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A full-scale test rig for railway rolling noise: simulation and measurement of dynamic wheelset–track interaction

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Abstract

A new outdoor rolling-noise test rig on a 25 m stretch of full-scale track will enable the study of vibrations of wheel and rail and of the pertinent noise emission under controlled conditions. The arrangement can be seen as a physical realization of the Track–Wheel Interaction Noise Software (TWINS) computer software. The track and wheel, which are not in mechanical contact, are excited in vertical and lateral directions using electrodynamic actuators. The track can be statically pre-loaded by up to 30 tonnes. The use of the rig is presently under development. The aim is that the radiated noise from separate railway components could be found as the wheel and the track can be excited both together and separately. Amplitude and phase of the applied forces are predetermined by use of an algorithm taking into account the real wheel–rail interaction properties. In that way different wheel–rail contact conditions can be simulated. Eight partners have co-operated in the development and operation of the CHARMEC/Lucchini Railway Noise Test Rig in Surahammar, Sweden.

In ongoing experiments, the dynamics of both the wheel and rail have been examined in the frequency domain. For the track, comparisons have been made between data obtained from the rig and those from field measurements on a standard Swedish line. Both dynamic response and spatial decay rates have been studied. The performance of the rig has also been compared to results from TWINS and to results from the literature. Good agreement was obtained in the frequency range from 100 to 5000 Hz. Some results from preliminary measurements of noise emission will be given.

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1. Introduction

With higher train speeds and larger traffic volumes, railway noise is increasingly important in our society. A dominating source is the excitation in the contact area between wheel and rail. Rolling noise is created and amplified by surface roughness (waviness, corrugation) on the wheel and rail surfaces. Moreover, squeal and screech noise arises when the wheel and rail surfaces slip in curves and during braking.

In the Railway Noise Test Rig (RNTR) located at Lucchini Sweden in Surahammar, it will be possible, in full scale, to simulate the vibrations and sound emission from the track, from the track and wheel, and also from a bogie. Track dynamics studies are also within the scope of the rig. The wheel and rail are excited in a way that reproduces real operations. Noise from wheel and noise from rail can be measured both separately and in combination. Different designs of track components (rails, clips, pads, sleepers) and wheels (solid, damped, spoked, perforated) and different bogie arrangements (shields, fairings) will be tested. Furthermore, it will be possible to study different noise barriers along the track. The ongoing CHARMEC research project ‘‘Vibrations and external noise from train and track’’ uses and develops the RNTR. A major aim is to demonstrate that in-field characteristics of different components can be predicted through tests on the RNTR. A review of the literature on railway noise is given in Ref. [1] where some full-scale and reduced-scale noise rigs are also discussed briefly. Comprehensive state-of-the-art reports were also written in the European projects Silent Track [2] and Silent Freight [3]. Recent developments in the field of railway noise research are summarized in Ref. [4].

2. The RNTR

The concept behind the development of the RNTR began in 1995 within the Swedish project ‘External noise of railway vehicles radiated from wheel-rail and tread braked systems’ which was partly financed by the Swedish National Board for Industrial and Technical Development (NUTEK, now VINNOVA). The participants in this project were Chalmers Applied Mechanics, ABB Corporate Research, SP (the Swedish National Testing and Research Institute) and Banverket (the Swedish National Rail Administration). As part of other activities, a working group began to discuss the features of a new RNTR. Construction of the infrastructure of the RNTR started in the Summer of 1997 and was finalized during the following Spring. The use of the RNTR is now controlled by a consortium consisting of ABB Corporate Research, Abetong Teknik, Banverket, Bombardier Transportation Sweden, Chalmers Applied Mechanics, Lucchini Sweden, SP and TrainTech Engineering (formerly SJ Rolling Stock Division). The rig is situated on the premises of the railway wheelset manufacturer Lucchini Sweden in Surahammar, Sweden.

3. General features

The RNTR (see Fig. 1) was developed for the study of rolling contact induced vibrations and noise in the frequency range from 500 to 5000 Hz. The vibrational behaviour of the wheel (in a wheelset) and of the track can be established from measured excitation forces and accelerations.

Fig. 1. The CHARMEC/Lucchini railway noise test rig (RNTR) in Surahammar.

