Proceedings of the Seventh International Conference on
Railway Technology: Research, Development and Maintenance
Edited by: J. Pombo
Civil-Comp Conferences, Volume 15, Paper 18.8
Civil-Comp Press, Edinburgh, United Kingdom, 2026
ISSN: 2753-3239, doi: 10.4203/ccc.15.18.8

Iterative Finite Element Model Validation of a Railway Track System through Experimental-Numerical Modal Correlation

I. Bravo[1,2], A. Astiazaran[1,2], J. Nieto[1,2] and P. Ciáurriz[1,2]

[1]Ceit-Basque Research and Technology Alliance (BRTA), Donostia – San Sebastián, Spain
[2]Tecnun, Universidad de Navarra, Donostia – San Sebastián, Spain

Abstract

This article describes a specialized computational framework designed to automate the calibration of Finite Element Models for railway infrastructure. By integrating MATLAB's optimization tools with Abaqus software, the system iteratively adjusts stiffness parameters for ballast and rail pads to match experimental vibration data. The methodology employs the Modal Assurance Criterion and the optimization algorithms to minimize discrepancies between numerical predictions and real-world measurements. A multi-start strategy is implemented to ensure that the calibration process converges toward global rather than local minima. Detailed architectural diagrams and technical analyses illustrate a modular hierarchy of scripts that handle everything from data normalization to the automated execution of simulation macros. Ultimately, this framework provides a high-precision tool for ensuring the structural integrity and dynamic fidelity of critical rail transport models.

Keywords: railway infrastructure, experimental modal analysis, parameter identification, structural dynamics, finite element model calibration, modal assurance criterion.

1 Introduction

The dynamic characteristics of the track define the response of the system to rolling irregularities. Knowledge of the dynamic properties is necessary to study the generation of noise and vibrations caused by defects in running surfaces, fastening systems, sleepers or ballast bed.

The structural characteristics and type of track determine the track's dynamic behaviour, which has been the subject of study by several researchers [1]. To know with precision the dynamic behaviour of a given section of track, it is necessary to perform an experimental characterisation because this behaviour can vary for many reasons (settling of the ground, ballast compaction, loosening of the rail fastenings, etc.). The dynamic behaviour of the track can be characterized through in-situ measurements, for example, by conducting an experimental modal analysis (EMA), and comparing the results with a numerical model.
Among other applications, modal analysis [2], [3] is heavily used to analyse any critical structure that is exposed to forces that might induce harmful or even destructive resonant frequencies with little damping, and can give an overview of the natural frequencies, damping parameters, and structural mode shapes [4]. It is a fundamental technique in structural engineering that allows the natural frequencies and vibration modes of a structure to be identified.

Modal test and analysis typically involve:

- One or more exciters (modal shaker or impact hammer)
- Force transducers to acquire the input excitation signals
- Accelerometers to acquire the output response signals
- A data acquisition device to display and record the test
- A modal test and analysis software application

To complement these experimental campaigns, numerical modelling, commonly through Finite Element Models (FEM), is used to simulate the track's dynamic behaviour and analyse varying structural conditions. Nevertheless, an initial numerical model often contains theoretical assumptions regarding complex components like the ballast and rail fastenings that may not match real-world conditions. Therefore, it is critical to integrate the experimental and the numerical approaches. By correlating the empirical data obtained from the modal tests with the initial numerical simulations, the model can be iteratively updated, ensuring its parameters accurately reflect the physical track.

Using the results of an EMA as a starting point, this article describes the process of fitting a numerical track model, outlining the methodology used in the experimental test and later data processing, the generation of the numerical model and its subsequent validation, and concluding with a general discussion and conclusions.

2 Experimental methodology

This section presents the outcomes derived from the experimental measurements of the dynamic response of the track at the Adif station in Altsasu (Navarra, Spain). The main characteristics of the track are shown in Table 1.

Track type	Ballasted track
Track gauge	1668 mm
Sleeper type	MR-93
Rail	UIC 54E1
Fastener type	SKL-1
Rail inclination	1/20

Table 1: Main track characteristics.

