A comparison between time and frequency domain models for description of railway noise generation

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Introduction

This paper describes and demonstrates why a model for prediction of railway rolling noise at least sometimes has to be performed in the time domain and not in the frequency domain, in order to obtain valid prediction results.

Mostly, railway noise is predicted using frequency domain models (e.g. TWINS [3] and RIM), since frequency domain models have many advantages: they are relatively easy to perform and their calculation times are short. However, the disadvantage is that they are incapable of describing all relevant effects taking part in rolling noise generation, which leads to prediction errors. Thus, it is not always possible to design and noise optimize a railway track properly using frequency domain models.

To illuminate this, numerical simulations with two models, one in the time domain and the other in the frequency domain, are performed and the results shown below. All track and wheel parameters used are identical in both models unless otherwise stated.

Figure 1: A railway wheel rolling on a railway rail with surface roughnesses. The rail/wheel contact force excites rail and wheel into vibrations which radiate away in the air as noise.

Models

Time Domain Model

Reference [1] describes the model in detail. Below, a short summary follows.

Rail and wheel responses (vertical deflections for the rail y_r and the wheel y_w) are calculated with the help of rail and wheel impulse response functions (g_r and g_w), which have been calculated using inverse Fourier transforms of the corresponding rail and wheel transfer functions (Green's functions). The wheel rotates and the rail/wheel contact point moves forward over the rail with train speed v. The contact force f is a function of the compression δ of the rail/wheel contact area, which in its turn is a function of rail and wheel deflections and the surface roughness amplitude y_s. Thus, the excitation force depends on the roughness amplitude **and** on rail and wheel deflections through a feedback coupling mechanism.

$$y_{\mathrm{r,w}} = \iint g_{\mathrm{r,w}} f \,\mathrm{d}x\mathrm{d}t \tag{1}$$

$$\delta = y_\mathrm{r} + y_\mathrm{s} - y_\mathrm{w} \tag{2}$$

Frequency Domain Model

The wheel stands still on a spot somewhere on the rail, it is not rotating. Only rail and wheel input receptances (α_r and α_w) need to be considered, not transfer receptances. A „ribbon“ with roughness spectrum $\Phi_{r_0r_0}(k_0)$ is drawn with speed v through the contact region between rail and wheel, where k_0 is the surface roughness wavenumber k_0. The excitation force spectrum $S_{F_0F_0}$ depends on the train speed, the roughness amplitude spectrum, rail and wheel input receptances, and the linearized Hertzian contact spring k_H [2, 3]. There is no feedback coupling from rail/wheel responses to the excitation force. Rail and wheel vibration spectra, $S_{Y_\mathrm{r}Y_\mathrm{r}}$ and $S_{Y_\mathrm{w}Y_\mathrm{w}}$, are obtained by simply multiplying the force spectrum with the corresponding receptance function.

$$S_{F_0F_0} = \frac{1}{v} \cdot \Phi_{r_0r_0}(k_0) \cdot \left| \frac{k_\mathrm{H}}{1 + k_\mathrm{H}(\alpha_\mathrm{r} + \alpha_\mathrm{w})} \right|^2 \tag{3}$$

$$S_{Y_\mathrm{r}Y_\mathrm{r}} = S_{F_0F_0} \cdot |\alpha_\mathrm{r}|^2 \tag{4}$$

$$S_{Y_\mathrm{w}Y_\mathrm{w}} = S_{F_0F_0} \cdot |\alpha_\mathrm{w}|^2 \tag{5}$$

We call this model a frequency domain model (as opposed to a time domain model) throughout this paper, but we could also have called it an impedance model, since mainly point impedances of rail and wheel determine the force and response levels.