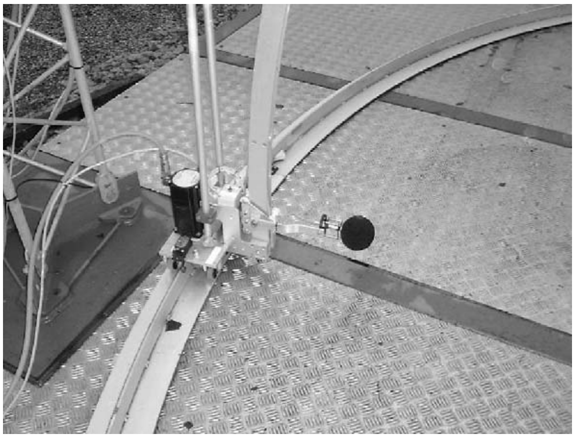


Fig. 2. The microphone positioning system for wheel noise measurements.

The attenuation of vibrations along the track can be measured. The sound emission from wheel and rail are measured using separate microphone positioning systems (see Fig. 2).

The dynamic contact forces between the wheel and the rail are applied using force actuators acting on wheel and rail. Using simultaneous excitation in the vertical and lateral directions, the correct displacement of the contact point between wheel and rail can be achieved. The track is 25 m long, with sand heaps covering an additional 10 m of rail at both ends to model boundary conditions similar to those of a segment of an infinite track. The track is regular Swedish track. UIC 60 rails are placed on monobloc concrete sleepers with nominally 0.65 m spacing using studded 10 mm thick rail pads and Pandrol fastenings (e-clips). The sleepers are placed on 0.30 m of ballast which rests on 3.5–5 m of clay and then moraine on bedrock about 8 m below ground level.

An important feature of the track in the RNTR is the ability to apply a static pre-loading to simulate train weight. A heavy concrete beam is placed underneath and along the track formation.

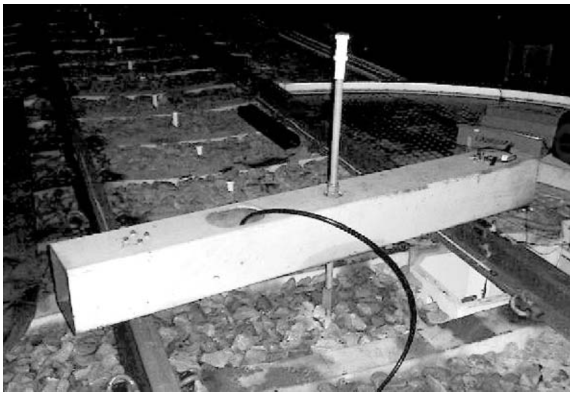


Fig. 3. The pre-loading beam with the hydraulic jack inside the RHS profile.

The concrete beam is fixed to the bedrock using pre-stressed anchors. In the middle of each sleeper bay, vertical steel rods protrude from the beam. The track can be pre-loaded via a steel beam across the track using a hydraulic jack, which is fixed to one of the steel rods (see Fig. 3). The system is designed for a pre-loading, which corresponds to axle loads up to 30 tonnes. The dynamics of the pre-loaded system is different from the dynamics of the unloaded system primarily because of large deformations of the pads which gives them a higher dynamic stiffness [5]. The dynamic force entering the pre-loading system is measured by a piezo-electric force transducer placed between the railhead and the air spring of the pre-loading beam (see also Fig. 10). The soft air springs are intended to reduce the influence of dynamic interaction between the track and the pre-loading beam. No results that include the pre-loading system will be presented here since it is currently under development.

4. Measurement system

The measurement system is composed of four main parts. A *computer* controls the *excitation system*, the *microphone positioning system* and the *data sampling system* during testing. The excitation system consists of force actuators, power amplifiers and function generators. A maximum force amplitude of 440 N up to 2000 Hz using sinusoidal excitation can be generated. Currently, two force actuators are available at the RNTR making possible, for example, a simultaneous vertical and lateral excitation of wheel *or* of rail. The vertical and lateral force levels, and the relative phase between them, can be set by the operator for single frequency sine excitation. When the RNTR is fully developed, four force actuators will be available making it possible to excite both the wheel and the rail simultaneously in both the vertical and lateral directions. Note that there is a gap between wheel and rail; that is, there is, no mechanical contact.