The followed methodology was described in previous papers related to the study and detection of failed fasteners [5], [6] and damaged railways [7], based on the idea that a damage on a track component will change system's stiffness, inertia, or energy dissipation properties, and therefore, the measured dynamic response of the system.
As mentioned by David Ewins [3], the most common application of EMA is to measure a structure's vibration characteristics and compare them to matching data generated by a finite element (FE) or another theoretical model. This way, all that is required from the test are accurate estimates of natural frequencies and damping (eigenvalue) and descriptions of mode shapes (eigenvector) with just enough precision and accuracy to identify and correlate them with those from the theoretical model. As a result, this correlation can be employed to create a model capable of predicting the impacts of changes to the original structure as evaluated.
One of the most common ways of exciting a structure is the use of a hammer. The equipment is just an impactor with a variety of tips and heads to extend the frequency and force levels for evaluating a variety of structures. A load cell, or force transducer, is typically included with the impactor to detect the magnitude of the force felt by the impactor, which is assumed to be equal and opposite to that experienced by the structure.

The magnitude of the impact is controlled by the hammer head's mass and velocity when it strikes the structure. Frequently, the operator will regulate the velocity rather than the force level itself, hence a suitable approach of modifying the order of the force level is to vary the mass of the hammer head. The rail was excited by impacts made with an instrumented hammer, while the response was recorded in the form of accelerations using accelerometers fixed at different points on the structure (Figure 1).
The extensive length of the track implies that it can be regarded as effectively infinite, resulting in its behaviour differing from that of a typical finite structure. From the perspective of excitation forces, given that it is an infinitely long beam, it is not feasible to apply the energy required to excite the entire structure, and any excitation ultimately diminishes before reaching the ends. However, despite these restrictions, modal analysis tools have been employed to obtain the different modes of vibration and modal deformations of the track.

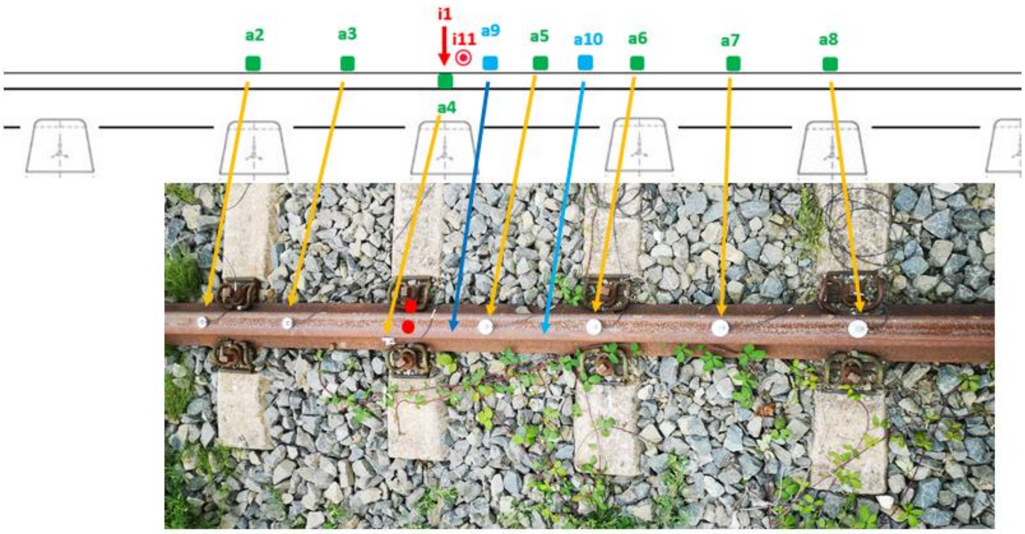


Figure 1: Layout showing measurement and impact points

The signals from the accelerometers and the hammer's load cell were captured and analysed to obtain the frequency response functions. The equipment used during the tests is described in Figure 2:

- Two data acquisition systems DEWESoft SIRIUSi
- Nine accelerometers Brüel & Kjaer type 4535-B
- A hammer with a force sensor Dytran 1051V4

Figure 2: Equipment used during the test: Data acquisition system (a), accelerometer (b) and hammer force sensor (c)

The analysis includes a frequency range from 0 to 1500 Hz, captured both vertically and horizontally. The excitation consisted of striking the head of the rail at impact points i1 in the vertical direction and i11 in the lateral direction, with the accelerations being recorded at points a2, a3, a4, a5, a6, a7, a8, a9 and a10 (Figure 1).

