Improving rolling noise predictions for new track designs

Benjamin BETGEN[1], Nicolas VINCENT;

[1] Vibratec, 28 Chemin du Petit Bois, 69130 Ecully, France.

ABSTRACT

Railway rolling noise dedicated software such as TWINS or STARDAMP perform reliable predictions for vehicles operating on tracks whose wave decay rates have been measured experimentally. On the other hand, accurate noise predictions of new track designs are difficult to achieve due to the limitations of TWINS to perform a precise estimation of vertical and lateral wave decay rates along the rail. A numerical approach allowing to improve TWINS ability to predict rolling noise related to new track designs is proposed and validated in this paper.

Firstly, limitations of TWINS track models are discussed and ways of improvement are pointed out. Then, a numerical approach to compute vertical and lateral wave propagation along the rail is proposed. This approach is based on the development of track F.E. models. It consists in replacing the actual measurement procedure by a 'virtual' measurement procedure on a 'virtual track'.

Comparisons between measured and computed decay rates in third octave bands are then presented for several track designs; the good correspondence between measurements and predictions confirm the relevance of the method.

Keywords: Wheel-rail noise, decay rate, FEM I-INCE Classification of Subjects Number(s): 11.7.2, 42, 43.2.1

1. INTRODUCTION

1.1 Background

Within a speed range from approximately 50 to 300 km/h, rolling noise is the predominant railway noise source. It is excited by the combined roughness of the wheel and the rail which then causes both the wheel and the track to vibrate and radiate sound. Frequency-domain models, notably TWINS (1) are widely used to quantify rolling noise and to study the effect of new designs. However, despite validation and widespread use, there remain aspects of the models and their use that deserve refinement.

Within the ACOUTRAIN project, several such areas have been studied (2). Concerning the low frequency range, these include sleeper radiation and different aspects of ballast behaviour. At high frequencies, the effect of rail cross-section deformation and the effect of rail cross mobility have been investigated. The latter phenomena have been analysed using Finite Element (FE) models of the track. A presentation of this approach can be found in reference (3), including also a first discussion of the use of FE models for the computation of decay rates. This aspect is developed in more detail in the present paper.

1.2 Use of analytical models in rolling noise simulations

The most used track model in TWINS consists of a Timoshenko beam on a continuous elastic support ('rodel' module). This model neglects the discrete support as well as any cross-section deformation of the rail.

The 'tinf' module consists of a Timoshenko beam on a discrete support, thus including the 'pinned-pinned' effect at frequencies where an integer number of half wavelengths equals the sleeper spacing.

The 'perm' module uses mass and stiffness matrices calculated with a FE model of a slice of rail. It therefore permits to reproduce the cross-section deformation of the rail that occurs at higher

[1] benjamin.betgen@vibratec.fr

frequencies (the first lateral bending of the rail occurs at around 2 kHz, the vertical 'foot flap' around 5 kHz).

The 'vibrail' module finally includes both the discrete support and cross-section deformation of the rail. However, it only permits to compute rail receptances and cannot be used for the computation of the complete rail response and sound radiation.

'Tinf', 'perm' and 'vibrail', which demand both a much more complex model setup as well as higher computation times are barely used outside academic research. Indeed, 'rodel' has shown to produce reliable results, especially when providing measured decay rates as an input. For the same reason, this model has also been chosen for STARDAMP (4), which is a tool similar to TWINS.

A typical use of TWINS is the simulation of the pass-by of a new rolling stock on a test track for which decay rates are known. In this case, a reference situation for which measurement data is available is often simulated in a first time. The model can then be tuned to fit the measured pass-by levels, before changing the wheel model to simulate a new rolling stock. The acoustic optimisation of wheels can as well be achieved without reproducing a specific track but by simply minimising the wheel radiated sound power compared to a reference wheel. For instance, geometric modifications of the wheel often permit to reduce wheel noise by several dB, without the use of any damping device. [N.B.: The wheel is always modelled using finite elements, hence TWINS is very sensitive to slight modifications of the wheel design. However, the wheel model is necessarily axisymmetric, which makes it difficult to treat wheels with holes or mounted brake discs for instance.]