Accelerometers and force transducers are used for measuring vibrations and excitation forces. Free-field microphones are used for measuring the noise from the excited structures. Furthermore, the ambient background levels away from the RNTR can be measured during testing. The force transducers and accelerometers are connected to an eight-channel charge amplifier and the microphones to a dual-channel microphone amplifier.

Microphone positioning for wheel and rail is performed using two separate robotic systems. For the measurement of noise from the wheel, the microphone can be moved over an imaginary surface of a quarter sphere, whilst for the rail, the microphone can be moved over an imaginary surface of a quarter cylinder. For both systems, the noise measurements can be carried out with the microphone either at a fixed position or whilst moving continuously. When using fixed microphone positions, the measurements can be made at a number of positions in a grid pattern to cover the surface of the quarter sphere or cylinder. Microphone movements can be categorized as two types: linear speed sweeps for directivity measurements and constant area sweeps for measurement of sound power. During directivity sweeps, noise data and microphone positions can be sampled over part of the sweep, whereas during power sweeps the data are averaged over the whole sweep.

5. Wheel measurements and analyses

Vibration measurements on a standard freight car wheelset SJ 57H (Fig. 4), with free–free boundary conditions were carried out, using modal hammer excitation. The exciting force was applied in the lateral and vertical directions at the contact point, and direct and cross receptances were measured. The direct receptances of the contact point are shown in Fig. 5. The wheel was not rotating during the measurements on the test rig (the subsequent inaccuracy of the results is considered to be very small).

To run Track–Wheel Interaction Noise Software (TWINS), an FE-model is required. Modal parameters from a dynamic FE-analysis provide necessary information for calculation of receptances, responses and noise radiation. There are several examples from validation of TWINS models [6,7], where the modal parameters of the FE-model have been tuned to measured receptances. For monobloc steel wheels particularly, the calculated and measured resonance frequencies are expected to differ very little. The main cause for these discrepancies is the variation

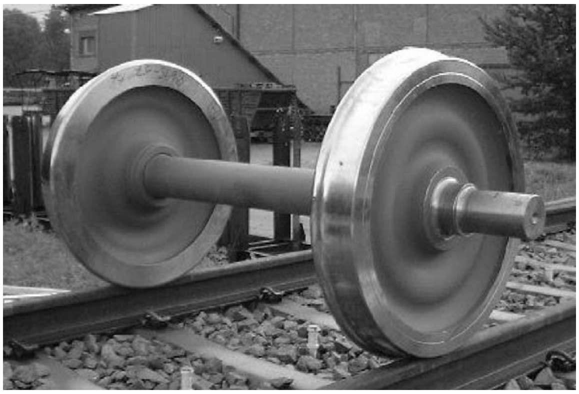
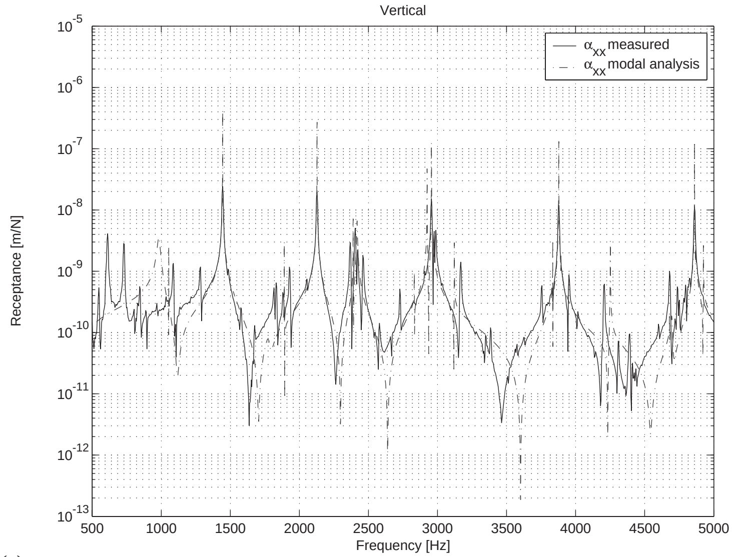
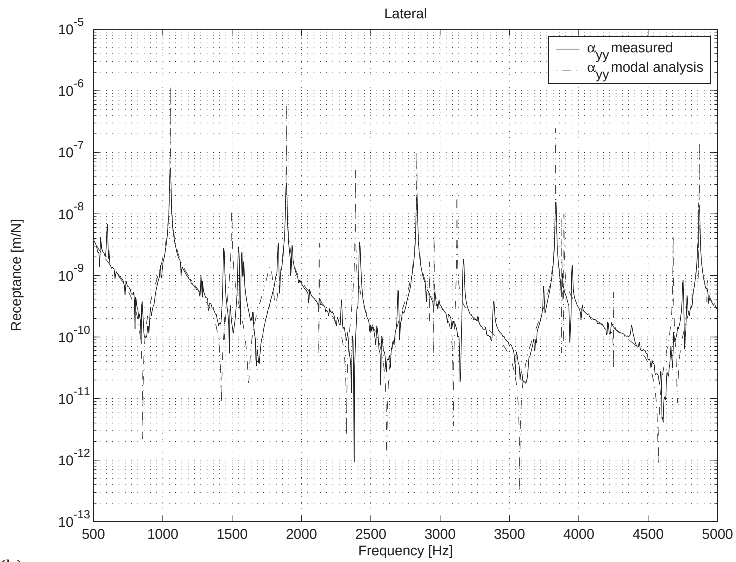