3 Processing and analysis of experimental data

As mentioned in section 2, the analysis includes a frequency range from 0 to 1500 Hz, captured both vertically and horizontally, although as shown in Figure 3, the coherence of the signals obtained is not good below 30 Hz because the excitation applied is not effective at such low frequencies.

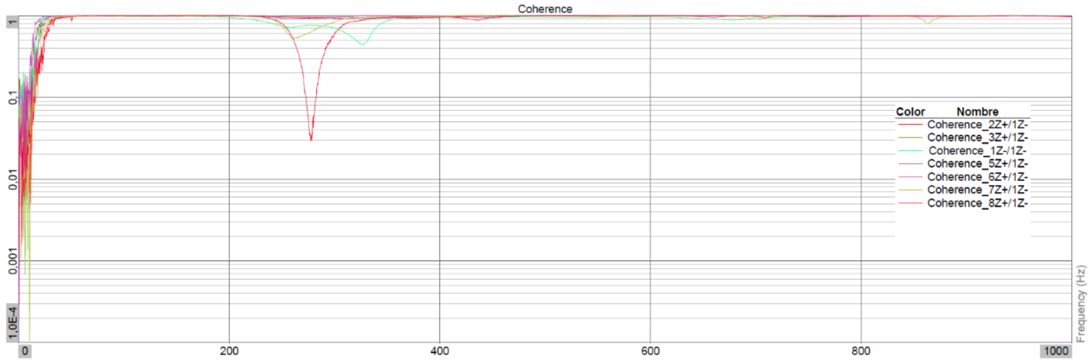


Figure 3: Response coherence, plotted up to 1000 Hz

The generated data consists of a series of force and acceleration signals, coherences, and Frequency Response Functions (FRFs) [8]. The magnitude of an FRF is often in units [m/s^2/N].

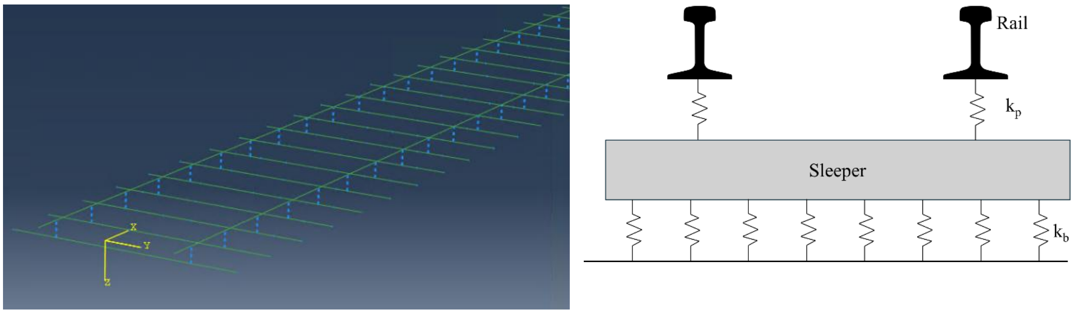
Correct differentiation between genuine and fictitious modes remains a critical task in modal testing. The first step consisted of estimating the vibration modes using a stabilisation diagram, which is helpful to select the poles, with the help of the Complex Mode Indicator Function (CMIF) [3]. The poles are the estimated natural frequencies and damping values of a system, obtained through modal analysis.

Once the poles are selected, we obtain the estimated global modal parameters, being damped resonance frequencies and damping ratios listed for all selected modes, as well as the Mode Complexity Factor (MCF), which defines how well all the DOFs for a given mode deflects totally in or out of phase compared to each other. We also obtain the AutoMAC graph, useful to verify that the selected and estimated modes are uncoupled from each other in the estimated model. These calculations come from the use of the FRFs applied to the registered acceleration signals.

The first mode was detected at approximately 130-150 Hz. As noted by other authors [5], [6], [9], the resonant frequency band of the rail and fastener is found between 450-550 Hz for the track-fastener system and around 700-790 Hz for the track-sleeper system. The first order bending resonance frequency of the sleeper is about 1150 Hz, and the pin-to-pin resonance is numerically obtained at approximately 1050 Hz.