The acoustic optimisation of tracks is a very different task and seems to date given less attention. One reason is doubtlessly the absence of limit values in terms of acoustic track behaviour, such as the TSI Noise for rolling stock. Nevertheless, the track contribution of noise is well known to be as important as the vehicle contribution, especially once vehicles have been optimised. Therefore acoustic track optimisation will become more and more important. The acoustic properties of a track can be improved in various ways: increase of decay rates or platform absorption as well as shielding by means of low height noise barriers. All these aspects can potentially be taken into account by numerical models, however, only decay rates are treated in the present paper.

1.3 The use of decay rates

The decay of vibration along the track, or decay rate (DR), can be expressed in dB/m and quantified, as defined in standard EN 15461 (5), by

$$DR = 4.343 \Bigg/ \sum_{n=0}^{n_{\max}} \frac{|A(x_n)|^2}{|A(x_0)|^2} \Delta x_n \tag{1}$$

where $A(x_0)$ and $A(x_n)$ are respectively point and transfer accelerances, and Δx_n is the distance along the track associated with each measurement position x_n. Decay rates are separately measured in vertical and lateral directions.

The decay rate is the most important descriptor of track dynamics with respect to rolling noise. Low decay rates lead to a greater length of radiating rail per wheel-rail contact and thus to high rolling noise emission from the track. High decay rates result in lower noise and can be obtained for example by the use of stiff pads between rail and sleepers. However, soft pads are often used for non-acoustic reasons, e.g. to minimise sleeper damage or ground borne vibrations.

As mentioned above, measured decay rates are widely used for the computation of rolling noise with models such as TWINS. In most situations adopting measured decay rates instead of analytically computed ones results in more accurate noise predictions. As an alternative to measurements this paper explores the possibility of using an accurate FE representation of the track to predict decay rates. As for measurements, equation (1) is used, the transfer functions being calculated instead of measured. Comparisons with decay rates measured on a test track have been made for validation purposes.

2. TRACK FINITE ELEMENT MODELS

Different track types have been modelled in MSC Nastran, including monobloc and bi-bloc ballasted tracks as well a slab track. Bi-bloc sleepers can be represented by concentrated masses, neglecting the relatively soft (in terms of bending) connection between both sleeper blocks. This way, one single rail can be modelled using only half of the bi-bloc sleeper. A monobloc sleeper track including two rails and sleepers modelled using solid elements has also been built. However, the behaviour of this much heavier model is very close to the bi-bloc track (2, 3).

The single rail model has therefore been used in the current investigations, see figure 1. It consists of a rail of 27.6 m length, including an “anechoic termination” of 6 m length (i.e. the length of useful rail is 21.6 m). This termination is obtained by gradually increasing damping of the rail from 0.2% to 100% and prevent any reflections while the length of modelled track remains reasonable. The rail and rail pads are modelled with solid elements, ballast is reproduced by discrete springs (15 springs per rail pad, arranged as 5x3). Displacements of the nodes on the rail cross-section opposite to the anechoic termination are blocked in longitudinal direction. Excitation is applied on this symmetry plane. Note that computed receptances or mobilities have to be multiplied by a factor 2 due to the fact of modelling only half a rail. The computation of decay rates in not affected because only ratios of transfer functions are used.

The modelled ballasted track has rail pads of 150 MN/m stiffness, corresponding to the test track (2, 3). Ballast properties are not known precisely; a constant stiffness of 90 MN/m is assumed here. The ‘half sleepers’ have a mass of 120 kg each.

