Estimation of track decay rates from laboratory measurements on a baseplate fastening system

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Abstract. The dynamic properties of the rail fastening system affect the damping of the rail and the degree of coupling between the rail and the foundation. It is well known that the rate of decay of vibration along the rail is closely linked to the noise performance of the track. For this reason, the track decay rate (TDR) is used as an important measurable input quantity for models that predict railway rolling noise. This paper investigates whether the TDR can be estimated correctly from laboratory measured dynamic properties of rail fastening systems. The system studied in this work is a commercial two-stage baseplate system that is mounted on a slab track fitted with 60E1 rail. Four different types of rail pads were fitted during the track decay rates measurements. The TWINS model was used to predict the rolling noise using the measured and calculated track decay. The comparison has shown a good agreement between the measured and predicted TDR.

Keywords: Track decay rates, Railpad, Baseplate, Fastening systems.

1 Introduction

When structural waves propagate along the track they decrease in amplitude with distance from the excitation point. This is quantified by means of the Track Decay Rates (TDR) that express the amplitude decrease in dB/m. They are usually represented in the form of one-third octave band spectra. They can be obtained by means of calculations or they can be measured on track according to standards [1-3]. TDRs are also used in noise prediction models to evaluate the noise emission from the track.

The static and dynamic stiffness of the resilient elements on the railway track play a key role in determining the noise radiated by the track and the vibration isolation [2]. While the lateral stiffness and roll stiffness of the fastening system are of interest, it is usually the vertical stiffness that is of main interest. The vertical stiffness is also important in relation to track deterioration and maintenance requirements. To predict noise, it is the stiffness at higher frequencies and small strains that is required. The dynamic stiffness of resilient elements can be measured in several ways. In this work, the indirect method is adopted for frequencies up to 1 kHz [4].

1.1 Track description

Measurements of track decay rate for vertical and lateral excitation were made on a non-operational slab track located at the National College for High-Speed Rail, Doncaster. The slab track has two adjacent sections (20 m long), one fitted with Pandrol rail fastening system as seen in **Fig. 1** and the other with another type of rail fastening system (not addressed here). For the Pandrol two-stage system, measurements were made with four different types of railpads fitted. **Fig. 2** shows a schematic view of the track sections.

Fig. 1. Photographs of the Pandrol two-stage fastening system.

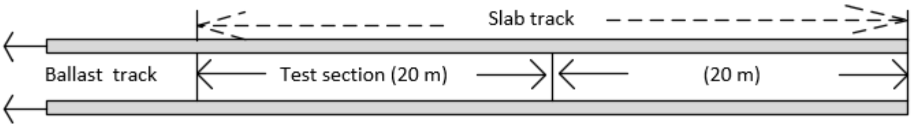

Fig. 2. A schematic view of the sections of the measured track.

2 Methodology

2.1 Dynamic stiffness

There are several methods for measuring the stiffness of track resilient elements. In [5, 6] an indirect method was established, which will be used here. The indirect method involves a test rig with two blocks of known mass; a resilient element can be inserted between them as shown in **Fig. 3**. The lower mass is mounted on flexible isolators. A hydraulic actuator applies a static preload and a high frequency excitation is applied by an inertial shaker. This allows the high-frequency transfer stiffness of resilient elements to be measured as a function of frequency in the low amplitude region. The frequency dependent dynamic stiffness and the damping properties of the resilient element can be obtained using the following equation:

$$K = -m_2\omega^2\frac{\ddot{x}_2}{\ddot{x}_1}\ ;\ \eta = \tan(\angle K) \tag{1}$$

where K is the transfer stiffness, η is the loss factor, m_2 is the mass of the lower block, $\ddot{x}_1$ is the acceleration of the upper mass, $\ddot{x}_2$ is the acceleration of the lower mass and ω is the circular frequency in rad/s.

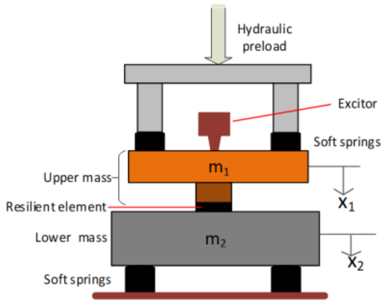

Fig. 3. Test rig for measuring the high-frequency dynamic stiffness of resilient elements [5].