Fig. 4. The standard wheelset SJ 57H used in the measurements.



(a)



(b)

Fig. 5. (a) Vertical and (b) lateral direct receptances from measurements and FE-based modal analysis on the wheel in Fig. 4.

in web thickness due to manufacturing tolerances. The discrepancy between the measured and the modelled amplitudes of the resonance peaks observed in Fig. 5 depends on two factors. Firstly, a relatively low frequency resolution was used for the measured values. Secondly, for tuning the FE-model the damping factors were extracted, from a high-resolution measurement, by using only the dominant resonance peaks. The overall agreement in the relevant frequency range is good.

6. Track measurements and analyses

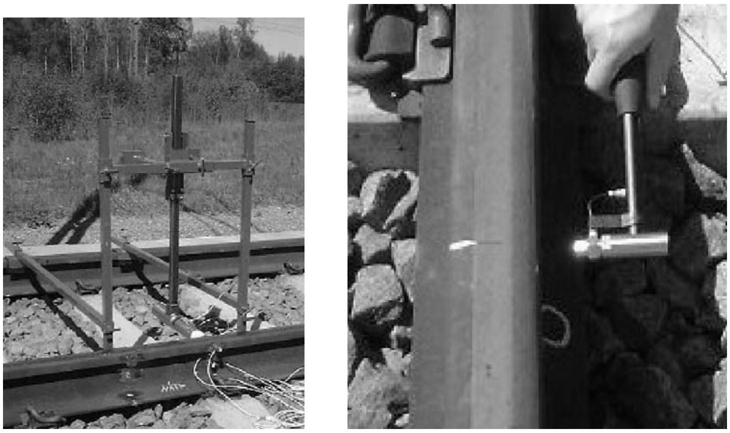
As the track section at Surahammar is only 25 m long with heaps of sand to supply damping at the ends, it is vital to investigate whether its vibrational behaviour is representative for a segment of a Swedish track in revenue-earning service. A standard Swedish track design includes UIC 60 rails, monobloc concrete sleepers, and soft 10 mm thick studded Pandrol pads (Fig. 6).

Measurements on a reference track section near Borlänge and at the RNTR in Surahammar were carried out in a manner similar to that used in Ref. [8]. In short, the vertical and lateral rail response was measured at the *midpoint* of a sleeper bay, and the excitation is applied both at the midpoint and at specified distances away from that point (in a grid). The same procedure was repeated with the response measured *above* a sleeper. The rail response was measured using nine accelerometers on the rail cross-section. The grid distance close to the excitation point was 0.13 m as the sleeper spacing in Sweden is nominally 0.65 m. In the RNTR in Surahammar, the average sleeper distance is 0.651 m, with a standard deviation of 0.020 m. On the reference section at Borlänge the average sleeper distance is 0.650 m and the standard deviation 0.021 m. The four sleeper bays closest to the excitation point were divided into a fine grid (20 points). From the fourth sleeper bay to the 21st sleeper bay, measurements were performed only *between* sleepers. The procedure was exactly the same at Borlänge as in the RNTR in Surahammar. The same measurement equipment was utilized at both sites. In the lower frequency range (50–1200 Hz) a sledgehammer with a thick polyethylene tip was used. In the higher range (1000–5000 Hz), a modal hammer was used (see Fig. 7). The track was excited first laterally and then vertically at each measurement point, and each point was excited 20 times or more in each direction.