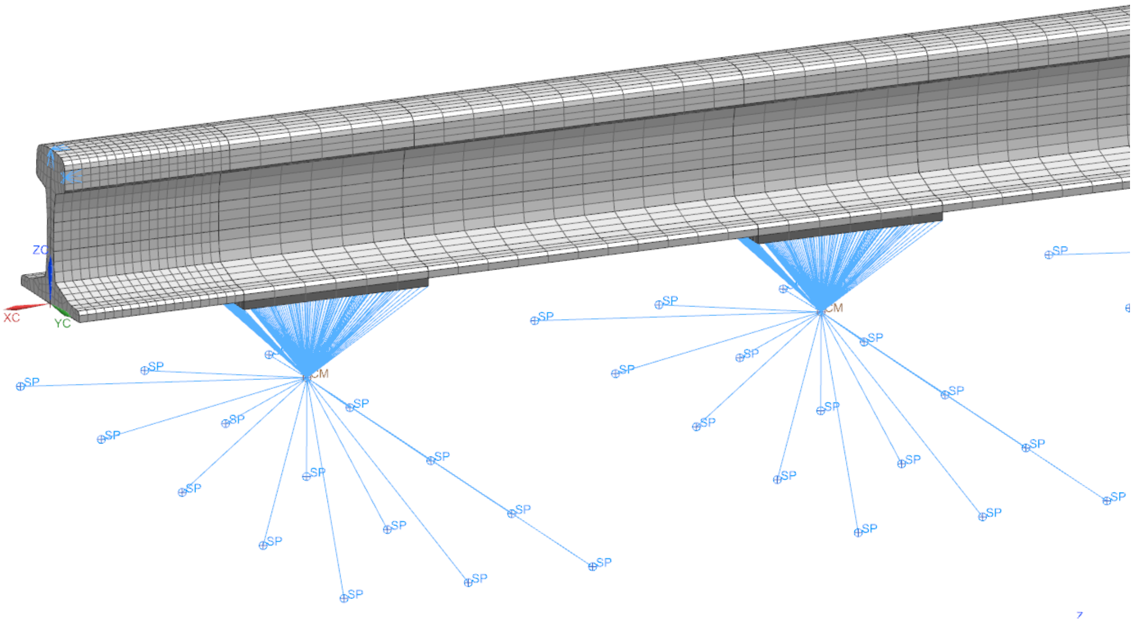

Figure 1 – FE model of a bi-bloc sleeper track.

The model has then be adapted to represent a slab track with a two layer fastening system such as displayed in figure 2. This fastening system features a first rail pad between rail and a load distribution plate of 1000 kN/mm dynamic stiffness. A second elastic layer of only 40 kN/mm is used between this steel plate and the slab. Such fastening systems permit to realise a soft (vertical) fastening while preventing excessive rail rollover due to the larger surface of the load distribution plate compared to the rail foot.

In a first time the mass of the sleeper has been decreased to fit with the mass of the steel plate (assumed to 5 kg) and ballast stiffness has been set to the stiffness of the soft pad. As will be shown this model did not result in a fully satisfactory behaviour, therefore a second model has been built including a solid mesh of the load distribution plates, see figure 3. For convenience, the soft pad between plate and slab is modelled using solid elements as well, whose properties are fitted to reproduce the same pad stiffness as in the previous model. The stiffness of the slab itself is assumed infinite in both models.

