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Calibration and validation of the numerical model of a freight wagon based on dynamic tests under operating conditions

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Abstract. This work presents an efficient methodology for the calibration and validation of a freight wagon numerical model by an iterative methodology based on experimental modal parameters identified in a dynamic test under real operating conditions. The dynamic tests involved the use of a minimalist on-board monitoring system, which did not cause any interference in the vehicle's operational logistics. The identification of the vehicle's dynamic properties was made through the application of operational modal analysis techniques to the data collected during the vehicle circulation. A 3D numerical model of the freight wagon was developed and calibrated using an iterative methodology through a genetic algorithm and based on the identified modal parameters. The applied methodology proved to be effective and robust in estimating three numerical parameters, besides a significant upgrade in the natural frequencies compared to the model before calibration. Finally, the dynamic response of the model was validated by means of a direct comparison between numerically simulated results, based on a vehicle-track interaction analysis, and experimentally collected time history responses. Comparisons revealed an excellent agreement between the experimental and numerical time series after calibration, especially for the frequency range covered by the identified modes.

Keywords: Freight wagon; dynamic tests; model updating; vehicle-track interaction; experimental validation.

1 Introduction

Over the years, rail freight traffic has undergone many changes towards increasing axle loads and train speeds. All these changes end up increasing the dynamic loads imposed by these vehicles on the structures, causing safety concerns to the engineers [1–3]. In order to know the magnitude of this dynamic loads, complex vehicle-structure interaction simulations must be performed. However, the accuracy of these simulations is highly conditioned to the fidelity of the numerical models of both the structure and the vehicle. Recent studies [4,5] have demonstrated that the quality of these models can be significantly improved by adequate calibration and validation procedures based on experimental modal data.

In this context, this work presents a complete methodology for the calibration and validation of the numerical model of a freight wagon. The model calibration is performed through an iterative methodology based on a genetic algorithm and experimental modal parameters identified through data collected during the vehicle operation using a very minimalistic experimental setup. After the calibration, the model was validated through the direct comparison between dynamic responses measured in the wagon during its operation and simulated responses based on a vehicle-structure interaction methodology.

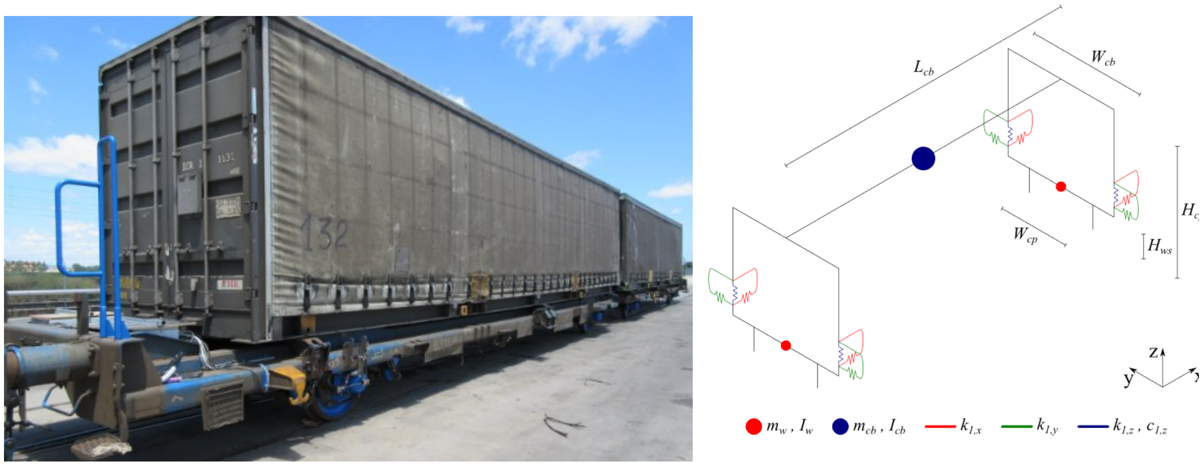
2 Freight railway wagon

2.1 Description

The Laagrss type freight wagon (Fig. 1a) is a platform vehicle, which is specifically designed for the transport of containers. This particular wagon operates on the Beira Alta line in the Portuguese railway network transporting rolls of paper. It has a total length of 14.8 m and it is supported by two axles spaced by 10 m. The axles are connected to the carbody by four UIC [6] parabolic leaf springs with progressive stiffness attached by UIC [6] double links. The wagon has a tare weight of 27.1 ton and can carry up to 24.9 ton of cargo.

