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Open source vibro-acoustic finite element modelling of rail tracks components

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Abstract

Two finite element models simulating vibrations and noise emissions of rail tracks components are presented. Dynamic analyses are performed in the frequency domain and using 3D elements to quantify and compare the response of rail tracks to typical excitation conditions. Acoustic pressure fields are obtained using monopole sources superposition. One model is the digital twin of a three-sleeper lab-scale experimental setup. The other model is based on dynamic substructuring to simulate much longer tracks in a computationally efficient way. Both models provide satisfactory results in the range of interest (300Hz to 1500Hz) and were used to evaluate rail pads performances regarding noise emissions and ballast protection.

Keywords: finite element modelling, vibrations, acoustics, dynamic substructuring.

1 Introduction

In the framework of the project "Novel Rail Pads for Improved Noise Reduction and Reduced Track Maintenance" conducted by FOEN (Federal Office for Environment), the University of Applied Science HEIG-VD / HES-SO studied how rail track components influence rolling noise. Indeed, the latter is known to come from the wheels and rail track components vibrations induced by the surface roughness of wheels and rails [1,2].

Parametric finite element models were developed to simulate the acoustic power radiated by the ballast, sleepers and rails vibrations in the frequency domain. The first one is a 3D solid model representing a three-sleeper lab-scale experimental setup. The second model uses dynamic substructuring to simulate rail tracks with much more sleepers, in a computationally more efficient way. Both models compute acceleration frequency response functions (FRF) at various points of the setup, as well as the total radiated acoustic power in a chosen frequency range. They also provide the acoustic power spectrum radiated by each component, which is very difficult to extract experimentally. More generally, these parametric models allow studying the influence of various parameters on the dynamic behavior of rail track systems.

The motivations behind these developments are, firstly, to provide better fidelity than beam models by representing the full 3D mode shapes. Moreover, existing models generally use spring-dashpot elements for the rail pads and cannot capture the effect of the pad geometry [3,4].

The monopole superposition principle is used to compute the acoustic pressure fields, based on the velocity field on the radiating surfaces. Finally, using dynamic substructuring drastically reduces computation time and allows simulating long rail tracks with good fidelity in 3D based on the geometry and material properties of the components. The models presented in this article were developed with Code_Aster, an open source Finite Element Analysis (FEA) software freely available on www.code_aster.org.

This article presents the process of developing the models and validating them with respect to experimental measurements. Some examples of what they can achieve are presented as well as the results of a parametric study of the influence of rail track components mechanical properties on noise emissions.

2 Methods

The three-sleeper cell model performs harmonic-acoustic simulations on the digital twin of the actual experimental set-up. Under-sleeper pads can also be included optionally. The workflow is described on Figure 1. Every part is modelled with linear hexahedral elements and the interactions between components are considered perfectly bonded contacts. The loading is a unit harmonic force at the tip of one rail at an angle of either 45° inward or 5.7° outward (
Figure 2). The first load represents loading conditions in a curve, while the second illustrates a pass-by on a straight line.

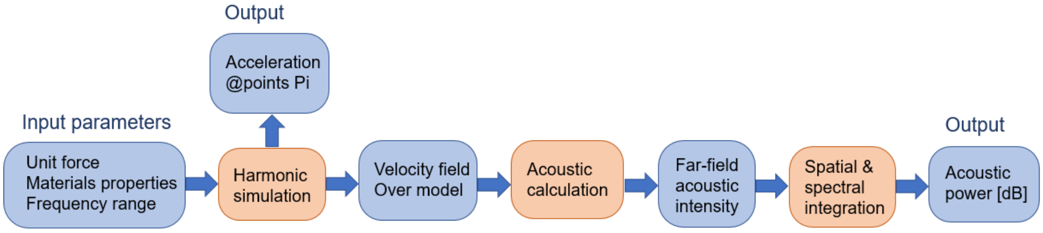

Figure 1: Workflow of the three-sleeper model.