4 Numerical model

The numerical simulation of the track system is conducted using the commercial FE software Abaqus (v2021), employing discrete supports. The model represents a 15.6-meter section of a standard ballasted track with an Iberian track gauge (1668 mm), using 3D beam elements (see Figure 4) for both rails and concrete sleeper bodies. This allows the representation of the bending stiffness and inertial properties while maintaining the computational efficiency for dynamic analyses. Each sleeper is supported by a system of independent bushings that represents the stiffness of the ballast. Each connection between sleeper and rail is modelled by a single bushing element representing the rail fastening. The model restricts the longitudinal degree of freedom, which simplifies the model and focuses the analysis on the vertical and lateral plane responses.



(a) (b)

Figure 4. 3D (a) and 2D (b) representation of the track model

4.1 Model parameters and configuration

The structural behaviour of the track components is defined by their geometric and material properties, which are summarized in Table II. Linear elastic properties were assigned to the rail and sleeper. Both the rails and sleepers are modelled using linear elastic properties. For the 54E1 rails, given their complex cross-sectional geometry, the model explicitly assigns the corresponding cross-sectional area (A) and moments of inertia (Iy, Iz). Conversely, the sleepers are defined using a simplified trapezoidal cross-section.

Component	Parameter	Symbol	Value
Rail – 54E1	Young's Modulus	Er	210 GPa
	Vertical moment of inertia	Iz	2337.9 cm^4
	Lateral moment of inertia	Iy	419.2 cm^4
	Area	Ar	69.77 mm^2
	Density	ρr	7850 kg/m^3
	Poisson's Ratio	ν r	0.3
Sleeper – Trapezoidal section	Young's Modulus	Es	23 GPa
	Long base	b	300 mm
	Short base	a	168 mm
	Height	h	249 mm
	Density	ρs	2400 kg/m^3
	Poisson's Ratio	νs	0.2

Table II. Mechanical and geometric parameters of the FE track model

The global response of the track relies on the flexibility of its support layers, which are simulated using discrete bushing elements. Specifically, the rail-sleeper interaction is modelled by a single bushing element per rail seat to represent the fastening system (k_p). Beneath the track, the ballast bed is discretized into eight independent bushing elements distributed under each sleeper (k_b). These supports are configured as purely elastic components. Consequently, viscous damping is intentionally omitted. The baseline stiffness parameters assigned to these elements, summarized in Table III, are adopted from the validated data utilized by [10] for the study of train wheel polygonalization.

Parameter	X (MN/m)	Y (MN/m)	Z (MN/m)	ϕx (kNm/rad)	ϕy (kNm/rad)	ϕz (kNm/rad)
K_p	-	17,6	220	282	469	260
K_b	-	8	30	3906	196	10000

Table III. Parameters of the track support model: stiffness in the translation and rotation directions for pad and ballast.

These simplifications are in line with previous studies that have demonstrated their validity for preliminary modal analyses on railways [9].
To evaluate the dynamic characteristics of the track model, a modal analysis was performed extracting natural frequencies up to 1500 Hz. This allowed for a detailed characterisation of the dynamic behaviour of the rail in the frequency range of interest focusing on identifying the fundamental vertical and lateral vibration modes. Results obtained made it possible to evaluate the model's capability to reproduce the dynamic behaviour observed in the test and served as the basis for the validation described in the following section.

5 Model updating

According to [3], verification of a model is the process of determining whether a given model can describe the behaviour of the subject structure, provided all individual model parameters are assigned the correct values. Validation of a model is the process of demonstrating or achieving the condition that the coefficients in a model are sufficiently accurate to enable the model to provide an acceptably correct description of the subject structure's dynamic behaviour. A model that is not verified cannot be validated, and such a validation procedure should not be attempted on an unverified model.

FEM updating in contemporary railway engineering is a fundamental process to ensure the fidelity of dynamic predictions and the structural integrity of the track. To assess the accuracy of the numerical model, a detailed comparison was performed between the modal parameters obtained experimentally and those predicted numerically. The validation focused on the first several modes, which are most relevant for dynamic behaviour in railway applications.