Results

Exactly the same rail and wheel parameters (see [1]) are used in the time domain model simulations as well as in the frequency domain model simulations. The time domain model uses a periodically supported rail. However, in the frequency domain model, it is assumed that the sleeper masses are evenly distributed under a continuously supported rail. Smearing out sleeper masses should roughly have the same effect as otherwise averaging over all different wheel positions between two sleeper positions. Another discrepancy between the frequency domain model and time domain model is that the rigid mass mode of the wheel is not included in the wheel receptance α_{w}, which introduces an error in the frequency domain model at low frequencies.

The numerical simulations are performed for two different pad stiffnesses: 100 MN/m for soft pads, and 1000 MN/m for stiff pads; and three different train speeds: 200 km/h, 300 km/h, and 400 km/h.

Below, contact force spectra as well as rail and wheel response levels are shown.

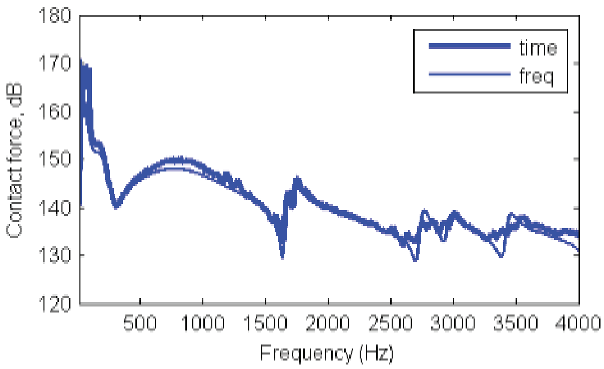

Figure 2: Contact force spectrum for time domain model and frequency domain model. Soft pads, 200 km/h.

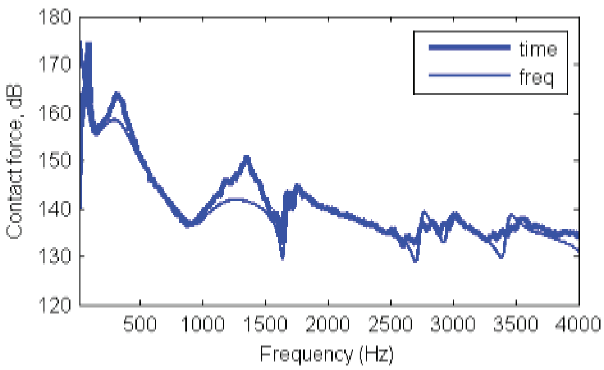

Figure 3: Contact force spectrum for time domain model and frequency domain model. Stiff pads, 200 km/h.

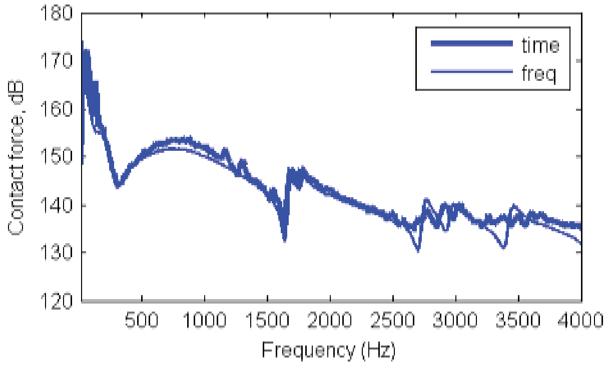

Figure 4: Contact force spectrum for time domain model and frequency domain model. Soft pads, 300 km/h.

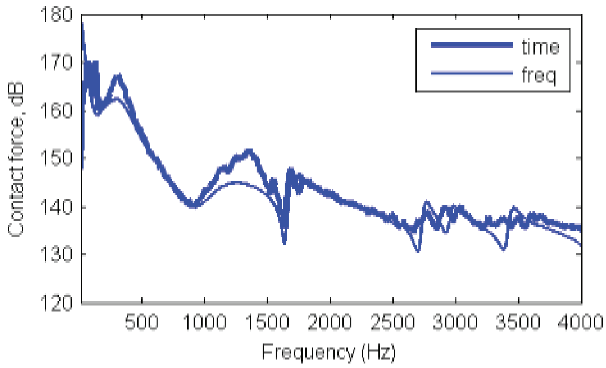

Figure 5: Contact force spectrum for time domain model and frequency domain model. Stiff pads, 300 km/h.