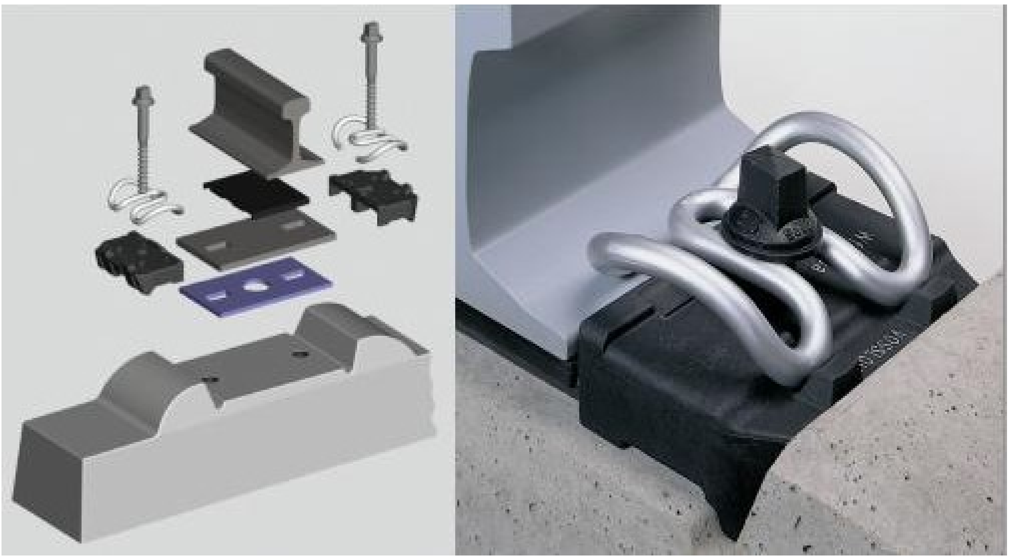

Figure 2 - Fastening system typical for slab tracks (from Vossloh product brochure).

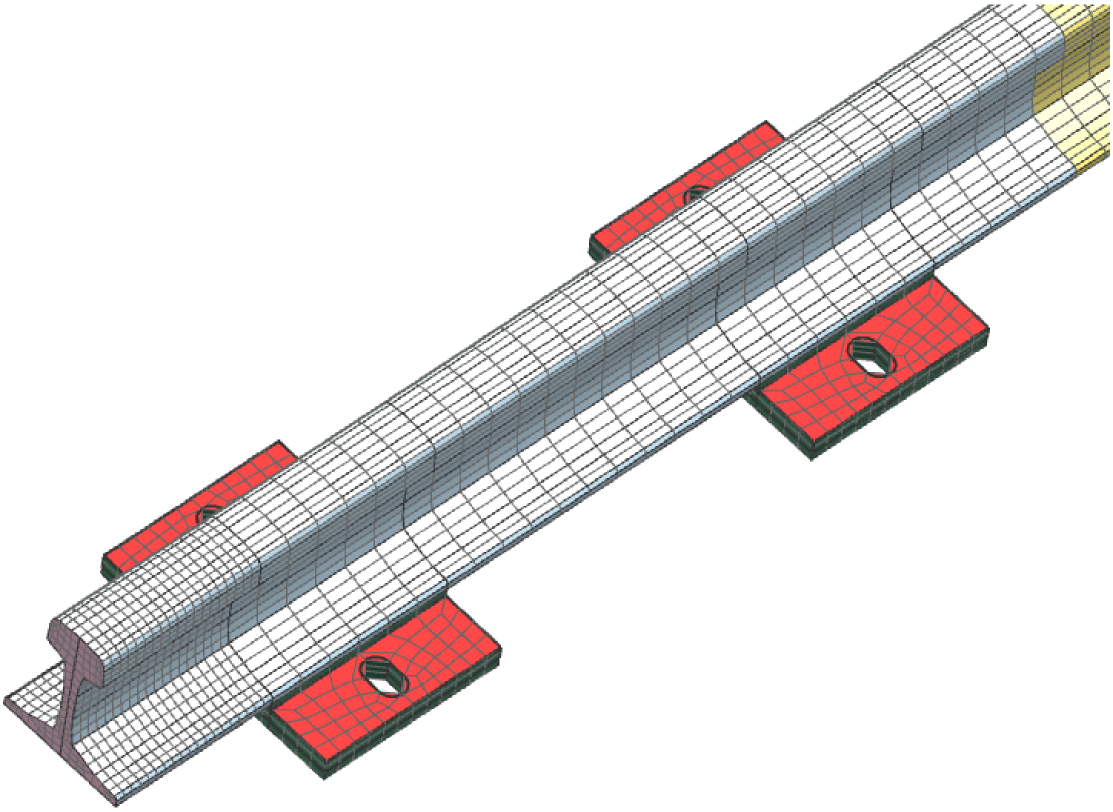

Figure 3 - FE-model of a rail with supports representative for slab tracks

With all models, the length of rail available for the determination of DRs is 21.6 m. According to (5) this is sufficient for an assessment of DRs down to 0.2 dB/m.

The post-processing of decay rates from the FE model has been performed analogously to measurements, i.e. by following the procedure of EN 15461 (5).