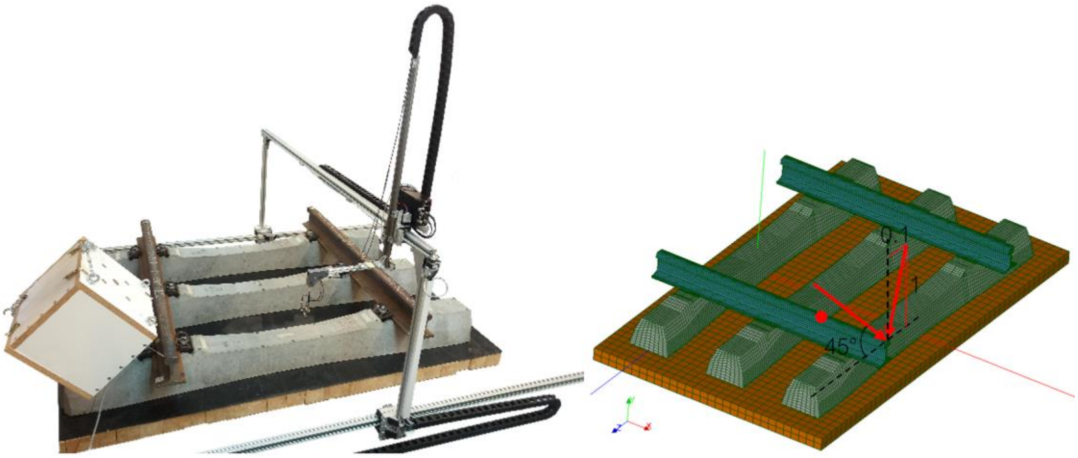

Figure 2: Three-sleeper experimental setup and numerical model.

All materials are linear, viscoelastic with hysteretic damping and the pad material properties and ballast are treated as frequency-dependent. The mechanical properties of the sleepers, rails and ballast were identified on experimental modal measurements while the rail pads material properties were derived from DMA measurements.

At each frequency, harmonic simulations provide the velocity field over a set of radiating surfaces. Each surface element is considered as a perfect monopole acoustic source and the superposition of all contributions provides the acoustic pressure at any point. The peak pressure generated by a monopole source can be computed as follows.

$$\hat{p}(r) = \frac{\rho c k v_{\perp} S}{4\pi r} \tag{1}$$

The monopoles superposition validity was demonstrated at a distance greater than four times the characteristic size of the model.

Dynamic substructuring consists in creating a macroelement instanced several times to build a larger model (Figure 3). The goal is to run a harmonic analysis on a long rail track, using a “rich enough” modal basis of the macroelement, composed of eigenmodes and static interface modes with the Craig-Bampton method. The multi-sleeper model performs frequency-domain, linear viscoelastic computations. An equivalent Rayleigh damping is implemented whose parameters are optimized to match hysteretic damping on a frequency band.

To simulate frequency dependent stiffness and damping in the macro element, the mode shapes are assumed to remain constant with respect to rail pad properties. At each frequency, the actual stiffness and mass matrix of the macro-element are projected on this fixed modal basis. A modal analysis is performed in generalized coordinates and results are projected back on the physical basis to obtain an updated macro element around a given frequency. Interface static modes are also added.

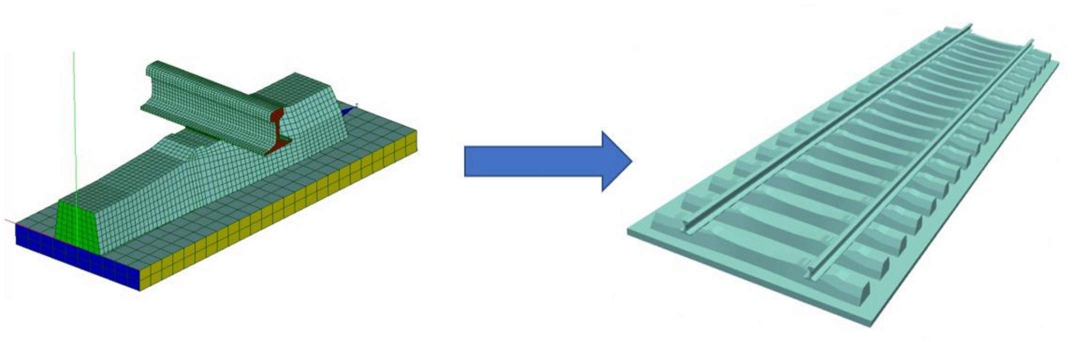

Figure 3: Macro-element and multi-sleeper model representation.

3 Results

For validation purposes, the accelerance FRF obtained at a point shown in Figure 2 are compared between the model and experiment. Note that the multi-sleeper model is used here with six macroelements, hence three sleepers. With the 45°-oriented load, which excites many modes, the results are excellent. As for the 5.7° load, the curves do not match as accurately, but the main features of the spectrum and the amplitude of the response are well represented.