2.2 Track decay rates measurements

TDR measurements were conducted in accordance with EN15461:2008 [1]. The test requires an accelerometer to be attached to the railhead and series of transfer functions to be measured between an input force, delivered with an instrumented impact hammer, and the measured acceleration. The TDR in each one-third octave band is obtained as

$$DR \approx \frac{4.343}{\sum_{n=0}^{n_{max}}\frac{|A(x_n)|^2}{|A(x_0)|^2}\Delta x_n} \tag{2}$$

where DR is the decay rate expressed in dB/m, $A(x_0)$ and $A(x_n)$ are the point and transfer frequency-response functions (FRFs) averaged in each one-third octave band and Δx_n is the spacing between adjacent measurement positions, in m. FRFs in the form of mobility (i.e. ratio between velocity and force) were used throughout the analysis.

2.3 Calculation of decay rate from models

The TWINS model [7-9] predicts the noise from the wheels and track. It can use the measured decay rate as an input parameter to predict the noise from the track or it can calculate the track decay rate. For the calculated decay rates, a model of a Timoshenko beam over a double elastic foundation is used. The rate of attenuation of vibration along the track is obtained from the imaginary part of the wavenumber of the propagating structural wave. The decay rate in dB/m is given by [2]

$$DR = -20\log_{10}\exp(k_i) = -8.686k_i \tag{3}$$

where k_i is the imaginary part of the wavenumber. It can be demonstrated that the TDR is directly related to the noise radiated from the track as follows

$$L_w \approx 10\log_{10}\left(4.343\rho_0 c_0 P\frac{v_{ref}^2}{W_{ref}}\right) + 10\log_{10}\sigma + 10\log_{10}\left(\frac{|v(0)|^2}{2v_{ref}^2}\right) - 10\log_{10} DR \qquad (4)$$

where L_w is sound power level in decibels, $\rho_0 c_0$ is the characteristic acoustic impedance of air, P is a perimeter length of the cross-section of which only the part that is projected onto a plane perpendicular to the motion is considered. W_{ref} is the reference value used for the definition of sound power level, v_{ref} is the corresponding reference value for velocity and σ is the radiation ratio which depends on frequency [2]. From this, it can be demonstrated that the sound power is influenced by the decay rate according to

$$L_w = -10\log_{10} DR + \text{const} \qquad (5)$$

2.4 Model for noise prediction

The TWINS model is used in this study to obtained noise predictions. The TWINS software allows the prediction of vibration levels on the wheels, rails and sleepers as well as the total rolling noise generated in the railway system. The model has been validated with measurements [7-9]. During the passage of a train, the noise can be calculated either in terms of sound power or the average sound pressure at specified positions at the trackside

3 Results

3.1 Dynamic stiffness

The dynamic stiffness of several rail-fastening systems, as well as individual railpads, was measured. A single value of dynamic stiffness was estimated from the magnitude of the measured stiffness using a fitted line for each preload (dashed lines in **Fig. 4 (a)**). The single values were extracted from the fitted line at a frequency of 200 Hz (vertical dashed-dotted line, as shown in **Fig. 4 (a)**). The damping loss factor was predicted from the phase angles of the dynamic stiffness following a similar procedure. Example results for dynamic stiffness as a function of static preload for different railpads are shown in **Fig. 4 (b)**. A summary of the results for some of the pads tested is presented in **Table 1**.

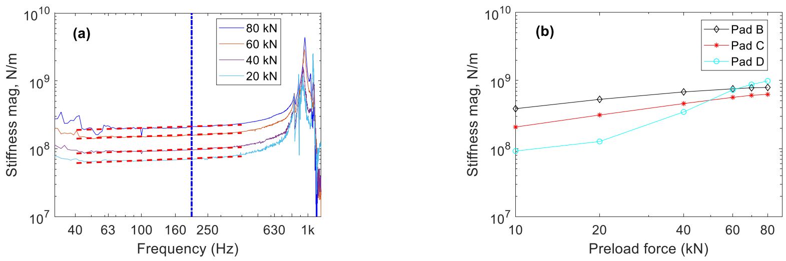

Fig. 4. Dynamic stiffness with frequency for a single railpad **(a)**, dynamic stiffness at 200 Hz as a function of static preload for different railpads **(b)**.

Table 1. Types of rail pads with the dynamic stiffness and damping loss factor measured in the laboratory. Results correspond to a preload of 20 kN and a frequency of 200 Hz.