3. RESULTS

3.1 Ballasted track

Measured and computed decay rates of a ballasted track are displayed in Figure 4. All main phenomena are well reproduced by the simulation: At low frequencies decay rates around 10 dB/m are observed which are due to the coupling between rail and sleepers. The frequency band of 315 Hz roughly corresponds to the resonance of the rail mass (36 kg/support) on the pad stiffness (150 kN/mm), above which waves begin to propagate freely along the rail. Increasing energy dissipation in the fixation system and the foot flap mode at 5 kHz (which in turn leads to increased energy dissipation in the rail pad) leads to an increase of decay rates towards higher frequencies.

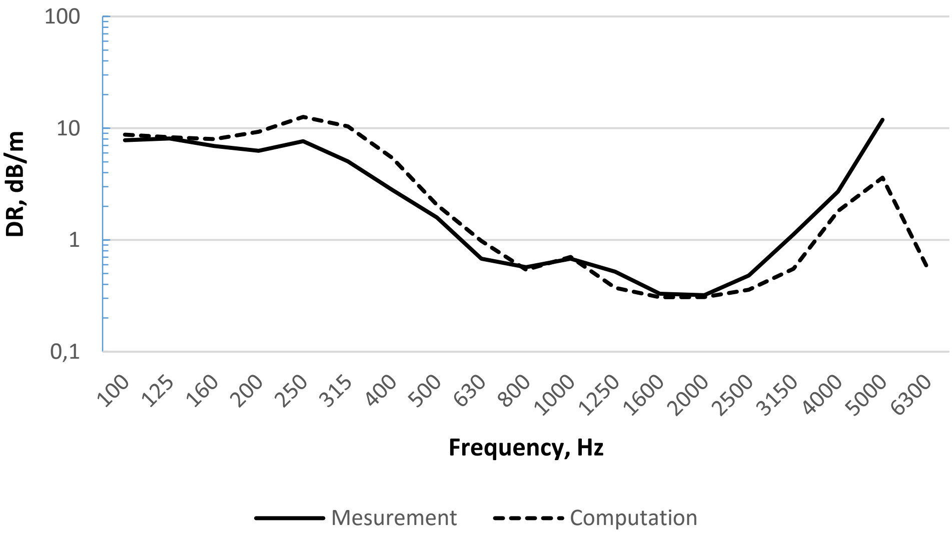

Figure 4 – Measured and computed vertical decay rates of a ballasted track

3.2 Slab track

In a first time the ballasted track model has been used with minor modifications to determine decay rates of a slab track, as described above. Computation results in comparison with measurements are shown in Figure 5. Especially below 500 Hz, decay rates are over-estimated, i.e. decoupling between base-plates and slab is obviously predicted too high.

Figure 6 shows results obtained with a model including base plates modelled using solid elements (model of Figure 3). The low frequency behaviour is improved despite the use of identical stiffness data. The bending of the plate obviously leads to a lower decoupling between rail and slab in comparison to the use of a point mass. According to Figure 7, the first vertical bouncing mode of a 60 cm piece of rail on the two layer support is at 220 Hz, involving a clear bending of the plate. The plate therefore has to be introduced for a correct prediction of decoupling between slab and rail. The first bending mode of the plate itself (but in presence of both elastic layers) is at 630 Hz, leading to high energy dissipation in the elastomeric material. The measured decay rate shows an increase in this frequency region as well; however, the effect is not as localized but distributed in the 500 Hz – 1 kHz range.

The obtained fit is not yet fully satisfactory, but improved with respect to the simpler model. The explicit consideration of the rail fixation system might be a way to improve the model. On the other hand, decay rates of tracks with this type of soft fixation will always remain difficult to predict. Indeed, significant differences can often be measured between two sites with nominally identical fixation systems. Also, temperature fluctuations can induce important variations of decay rates.