The experiments were also compared to calculations using TWINS, a software based on the work in Ref. [9]. The track model called TINF (with a Timoshenko beam on discrete periodic supports) was used and the tuned parameters are given in Table 1.

(a) (b)

Fig. 6. (a) A Swedish standard track section located at Borlänge. (b) Detail with track components.



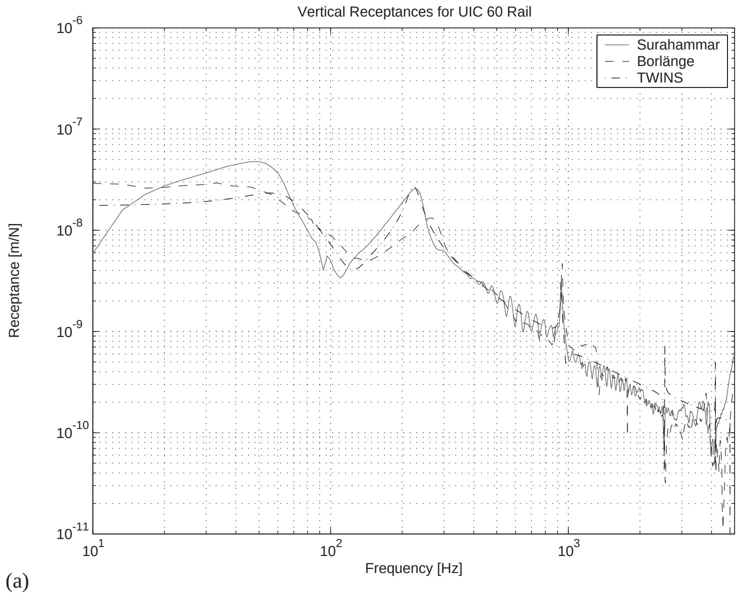
(a) (b)

Fig. 7. The two methods with (a) sledgehammer and (b) modal hammer, used for exciting the standard reference track at Borlänge and at the RNTR in Surahammar in Sweden.

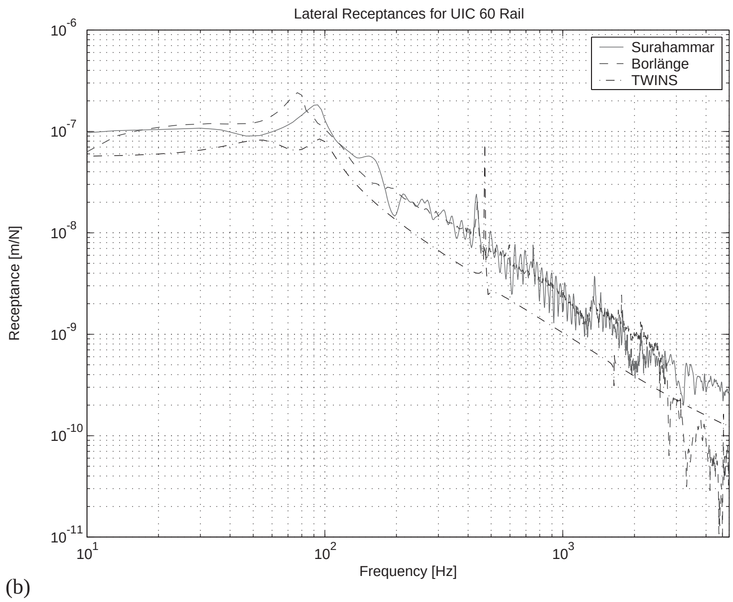
Table 1
Input data for the TWINS calculations

TWINS model	TINF and RODEL
Vertical pad stiffness	60 MN/m
Lateral pad stiffness	10 MN/m
Vertical pad loss factor	0.1
Lateral pad loss factor	0.1
Vertical ballast stiffness	25 MN/m
Lateral ballast stiffness	20 MN/m
Ballast loss factor	0.7
Type of rail	UIC 60
Rail loss factor	0.007
Sleeper spacing	0.65 m
Cross receptance factor	−13 dB
Static load	100 kN
Train speed	100 km/h

Measured receptances along with TWINS simulations are shown in Fig. 8 for excitation between sleepers. A good overall agreement between the test rig results from Surahammar and the results for the reference track at Borlänge was obtained. However, some ripples on the receptance curves from Surahammar can be observed in the frequency interval 500–2000 Hz. They are believed to be caused by the finite length of the track with its reflections from the sand heaps. The Borlänge measurements show dips at frequencies higher than 4 kHz. At these dips the coherence is low and the agreement between the results from Surahammar and Borlänge is not as good. For the cross receptances of the rail, a similar agreement was found between Surahammar and Borlänge.