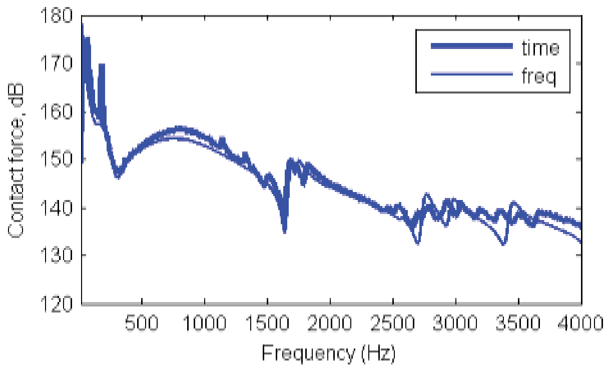

Figure 6: Contact force spectrum for time domain model and frequency domain model. Soft pads, 400 km/h.

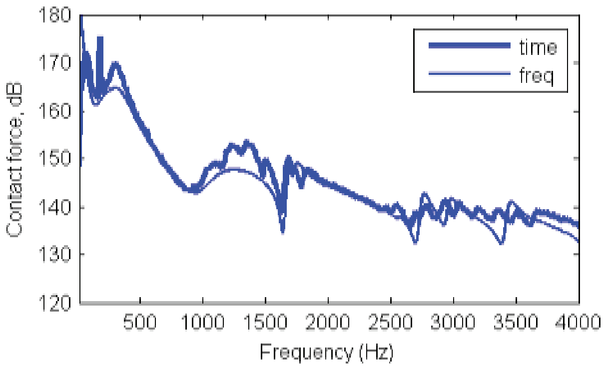

Figure 7: Contact force spectrum for time domain model and frequency domain model. Stiff pads, 400 km/h.

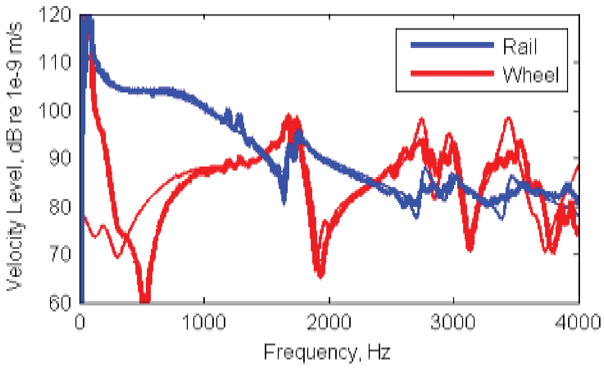

Figure 8: Rail and wheel response vibration levels for time domain model (thick line) and frequency domain model (thin line). Soft pads, 200 km/h.

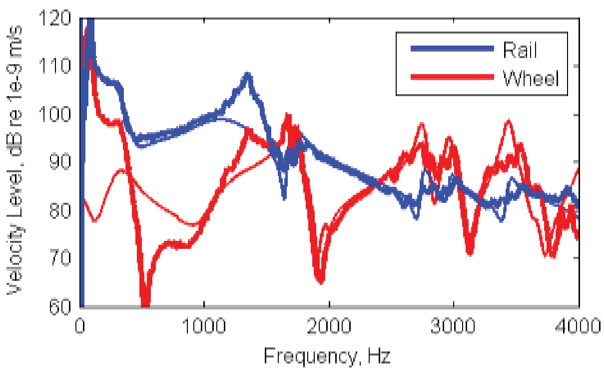

Figure 9: Rail and wheel response vibration levels for time domain model (thick line) and frequency domain model (thin line). Stiff pads, 200 km/h.