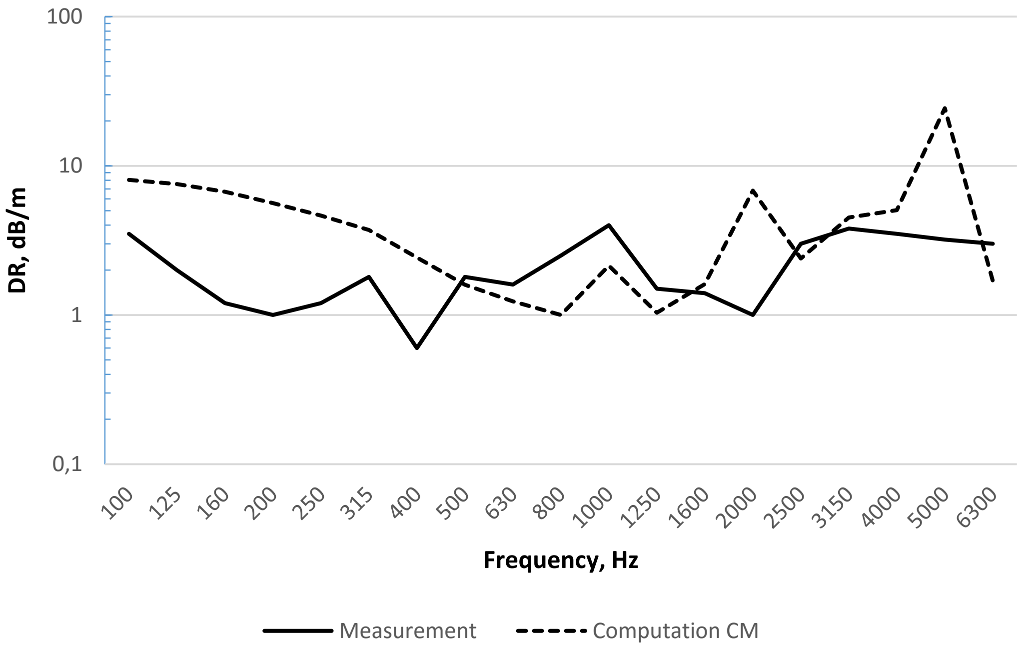

Figure 5 – Measured and computed vertical decay rates of slab track – simplified model

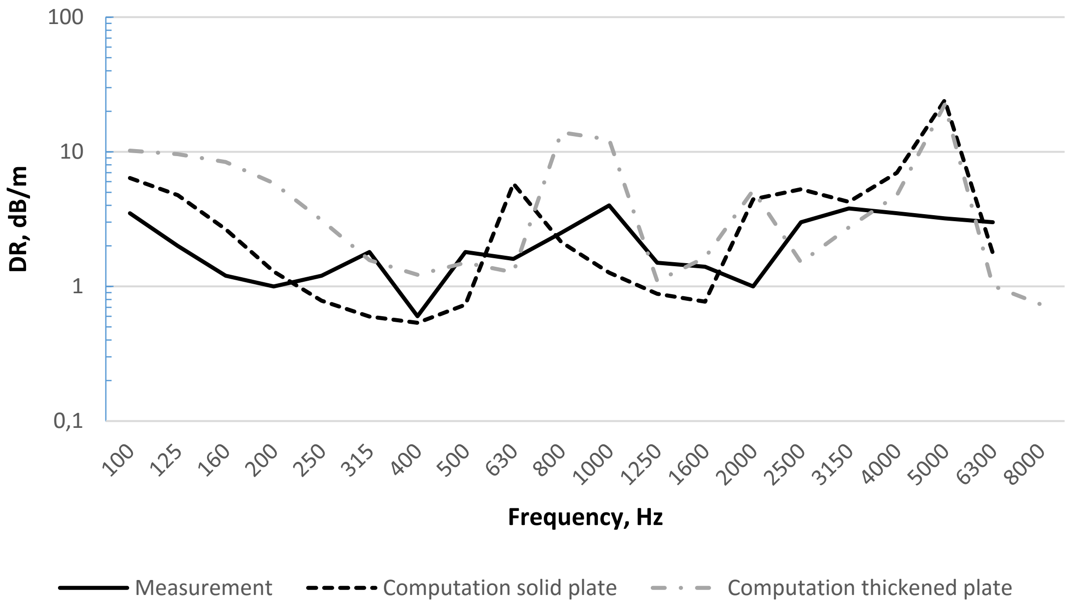

Figure 6 – Measured and computed vertical decay rates of a slab track – model including solid plates

Despite certain imperfections, the model can be used for parameter studies. One example is given in Figure 6, where the thickness of the baseplates has been increased from 16 mm to 24 mm. Clearly the low frequency behaviour of the track is improved and the peak related to plate bending is shifted to higher frequencies. The baseplate may therefore be tuned to shift the maximum decay rate in a frequency range where track noise radiation is highest.