(a)



(b)

Fig. 8. (a) Vertical direct receptance at a position between two sleepers and (b) lateral direct receptance at a position between two sleepers for Borlänge, Surahammar and TWINS.

7. Track spatial decay rate

An important characteristic of a track is its spatial decay rate. It is important to take this property of the RNTR into consideration, as the finite track has two heaps of sand at its ends to imitate the non-reflecting ends of a stretch of an infinite track. The results in Fig. 9 show that the

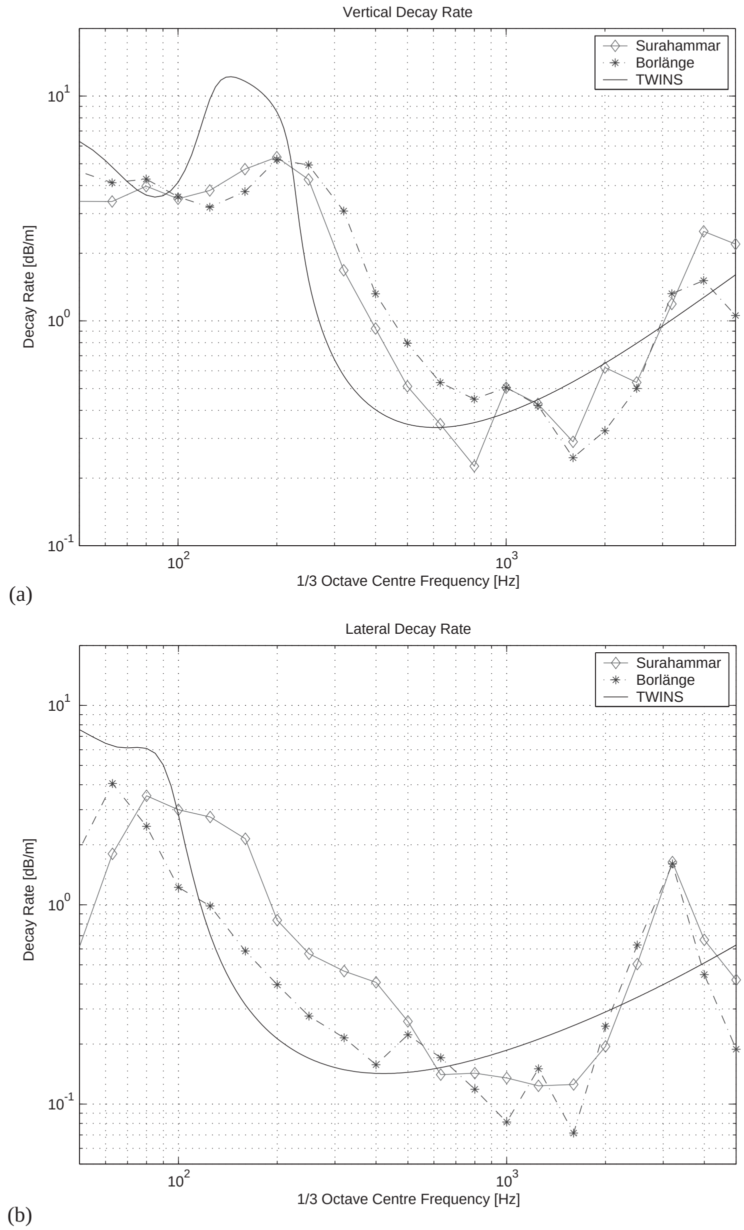


Fig. 9. Results from measurements at Borlänge and Surahammar (in 1/3 octave bands) and from calculations by use of TWINS. (a) Vertical decay rate and (b) lateral decay rate.