Type	Pandrol rail pads	Stiffness (MN/m)	Damping Loss factor
A	HDPE EVA plain	1200	0.1
B	21422 EVA studded	528	0.1
C	9970 EVA studded	310	0.1
D	8854 NR Studded	120	0.2
E	Lower pad double studs	50 at 1 kN	0.2

3.2 Track decay rates

Measured vertical and lateral track decay rates for the different fastening systems and rail pad configurations are shown in **Fig. 5.(a)** and **(b)**.

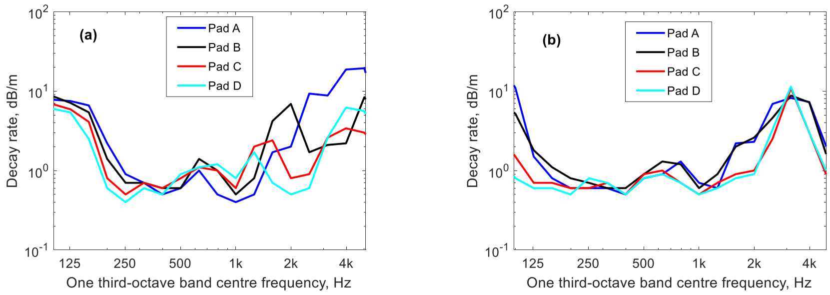

Fig. 5. Measured track decay rate **(a)** vertical, **(b)** lateral.

3.3 Comparison of the measured and predicted decay rates

The track model implemented in TWINS was used to compare the measured decay rates with analytical predictions. The baseplate was modelled as a rigid mass. In order to predict the decay rates, the stiffness and damping loss factor of the rail pad were taken from the measurements at the preload of 20 kN. The stiffnesses in the lateral direction were not measured in the laboratory. In the prediction, the lateral stiffness was obtained by arbitrarily scaling the vertical stiffness by a factor of 0.2. **Fig. 6** shows comparisons between the measured and predicted track decay rates obtained for one example rail pad configuration.

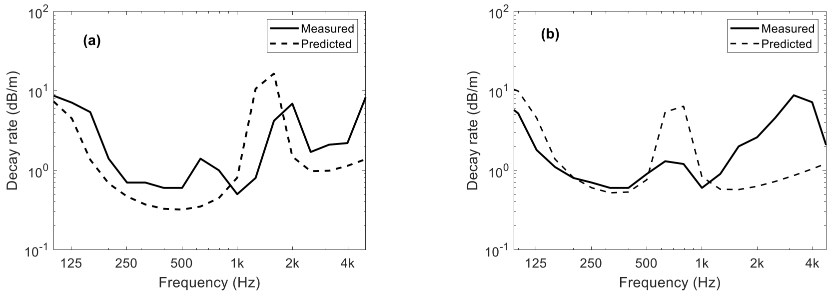

Fig. 6. Comparison between the measured and predicted track decay rate for rail pad B **(a)** vertical, **(b)** lateral.

A reasonably good agreement was found between the measured and predicted TDR using the measured stiffness. However, the measured results also showed some additional peaks at higher frequencies. These peaks could be due to the modal behaviour of the baseplate.

3.4 Noise level prediction using TWINS model

The sound power levels were predicted using both measured and calculated track decay rates in TWINS. A wheel with diameter 0.84 m and a straight web was used, and a nominal roughness corresponding to cast-iron brake blocks was used. The results are presented for 120 km/h in **Fig. 7**. In these figures, the results obtained for both the measured decay rates and the analytical ones are shown. **Fig. 8** shows the total sound power level and the contributions of baseplate, rail, and the wheel for one example case.

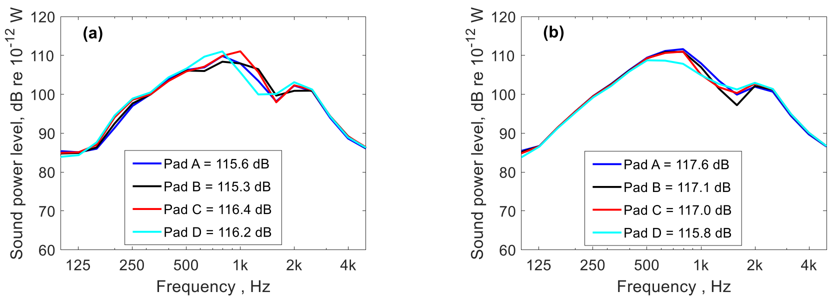

Fig. 7. Total A-weighted sound power level prediction for 120 km/h **(a)** using the measured track decay rate, **(b)** using the calculated track decay rate and the measured dynamic stiffness.