On the other hand, further investigations are necessary to assess the potential radiation from the baseplates themselves. Indeed, measured pass-by levels on slab track are often higher than (measured) decay rates would suggest. This behaviour cannot be explained by lower noise absorption (due to missing ballast) alone but may be due as well to high noise radiation from rail fastening systems using base plates.

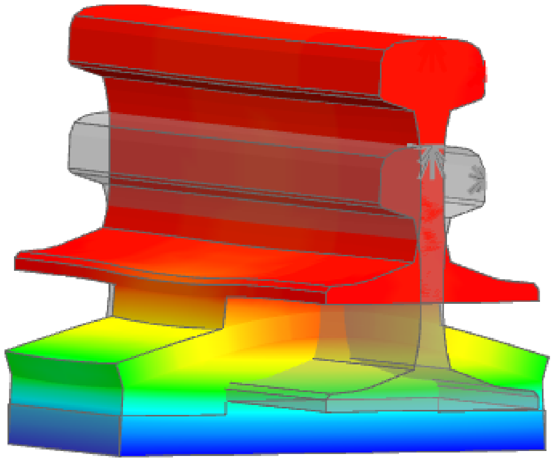

Figure 7 – Vertical mode at 220 Hz of a piece of rail (60 cm) on a double layer support

4. CONCLUSIONS

Finite Element Models have been shown to be suitable for the prediction of decay rates of conventional tracks, reproducing correctly relevant phenomena such as the pinned-pinned effect or cross-section deformation. As the sleepers do not modify the high frequency behaviour of the track, their representation as concentrated masses seems sufficient

The modelling of a double layer support such as used with slab tracks has shown that the bending modes of the load distribution plate considerably modify the decay rate. Their representation as concentrated masses is therefore not sufficient. However, despite the introduction of solid plates the fit between measurements and simulations remains less good than with conventional ballasted track. Further improvements may be possible through a more detailed representation of the fastening system. However, according to the authors experience measured decay rates for different slab tracks with nominally identical or very similar characteristics (pad stiffness, etc.) also show higher discrepancies as it is the case for ballasted tracks. The definition of input data for simulations is therefore a difficult task.

Despite certain differences between simulated and measured decay rates, Finite Element Models seem suitable for the optimization of the acoustic behaviour of tracks. As an example, a computation with thicker load distribution plates has shown a significant impact on decay rates.

REFERENCES

1. Thompson D.J., Hemsworth B., Vincent N., Experimental validation of the TWINS prediction program for rolling noise, part 1: description of the model and method. *Journal of Sound and Vibration* 193(1), 123-135, (1996)
2. Squicciarini G. et al.: Acoutrain deliverable 2.7 – Improved model components, ACOUTRAIN project, 2014, accessible at http://www.acoutrain.eu/wp-content/uploads/2015/08/D2_7_Improved_model_components.pdf.
3. Betgen B., Squicciarini G., Thompson D.J., On the prediction of rail cross mobility and track decay rates using Finite Element Models, Proceedings of Euronoise, Maastricht 2015
4. Betgen B., Bouvet P., Squicciarini G., Thompson D.J, Jones, C.J.C. Estimating the performance of wheel dampers using laboratory methods and a prediction tool, *Notes on Numerical Fluid Mechanics and Multidisciplinary Design* 126 pp 39-46, (2015)
5. Standard EN 15461:2008+A1:2010:E. Railway applications. Noise emission. Characterization of the dynamic properties of track selections for pass by noise measurements.