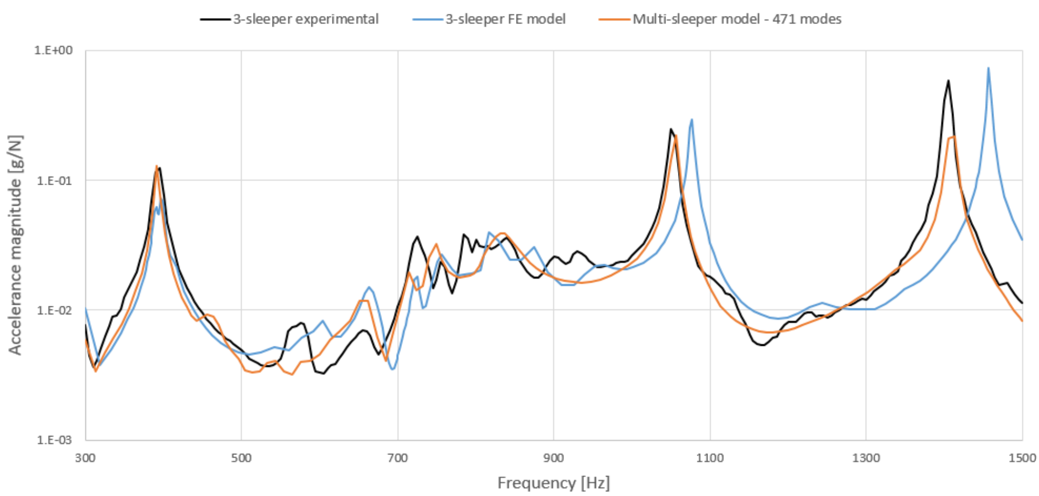

Figure 4: Accelerance FRF (EVA pads, 45° load).

On Figure 6 and Figure 7 are presented the spectrums of the total radiated acoustic power for both load directions. These curves are obtained by integrating the acoustic intensity $I = p^2/\rho c$ over the acoustic grid (in this case the hemisphere in Figure 8). Again, the numerical results match the experimental ones more accurately with the 45° load. It is important to note that when doing comparative analyses, having perfectly-matching curves is not necessary.

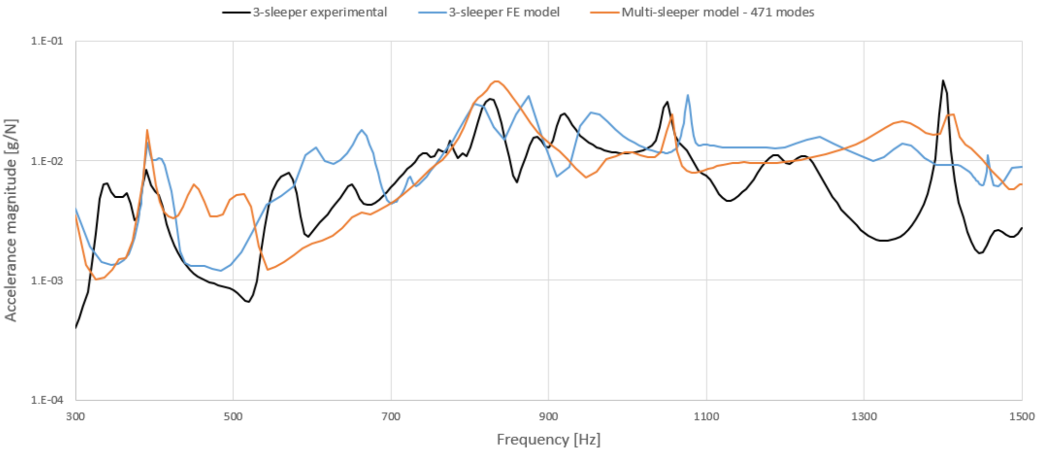

Figure 5: Accelerance FRF (EVA pads, 5.7° load).