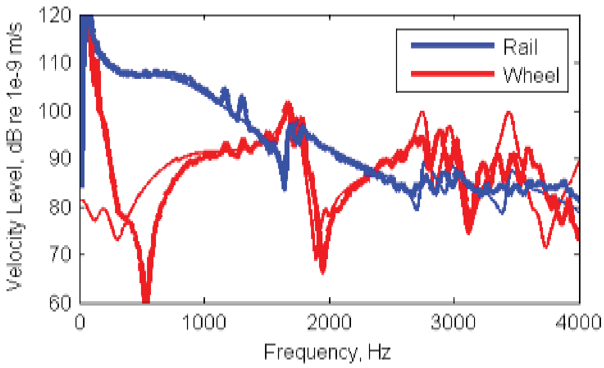

Figure 10: Rail and wheel response vibration levels for time domain model (thick line) and frequency domain model (thin line). Soft pads, 300 km/h.

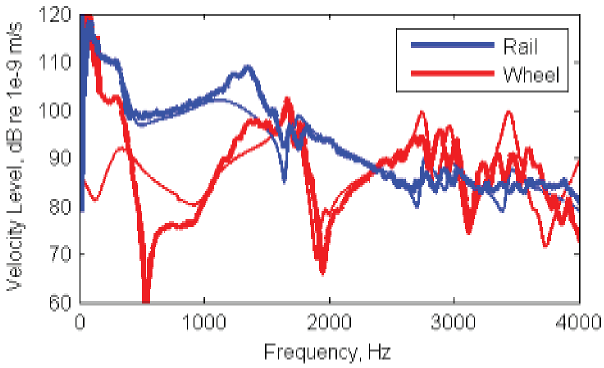

Figure 11: Rail and wheel response vibration levels for time domain model (thick line) and frequency domain model (thin line). Stiff pads, 300 km/h.

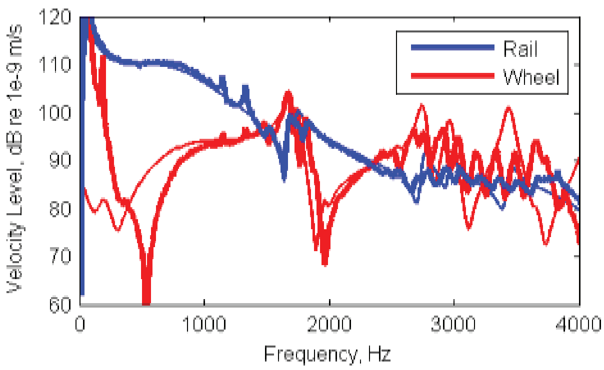

Figure 12: Rail and wheel response vibration levels for time domain model (thick line) and frequency domain model (thin line). Soft pads, 400 km/h.

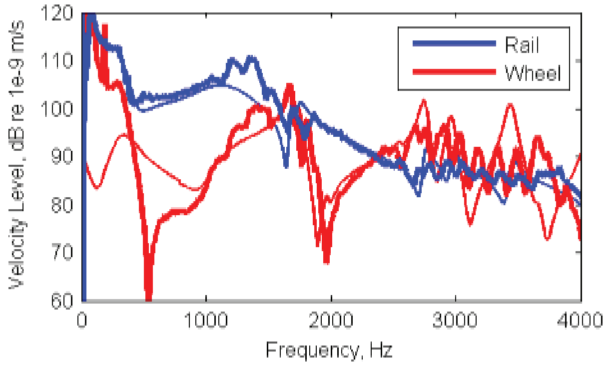

Figure 13: Rail and wheel response vibration levels for time domain model (thick line) and frequency domain model (thin line). Stiff pads, 400 km/h.

Discussion

The frequency domain model under predicts rail vibration levels below 1500 Hz. For rail on soft pads the under prediction is about 5 dB (Figure 14), and occurs at the two frequencies to where the pinned-pinned frequency has shifted, due to the forward speed of the wheel [1]. For rail on stiff pads the error is around 10 dB (Figure 15) in a broad region around the pinned-pinned frequency, which occurs around 1250 Hz for the sleeper distance used in these simulations.