RNTR is a good representation of a standard track in Sweden. The decay rate, Δ, can be estimated [10] by

$$\Delta = \frac{4.343}{\int_0^\infty (|A(x)|^2/|A(0)|^2)\,\mathrm{d}x} \approx \frac{4.343}{\sum_{i=1}^{N}(|A(x)|^2/|A(0)|^2)\Delta x_i}\,[\mathrm{dB/m}], \tag{1}$$

where $A(x)$ is the transfer accelerance at position i, $A(0)$ is the point accelerance at the point of excitation, Δx_i is the distance between adjacent points and N is the number of measurement points. Decay rate is measured both in the lateral and the vertical direction (see Fig. 9).

Vertical and lateral decay rates are estimated using linear curve fitting for the measured response at different distances [10]. The rapid decay closest to the excitation point (near-field decay) is not included in the estimations. Also a comparison has been made with results from TWINS calculations using a track model called RODEL (see Table 1). In this case, the track is modelled as having continuous support, which means that the characteristic pinned–pinned resonance frequencies (the fundamental ones at 940 Hz vertically and 470 Hz laterally) are absent (Fig. 9). For a detailed analysis on noise from a wheel–rail system it is recommended to use measured decay rates.

8. Parallel excitation of wheel and rail

It is possible to realize a simultaneous parallel excitation using the RNTR. The relative displacement excitation model presented in Ref. [9] has been implemented in order to determine the forces to be applied on the wheel and the rail. This is a technique that couples the vibrational behaviour of the wheelset and that of the track via an assumed roughness profile and a finite contact patch length. The applied roughness introduces a relative displacement in the vertical direction between the wheel and rail. The wheel and rail are coupled in the lateral and the vertical direction at the contact point. A contact receptance has been introduced where Hertzian contact is assumed and no creepage or spin is present. It has been found that the inclusion of creep gives little difference for a normal range of parameters (wheel diameter, static load, railhead curvature). The coupling between the wheel and the rail [9] is given by

$$\mathbf{P} = \boldsymbol{\alpha}^{-1}\mathbf{z}, \tag{2}$$

where $\mathbf{P} = \{P_{vert}, P_{lat}\}^{\mathrm{T}}$ [N] contains the (frequency dependent) vertical and lateral forces, α [m/N] is the 2×2 wheel–rail system receptance and $\mathbf{z} = \{z_{vert}, z_{lat}\}^{\mathrm{T}}$ [m] contains the relative displacement of the wheel–rail contact introduced via an assumed roughness profile (in this case, only in the vertical direction). The wheel–rail system receptance is the sum of the receptances of the individual parts

$$\boldsymbol{\alpha} = \boldsymbol{\alpha}^{r} + \boldsymbol{\alpha}^{c} + \boldsymbol{\alpha}^{w}, \tag{3}$$

where superscript r, c and w are used to indicate rail, contact zone and wheel, respectively. The receptances for the rail and wheel can readily be measured as described in previous sections. Assuming Hertzian contact, the contact zone receptances for vertical and lateral directions take the form

$$\boldsymbol{\alpha}^{c}_{vert} = \frac{\xi}{2}\left(\frac{2}{3(E^{*})^{2}R_{e}P_{0}}\right)^{1/3}, \tag{4}$$

$$\boldsymbol{\alpha}^{c}_{lat} = x\boldsymbol{\alpha}^{c}_{vert}, \tag{5}$$

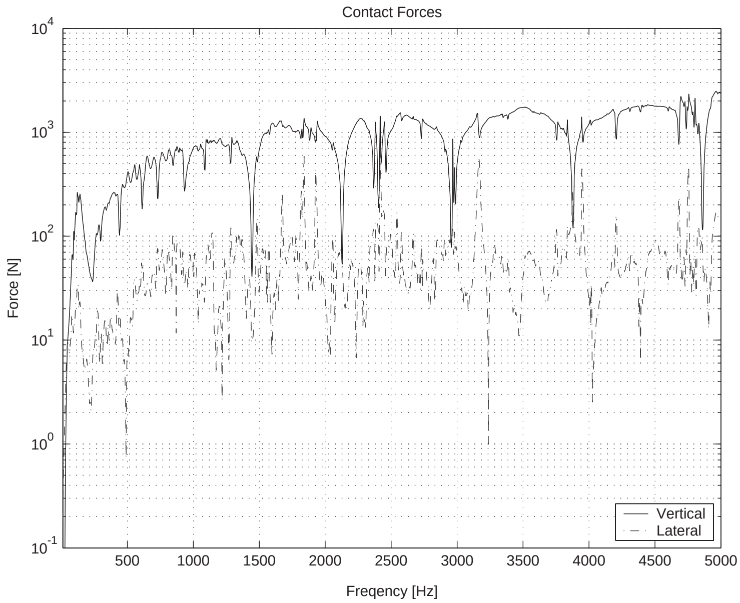


Fig. 10. Calculated vertical and lateral forces on wheel and rail for simulation of a roughness amplitude of 1 μm.

