections are proposed for the effect of near-field waves in the rail.

1. Introduction

In most situations, at least for conventional speeds, rolling noise originating from the interaction of the wheel and rail is the dominant source of environmental railway noise. It is caused by the combined surface roughness of the wheel and rail, which excites vibration of the wheel, rail and sleepers. In turn this vibration radiates sound. Models for this phenomenon have been developed and validated against field measurements [1–4]. Generally, the sleeper is found to be the major source of noise below around 400 Hz, the rail is dominant between 400 and 1600 Hz and the wheel is of increasing importance at higher frequencies [5]. Both the track and the wheels can contribute significantly to the overall noise level. Their relative

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contributions depend on the roughness spectrum, the train speed and various design parameters of the wheel and track. Particularly important is the track decay rate, the rate of attenuation of vibration along the rail, which governs the sound radiated by the rail [6,7]. In order to separate the wheel and track components, either advanced experimental techniques [8] or theoretical models [1,2] are required.

Theoretical models are also important to facilitate the development of appropriate noise-mitigating solutions. Increasing the stiffness (and damping) of the rail pads, fitted between the rail and sleepers, can decrease the rail component of rolling noise by increasing the track decay rate and hence reducing the effective radiating length of rail. Conversely this can increase the noise radiated from the sleeper due to the increased coupling [9]. Among the various alternatives developed in the last decades [10] this paper focuses on rail dampers, which are a good and efficient candidate for reducing the noise at source [11]. Rail dampers are particularly effective when tracks are fitted with soft pads (low strain stiffness below 250 MN/m).

Various forms of rail dampers have been designed to increase the track decay rate without the need for higher pad stiffness [12–15]. Usually, such dampers are bolted or clipped onto the rail between the sleepers. They are designed to be effective primarily for the vertical vibration of the rail as, for a wheel rolling on a straight track, this is the most important direction of excitation. However, as the vertical and lateral directions are coupled, and the lateral decay rates are often lower than the vertical ones, the lateral vibration can also contribute to the noise and this has to be accounted for in a numerical model for noise radiation. Overall reductions in noise level ranging from 2 to 6 dB(A) have been obtained in comparative measurements of various types of damper installed on a track [15–17]. Reductions are greatest for a track with softer rail pads where the rail component of rolling noise is more dominant due to the low initial decay rates [17], whereas noise reduction on tracks equipped with stiff pads can be as low as 1 dB(A).

Currently, there is no standardized method to assess the in-situ performance of dampers. Such an assessment would require field tests following their installation in a track, which can be expensive to carry out. An international project, STARDAMP [18], was therefore set up with the aim of developing standard procedures to verify wheel and rail damper effectiveness without the need for full installation in the field.

The aim of this paper is to present the theoretical basis behind the method developed to predict the acoustic performance of rail dampers and to evaluate it in order to establish its range of applicability. This method consists of a combined experimental–numerical procedure and can be summarised in the following steps. First, measurements are made of the decay rates of a length of free rail fitted with rail dampers. Second, those of the undamped target track are measured. To estimate the decay rates of the damped track, the two results are added. Finally, the track decay rates of the damped and undamped cases are used in a prediction model to determine the acoustic effect of the dampers. This is expressed in terms of the reduction in acoustic power radiated by the track or the overall noise reduction for a given wheel design. The advantages of this approach are that it avoids expensive field tests and that the results can be used for different target situations, including different track designs, based on the same laboratory test results.

A requirement of the proposed method is an accurate and repeatable procedure for determining the decay rates of a free rail fitted with dampers. These have previously been obtained from measurements of a 4 m section of freely supported rail [15]. These were determined at low frequency from the modal properties of the rail, and at high frequency directly from point and transfer frequency response functions (FRFs) at either end of the rail. The decay rates of a damped free rail can also be derived from FRFs measured at intervals along a longer section of rail, similar to the track decay rate method described in EN 15461:2008 [6]. However, this approach has not previously been tested for free rails. In this paper, methods based on the three experimental approaches outlined above are tested, compared and critically evaluated.

The expected reduction in acoustic power from the track in each one-third octave band, ΔL, can be calculated from the undamped track decay rate, DR_u and the damped track decay rate, DR_d, according to [5,7]

$$\Delta L = 10\log_{10}\frac{\mathrm{DR}_u}{\mathrm{DR}_d} \tag{1}$$

However, the main assumptions behind Eq. (1) are that there is only one wave propagating in the rail and that three-dimensional effects on the rail radiation ratio can be neglected (see [19]). The in-situ performance of dampers will depend not only on their effect on the rail noise but also on the relative contributions of the wheels and individual track components to the overall noise. These contributions can be predicted using models such as TWINS [1]. A software tool that implements TWINS-like predictions of rolling noise has been developed by the authors within the STARDAMP project [20,21].

2. Measurements of track decay rate: in situ, test track and ‘long rail’ methods

2.1. Method based on EN 15461

Track decay rates are measured in the field using a method described in EN 15461:2008 [6]. This relies on measurements of the driving-point frequency response function (FRF) at a reference point $A(x_0)$, and the transfer FRFs, $A(x_n)$, between the