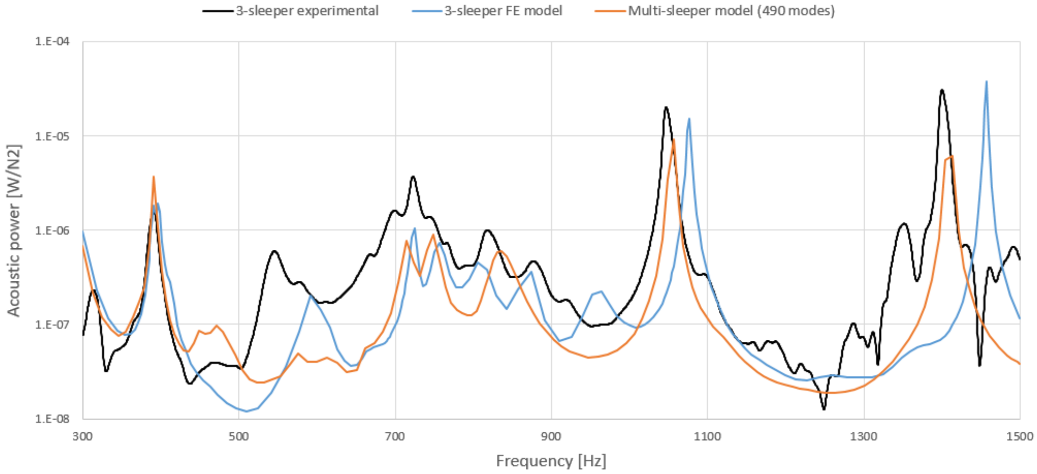

Figure 6 Acoustic power FRF (EVA pads, 45° load).

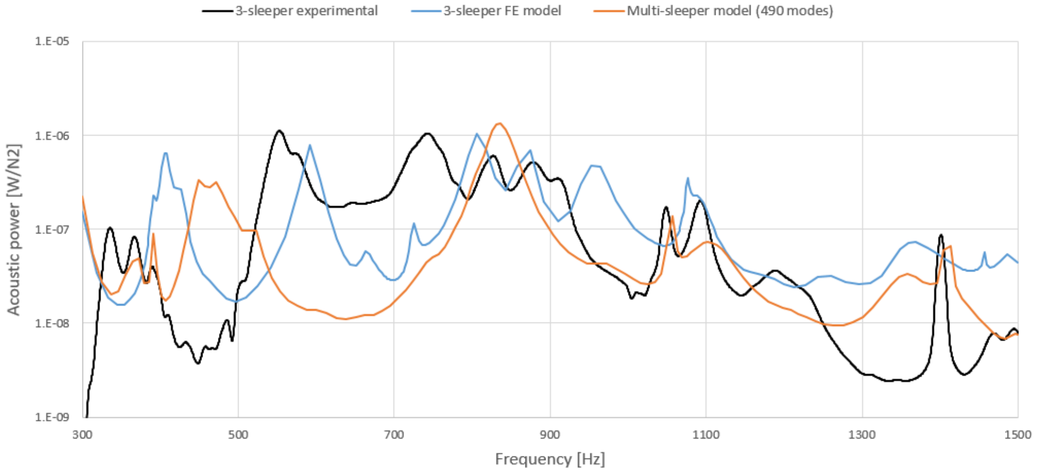

Figure 7: Acoustic power FRF (EVA pads, 5.7° load).

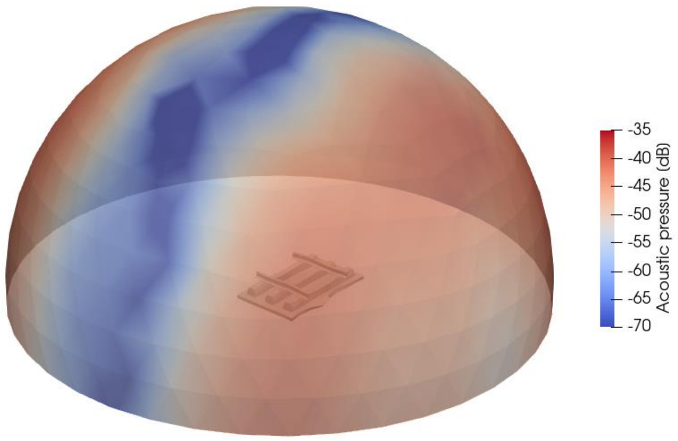

Figure 8: Acoustic pressure field.

A parametric study was performed to determine the most influent parameters on noise emissions, using the three-sleeper model with under-sleeper pads (USPs). The parameters considered are shown in Table 1. Each simulation involves setting all parameters but one to their reference value.

Parameter	**Reference value**	**Value when varied**
USP	Included	Removed
E ballast	5 MPa	2.5 MPa
TanD ballast	0.6	0.3
E pad	1 (coefficient)	0.5 (coefficient)
TanD pad	1 (coefficient)	2 (coefficient)
Nu pad	0.484	0.4
E USP	6 MPa	12 MPa
TanD USP	0.25	0.15

Table 1

The acoustic power L_w was computed for each simulation as the integral under the spectrum between 300Hz and 1500Hz. It is expressed in dB(W/N^2) and can be helpful to compare systems. Figure 9 represents the dimensionless derivative of L_w with respect to the parameters x_i.

$$\left.\frac{\partial L_w}{\partial x_i}\right|_{x_{i0}} \cdot \frac{x_{i0}}{L_{w,0}} \tag{2}$$

The multi-sleeper model was firstly simulated with six macroelements to validate it with respect to the existing three-sleeper model with a large number of modes (about 1400) and the same load cases and parameters. The components of the generalized coordinates vectors are analyzed to extract the maximum of their contribution over the frequency band (Figure 10). The obtained envelope indicates that most modes do not contribute much to the response. Hence, an optimized modal basis is created by selecting the union of the N most influent modes for each loading case. Here, $N = 400$ is a good compromise between accuracy and computing time.