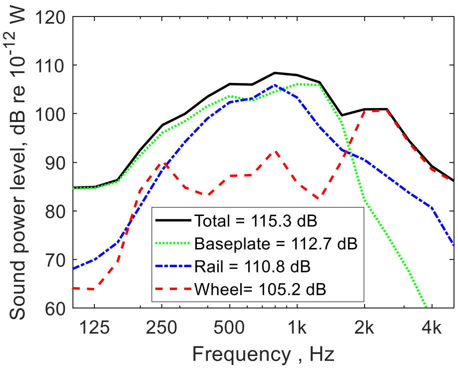

Fig. 8. A-weighted sound power level prediction for rail pad B at 120 km/h using the measured track decay rate, showing the contributions of the baseplate, rail, wheel total.

From these results, there is only a small difference in the noise predicted for the different configurations. The results are summarized in **Table 2**. Using different rail pad configurations the predicted noise varies by up to 1.2 dB between measured and calculated track decay rates. However, the configuration with rail pad B was found to have the lowest noise prediction in both cases.

It was discovered that the sound radiation from the baseplate in the model could be quite significant for high values of rail pad stiffness. In order to obtain more reliable results, further analysis of the baseplate vibration will be conducted to allow a better model of the baseplate to be used for noise predictions.

Table 2. The overall sound power levels for 120 km/h.

Sound power level dB(A) Rail pad configuration	Noise from measured decay rates				Noise from the calculated track decay rates		
	Wheel	Rail	Baseplate	Total	Rail	Baseplate	Total
Pad (A)	105.3	112.3	111.9	115.6	114.4	114.2	117.6
Pad (B)	105.2	110.8	112.7	115.3	113.1	114.4	117.1
Pad (C)	105.3	111.1	114.4	116.2	112.8	114.4	117.0
Pad (D)	105.2	111.3	113.9	116.2	112.0	112.7	115.8

4 Conclusions

The results show generally good agreement between the measured and the predicted TDR. The measured stiffness and TDR have been used to predict the track noise. The noise from the baseplate itself is found to be significant, especially for high values of railpad stiffness. Further analysis of the baseplate vibration will be conducted to allow a better model of the baseplate to be developed.

The results from noise predictions using the measured and calculated track decay rates have shown that the radiated noise has quite small differences between the different configurations. The results have shown that using any of the rail pad configurations, the noise prediction will vary by up to 1.2 dB between measured and calculated track decay rates.

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References

1. British Standard. *EN 15461:2008+A1:2010 -Railway applications. Noise emission. Characterisation of the dynamic properties of track selections for pass by noise measurements*. 2010, British Standard Institution.
2. Thompson, D.J, *Railway Noise and Vibration mechanisms, modelling and means of Control* 2009, Oxford: Elsevier Ltd.
3. Jones, C.J.C, D.J Thompson, and R.J Diehl, *The use of decay rates to analyse the performance of railway track in rolling noise generation.* Journal of Sound and Vibration, 2006. **293**(3): p. 485-495.
4. 10846-3, B.E.I., *Acoustics and vibration-Laboratory measurement of vibro-acoustics transfer properties of resilient elements*
Part 3: Indirect method for determination of the dynamic stiffness of resilient supports for translatory motion. 2002.
5. Thompson, D.J and J.W Verheij, *The dynamic behaviour of rail fasteners at high frequencies.* Applied Acoustics, 1997. **52**(1): p. 1-17.
6. Thompson, D.J, W. Van Vliet, and J. Verheij, *Developments of the indirect method for measuring the high frequency dynamic stiffness of resilient elements.* Journal of Sound and Vibration, 1998. **213**(1): p. 169-188.
7. Thompson, D.J, *Wheel-rail noise generation, part II: wheel vibration.* Journal of sound and vibration, 1993. **161**(3): p. 401-419.
8. Thompson, D.J, P. Fodiman, and H. Mahé, *Experimental validation of the TWINS prediction program for rolling noise, part 2: results.* Journal of sound and vibration, 1996. **193**(1): p. 137-147.
9. Thompson, D.J, B. Hemsworth, and N. Vincent, *Experimental validation of the TWINS prediction program for rolling noise, part 1: description of the model and method.* Journal of sound and vibration, 1996. **193**(1): p. 123-135.