where ξ is a dimensionless quantity dependent on the radii of curvature of the two surfaces, and hence the shape of the contact patch, typically close to 2 for this application. The plane strain elastic modulus is $E^* = E/(1 - v^2)$, the factor x has empirically been found to be in the range of 1.0–1.2 and the static vertical load is P_0. The effective radius of curvature is expressed as

$$R_e = \left(\frac{1}{2}\left(\frac{1}{R_{11}} + \frac{1}{R_{12}} + \frac{1}{R_{21}} + \frac{1}{R_{22}}\right)\right)^{-1}, \tag{6}$$

where R_{11} is the radius of the wheel, R_{12} is the lateral curvature radius of the wheel, R_{21} is the radius of curvature of the rail in its longitudinal direction and R_{22} is the curvature of the railhead. In the special case here, these radii are taken as 0.46 m, ∞, ∞ and 0.30 m respectively, corresponding to a wheel with diameter of 920 mm and straight running profile, and a straight rail section. A typical value of the vertical contact tangential stiffness for a pre-load of 100 kN in this application is 1.45 GN/m.

The contact forces have been calculated using measured receptances for the wheel and the rail in the RNTR (see Fig. 10), with a roughness amplitude of 1 μm applied over the entire investigated frequency interval. Using the relationship for the calculated forces on the wheel and the rail in the RNTR will replicate actual vibrational behaviour and noise emission.

9. Sound emission

A main aim of the RNTR is to make possible a detailed measurement of the noise emitted from wheel and track. In the present set-up, a traversing microphone scans a spherical surface around

Fig. 11. Visualization of the emitted noise levels from a wheel prototype (not SJ 57H) measured in the RNTR at the resonance frequency 875 Hz. Dark colour on the surface of a quarter sphere indicates high sound pressure levels whereas white indicates low levels.

the wheel and a cylindrical surface around the rail. A control system has been designed which enables different continuous or step-wise methods of positioning the microphone. Fig. 11 shows, as an example, the sound pressure level measured on the half-quarter sphere surface when the wheel was excited vertically at a resonance frequency. During the measurements, the opposite wheel was sound insulated to prevent reflections and/or radiation from that wheel.

A future report will focus on noise measurements. The RNTR was, however, used in a preliminary study on wheel noise within the Silent Freight project [11]. In that study, wheel sound power was predicted by an approach combining measured acoustic transfer function spectra with contact point excitation forces calculated in TWINS based on measured receptances. The advantage with complex-valued spectra is that separate excitations with lateral and vertical forces can be applied, followed by a superposition of the results. The results from the preliminary study showed that this approach is potentially effective and some improvements to be considered in future experiments were identified.

10. Conclusions and discussion

Railway noise is a key issue in many countries, and the detailed mapping and knowledge of the noise sources is of increasing significance in order to meet noise legislation criteria. Low noise levels are also a competitive factor for the manufacturers. The CHARMEC/Lucchini Railway Noise Test Rig (RNTR) in Surahammar will increase the understanding of the noise generation for certain key components as well as act as a validating complement to modelling.

The wheel and track vibration characteristics of the CHARMEC/Lucchini RNTR in Surahammar in Sweden have been described. The coupling of the dynamic characteristics of track and wheelset has been investigated. The aim is to be able to predict experimentally the radiated noise from different railway components in a detailed and controlled manner. Preliminary sound emission measurements have already been carried out.

Future improvements to the test rig include an excitation system that applies the same counteracting forces to the wheel and the track when they are not in mechanical contact. Also wheel rotation could be accommodated, which, however, would require excitation vertically and laterally without mechanical contact (for example, by magnetic methods). Furthermore, the current pre-loading beam with its air springs will be modified to allow the rail behaviour be studied at all frequencies of interest. The lower frequency limit of the measurement system (500 Hz) will be lowered to about 50 Hz.

Acknowledgements

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