The system utilizes an optimization architecture based on the 'fmincon' local search function in MATLAB and the Modal Assurance Criterion (MAC), to iteratively adjust ballast and rail pad stiffness parameters. Through a modular structure of scripts, the framework automates the complete cycle of parameter updates, simulation execution, result extraction, and convergence evaluation, providing a high-precision tool for model validation against experimental data.

5.1. Optimization System Architecture

The framework is organized into a modular multi-level hierarchy to ensure total process automation:

I. **Input Phase**: The complex modes of EMA testing are loaded by the main script and, after a 'realification' process [11], are normalized and ready to be used.

II. **Optimization Loop**: The main script orchestrates the process, defines search boundaries, and implements a multi-start loop, which generates random starting vectors within predefined limits (typically 50% of the nominal value), vital for exploring regions of the search space that might otherwise be hidden by premature convergence toward sub-optimal local minima. The objective function iteratively updates parameters, executes Abaqus, reads the results, and calculates the modal score.

III. **Post-processing Phase**: After convergence, the system generates AutoMAC and Cross-MAC 3D matrices, frequency comparisons, and curvature (UR) diagnostics for final validation.

This FEM Updating (FEMU) framework workflow is shown in Figure 5.

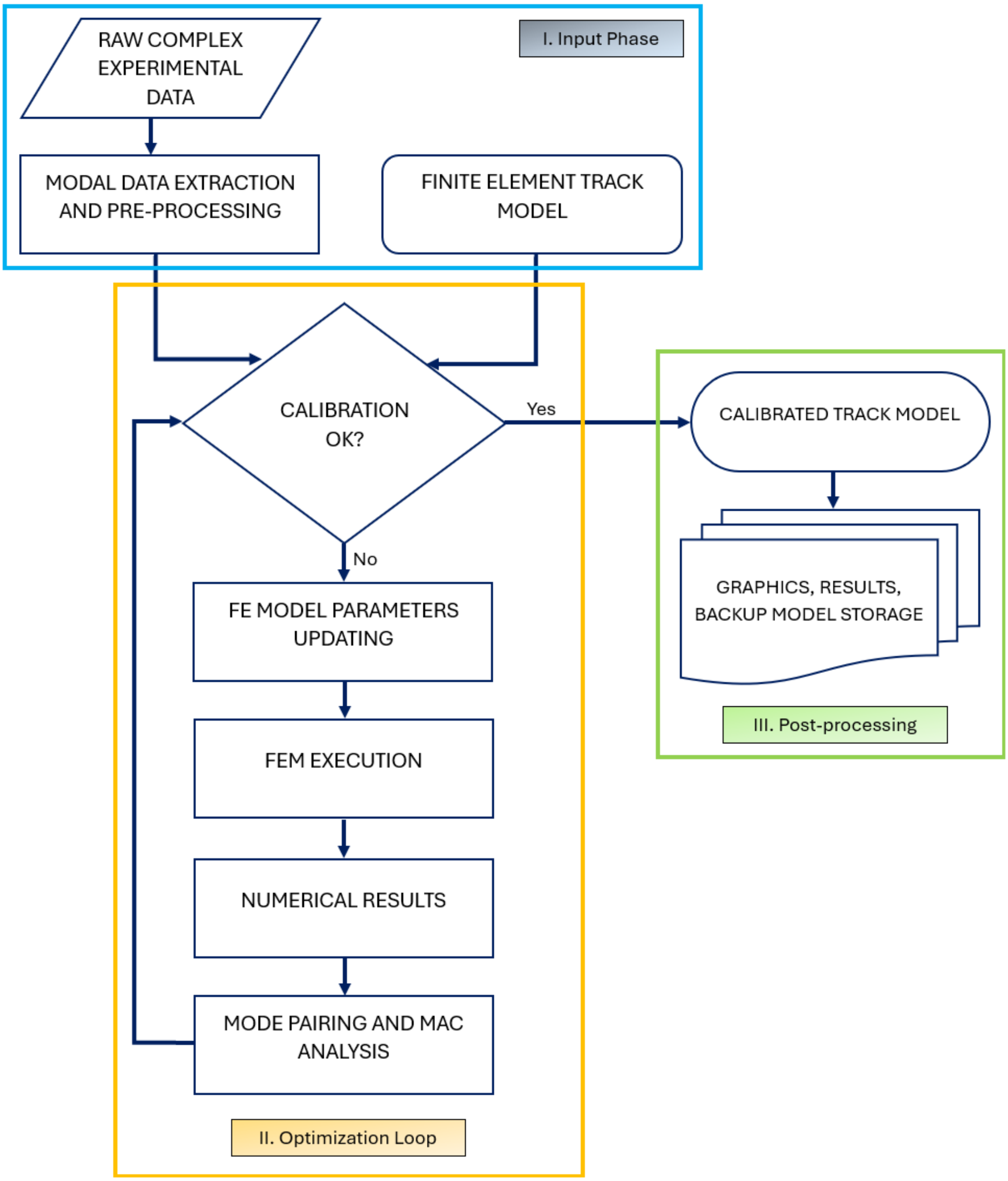


Figure 5. FEMU Framework: Workflow and Dependency Diagram

5.2. Optimization Methodology

The success of the calibration depends on the definition of the objective function. The system minimizes a scalar discrepancy based on mode shape and frequency and calibrates several dimensionless factors that scale physical track stiffnesses.
The objective MAC-Abaqus function represents the bridge between MATLAB and the Abaqus calculation engine. During each evaluation, the function performs a sequence of operations: updating the external database, executing the solver, and recovering modal data.

The optimizer minimizes a cost function:

$$J = -ScoreTotal \tag{1}$$

where the total score is a weighted combination of shape score (MAC), a statistical metric evaluating the linearity between numerical and experimental mode shape vectors and frequency score, that penalizes natural frequency deviations, ensuring the model reflects the actual mass and global stiffness distribution:

$$J = \omega_{Diag} \cdot E_{diag} + \omega_{Off} \cdot E_{off} + \omega_{Freq} \cdot E_{freq} \tag{2}$$

Where E_{diag} penalizes lack of correlation in the MAC matrix diagonal, E_{off} penalizes high values out of the diagonal and E_{freq} penalizes frequency deviations using a reference error (typically 5%).

To mitigate mode switching (where changes in stiffness alter frequency order), the system employs a greedy pairing. It sequentially selects pairs with the highest combined score (MAC + frequency proximity), providing more robustness against numerical spurious modes than global assignment methods. This approach sequentially selects pairs with the highest combined score of MAC and frequency proximity, proving more robust than global assignment methods when spurious or incomplete numerical modes are present.

5.3. Experimental and Numerical Data Processing

In the input phase, after loading the experimental raw data, a mode 'realification' process is carried out by using a script that transforms complex experimental modes into equivalent real modes using an optimized phase rotation (θ), allowing direct comparison with undamped numerical results. 'Realification' consists of rotating the complex vector in the complex plane by an angle θ such that the energy is concentrated maximally in the real part [11].

The first step of the optimization loop is the execution of Abaqus, divided into three stages:

- Generation of an .inp file for the solver.
- Eigenvalue analysis.
- Extraction of nodal displacements U_2, U_3 .

Communication is managed through system calls, updating the file database by injecting the new stiffness values proposed by 'fmincon'.

5.4. Post-Optimization Validation and Diagnosis

The frequencies chosen to test the optimisation algorithm were selected by identifying the stabilisation poles in the DEWESoft signal acquisition software used in the EMA and analysing the resulting experimental AUTOMAC. Only four poles were selected to reduce computation time; these were the first mode found around 250 Hz and the modes close to the resonance frequency band for the track-sleeper system (around 700–800 Hz), the first-order bending resonance frequency of the sleeper (around 1150 Hz) and the fourth-order bending mode of the sleeper (around 1320 Hz), according to data from different research studies [5], [6], [7], [9].

The system includes advanced tools to ensure model fidelity, as 3D MAC visualization, which generates three-dimensional bar charts where a maximized dominant diagonal and minimized out of the diagonal values indicates successful calibration (Figure 6).