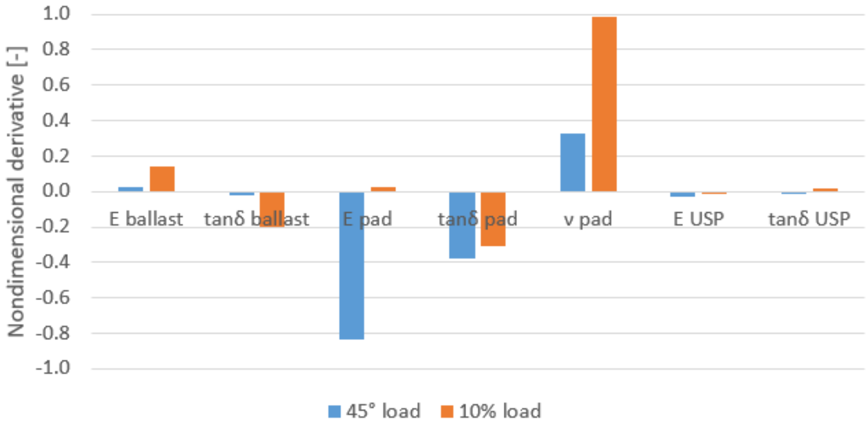

Figure 9: Parameters influence on noise emissions (300-1500Hz).

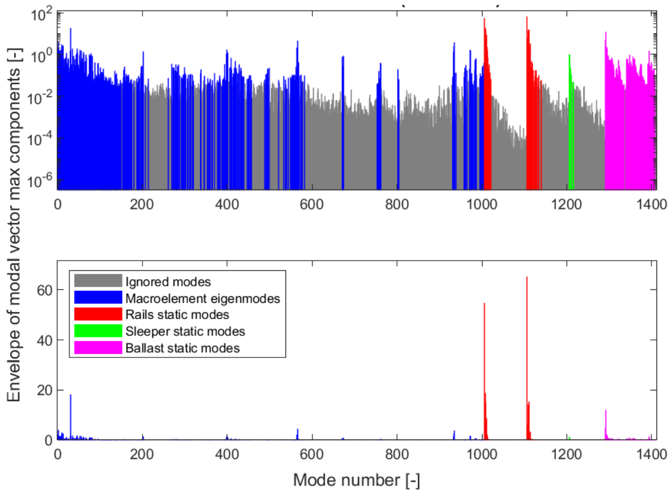

Figure 10: Modal basis definition (471 modes).

4 Conclusions and Contributions

The three-sleeper model has been used to systematically test the pads prototypes which were designed during the “Novel Rail Pads for Improved Noise Reduction and Reduced Track Maintenance” project. It is first and foremost aimed at being used for comparison purposes, since the trends are generally well represented between systems, but the absolute results might not match the experiments perfectly.

One of its main limitations is the contact modelling between components, in particular between pads and rails or sleepers. The “no-slip” condition implies that the pads eventual texturing or patterns have very little influence on the dynamic behavior of the track, whereas experiments demonstrate the opposite. Another limitation is the computational cost of the simulations. It takes about 20 hours to simulate a full frequency sweep, and simulating a more realistic case, a 20-sleeper-long model for instance, would take weeks.

For a longer track, the added value of the multi-sleeper model: using dynamic substructuring saves a lot of computational of time while preserving the overall fidelity of the predictions. And since the longest part of a simulation is the computation of the macro-element modes, simulating a longer model does not make a large difference in computation time. Namely, to simulate a three-sleeper-long model, using dynamic substructuring is about three times faster with equivalent parameters, and this factor grows exponentially when increasing the number of sleepers.

These validated models will be published in open-source as a part of the large software toolbox for rail track performance prediction and optimization. As experienced during our own rail pad developments, these software prediction tools provide an efficient basis for optimizing railway components where complex tradeoffs have to be found between noise, vibration and ballast protection requirements.

Acknowledgements

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