The reason for under prediction of rail vibration levels using the frequency domain model must be due to the fact that the frequency domain model cannot include the effect of the feedback coupling mechanism between wheel/rail response and contact force, which on the other hand the time domain model does (see eq. (2)). Possibly, for very rough rails, this effect might not be as pronounced as in these examples using a rather low rail roughness level (10 dB below TSI).

At the pinned-pinned frequency, the rail vibrates with great amplitudes. Through the feedback coupling mechanism, the pinned-pinned mode thus strongly takes part in the excitation mechanism around the pinned-pinned frequency, explaining the big difference in vibration levels obtained with the two models.

Thompson concluded [3], that the pinned-pinned mode usually only has a small effect on noise radiation, although it was emphasized by Nordborg in [2] that stiff pads, making the pinned-pinned mode more pronounced, increases modulation around the pinned-pinned frequency, which increases noise radiation. Obviously, more work is needed to determine the importance of the pinned-pinned mode in rail noise radiation.

Above 2000 Hz the agreement is better; however, the frequency domain model fails to capture splitting of the wheel resonance peaks as train speed increases.

To predict railway rolling noise, and to design and noise optimize a railway track properly, it seems to be necessary to use a time domain model, at least for smooth rails on stiff pads.

Future work will investigate under what circumstances, i.e. rail roughness level, train speed, pad stiffness, etc., a time domain model has to be used. Noise radiation has to be included in the model.

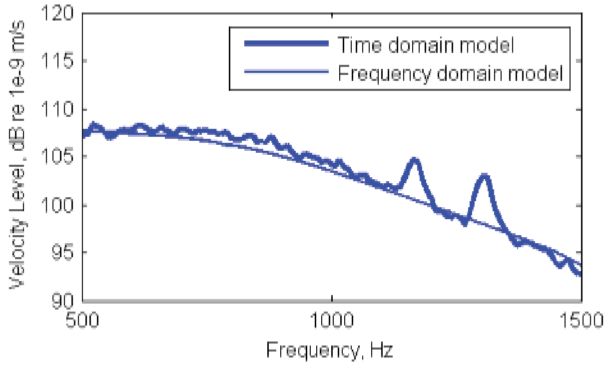

Figure 14: Rail response vibration levels for time domain model (thick line) and frequency domain model (thin line). Soft pads, 300 km/h. See also Figure 10.

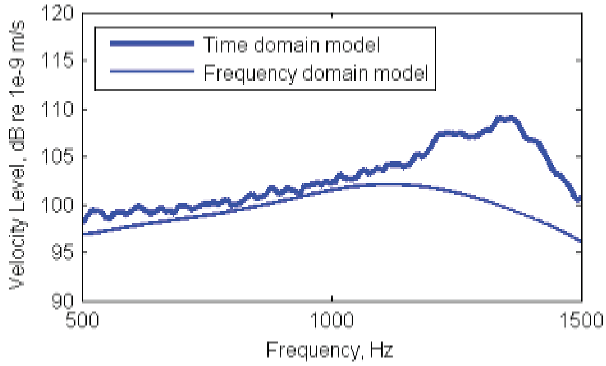

Figure 15: Rail response vibration levels for time domain model (thick line) and frequency domain model (thin line). Stiff pads, 300 km/h. See also Figure 11.

References

[1] A. Nordborg, T Kohrs: Ein Zeitbereichsmodell zur Beschreibung der Rollgeräuschentstehung. DAGA 2015 Nürnberg.

[2] A. Nordborg: Vertical Rail Vibrations: Parametric Excitation. ACUSTICA · acta acustics 84 (1998), 289-300.

[3] David Thompson: Railway Noise and Vibration. Elsevier, 2009.