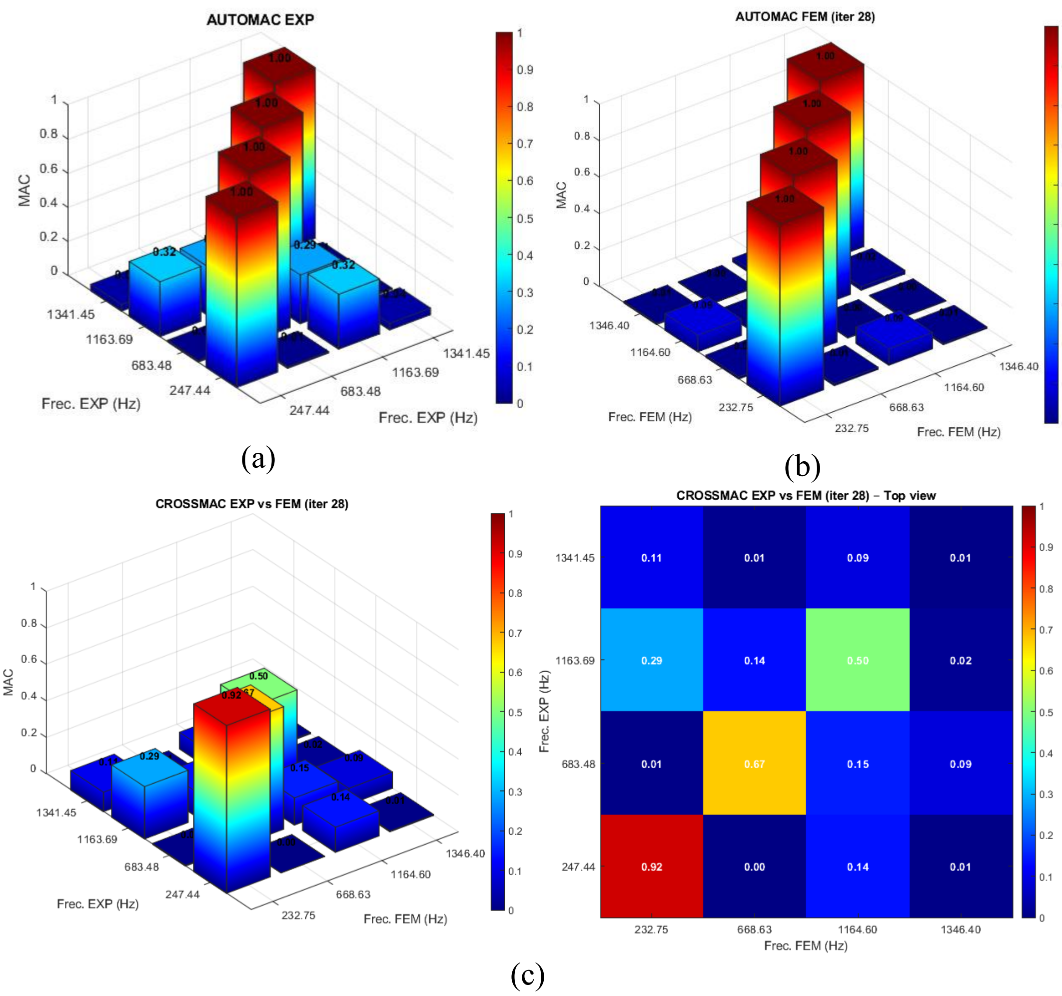


Figure 6. Experimental AUTOMAC (a), FEM AUTOMAC (b) and CROSSMAC Experimental vs FEM (c) for lateral impact and modal reply

6 Conclusions

This article presents an automated FEM parameters tuning process framework for a rail track. The FE model used is a beam simplification to avoid high computational time, as the purpose was to debug and test the optimization scripts and framework. In addition, and for the same reason, only the vertical and lateral stiffnesses of pad and ballast were optimized.

Overall, the model demonstrates sufficient fidelity for modal analysis purposes, especially in the low-frequency range. Future improvements may include incorporating ballast and pad damping values, refining support modelling to better capture local effects, as well as generating a FE solid model, as well as a new EMA incorporating additional measurement points.

The ultimate purpose of the FE model is to be uploaded into a Multi-Body System (MBS) software, incorporating the desired track defects on it as other researchers have done in the past for several purposes [12].

Different track scenarios can be created based on real track measurements and the FE section of the flexible track can be uploaded into it, so that several simulations can be carried out, with many different defects, and many dynamic results can be obtained.

This information will be used to develop defects detection algorithms that will be tuned with the numerical dynamic results of the MBS software and validated against real track defects.

In addition, once the model is validated, with enough synthetic data, Machine Learning models could be trained to further research in the field of predictive maintenance.

Acknowledgements

Funded by the European Union. Views and opinion expressed are however those of the author(s) only and do not necessarily reflect those of the European Union or Europe’s Rail Joint Undertaking. Neither the European Union nor the granting authority can be held responsible for them. The project FP3-IAM4Rail [Grant Agreement No. 101101966] is supported by the Europe's Rail Joint Undertaking and its members.

Co-funded by the European Union

References

[1] I. Bravo, U. Alvarado, J. Nieto, and P. Ciáurriz, “On-board accelerometers in railway track condition monitoring. A systematic review,” *Transp. Res. Interdiscip. Perspect.*, vol. 33, p. 101572, Sep. 2025, doi: 10.1016/J.TRIP.2025.101572.

[2] W. Heylen, S. Lammens, and P. Sas, “Modal Analysis Theory and Testing,” *Katholieke universiteit Leuven (KUL)*, 1998.

[3] D. J. Ewins, *Modal testing: theory, practice and application*, 2^{a} ed. Baldock (U.K.): Research Studies Press LTD., 2000.

[4] DEWESoft, “What is Modal Analysis: The Ultimate Guide,” *https://dewesoft.com/blog/what-is-modal-analysis*.

[5] Hao Sun *et al.*, “Looseness Detection of Rail Fasteners Using MEMS Sensor and Power Spectrum Entropy,” in *IEEE 1st International Conference on*

Micro/Nano Sensors for AI, Healthcare, and Robotics (NSENS), 2018. doi: 10.1109/NSENS.2018.8713630.

[6] Z. Zhan *et al.*, “Wireless Rail Fastener Looseness Detection Based on MEMS Accelerometer and Vibration Entropy,” *IEEE Sens. J.*, vol. 20, no. 6, pp. 3226–3234, Mar. 2020, doi: 10.1109/JSEN.2019.2955378.

[7] M. Oregui, Z. Li, and R. Dollevoet, “Identification of characteristic frequencies of damaged railway tracks using field hammer test measurements,” *Mech. Syst. Signal Process.*, vol. 54, pp. 224–242, Mar. 2015, doi: 10.1016/j.ymssp.2014.08.024.

[8] W. Heyle, S. Lammens, and P. Sas, *Modal analysis theory and testing*, 1ª ed. Leuven: Katholieke Universiteit Leuven, 1997.

[9] M. Oregui, Z. Li, and R. Dollevoet, “An investigation into the vertical dynamics of tracks with monoblock sleepers with a 3D finite-element model,” *Proc. Inst. Mech. Eng. F J. Rail Rapid Transit*, vol. 230, no. 3, pp. 891–908, Mar. 2016, doi: 10.1177/0954409715569558.

[10] A. Johansson and C. Andersson, “Out-of-round railway wheels—a study of wheel polygonalization through simulation of three-dimensional wheel–rail interaction and wear,” *Vehicle System Dynamics*, vol. 43, no. 8, pp. 539–559, Aug. 2005, doi: 10.1080/00423110500184649.

[11] N. Niedbal, “ANALYTICAL DETERMINATION OF REAL NORMAL MODES FROM MEASURED COMPLEX RESPONSES.,” in *Collection of Technical Papers - AIAA/ASME/ASCE/AHS/ASC Structures, Structural Dynamics and Materials Conference*, AIAA (CP845), USA AIAA 84-0995, 1984, pp. 292–295. doi: 10.2514/6.1984-995.

[12] Y. Bezin, S. D. Iwnicki, M. Cavalletti, E. De Vries, F. Shahzad, and G. Evans, “An investigation of sleeper voids using a flexible track model integrated with railway multi-body dynamics,” *Proc. Inst. Mech. Eng. F J. Rail Rapid Transit*, vol. 223, no. 6, pp. 597–607, Nov. 2009, doi: 10.1243/09544097JRRT276.