2.2 Numerical model

Fig.1b presents an overview of the developed numerical model, which is based on rigidi body dynamics concepts. This simplified model was developed in ANSYS®[7] software making use of rigid beam elements representing the geometry of the carbody and wheelset, while all their inertial properties were modelled by concentrated mass elements positioned at their corresponding gravity center.



a) b)

Figure 1. The Laagrss type freight wagon: a) Wagon overview, b) numeric model

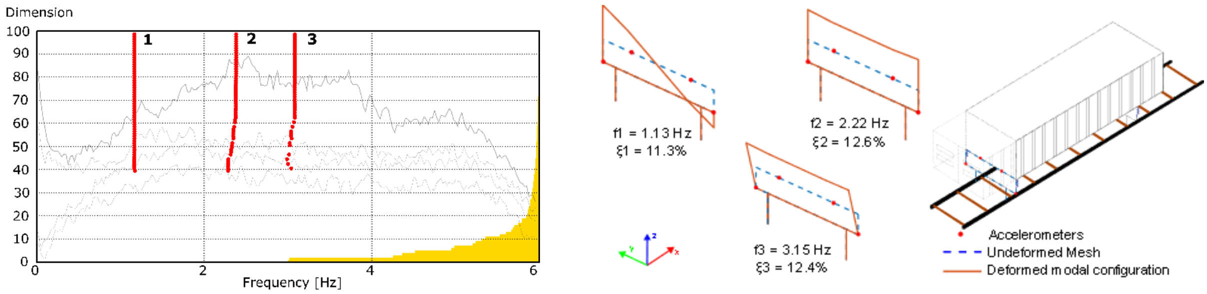
3 Dynamic tests

The dynamic tests involved the installation of 10 accelerometers and 5 LVDT's, whose positions were carefully chosen in order to not interfere with the disposition of transported goods, neither with the loading and unloading operation of the wagon. The accelerometers were arranged in order to have two triaxial acceleration measurement points on the carbody and also two triaxial acceleration measurement points on the wheelset (Fig. 2b). The LVDT's were installed between the axle box and the carbody to measure relative displacement between these two components.

The data was collected during the wagon's regular operation on the Beira Alta line and the modal identification was performed by the application of the Stochastic Subspace Identification (SSI) technique, implemented on the commercial software ARTeMIS®[8], to the measured acceleration time histories. Fig. 2a presents the stabilization diagrams corresponding to the three rigid body modes of the carbody which were identified.

Fig.2b presents the natural frequencies, damping coefficients and mode shapes of the three identified modes.

The first mode is associated with a rotation along the x axis (rolling) coupled with some lateral translation of the carbody's gravity center, the second is a bouncing movement of the carbody and the third is a pure rotation along the y axis (pitching). Due to the minimalist experimental setup, which was restricted to the front of the wagon, the second and third modes could only be distinguished by the longitudinal movements associated with the third mode. These movements are present since a rotation along the y axis is associated with both vertical and longitudinal displacements. Additionally, the values found for the damping coefficients are typical for the leaf spring suspensions equipping this wagon [9].



a) b)

Figure 2. Identified modes: a) SSI stabilization diagram, b) experimental modal configurations.

4 Calibration

Fig.3 presents the implementation strategy, based on the software MATLAB® [10] and ANSYS® [7], of an iterative calibration technique which relies on an Genetic algorithm.

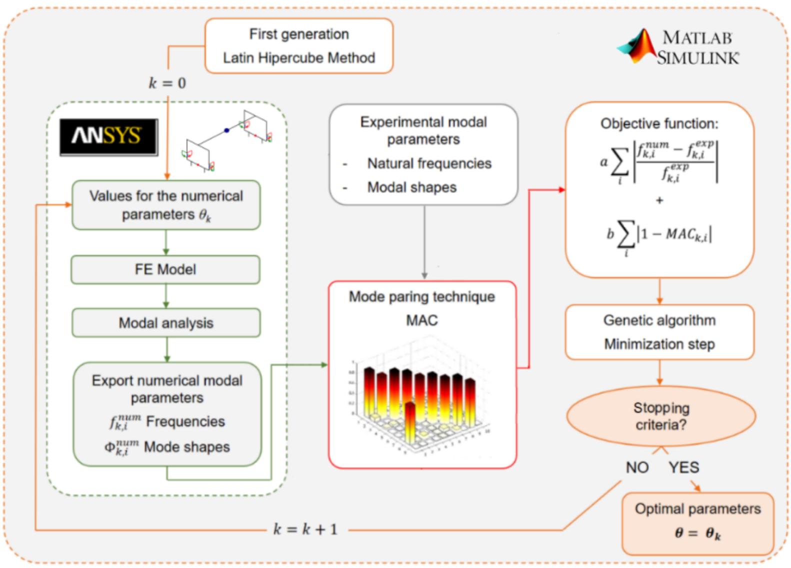


Figure 3. Calibration implementation flowchart.

First, in ANSYS® [7], a numerical model is generated through a set of numerical parameters which, at the initialization of the algorithm, are randomly generated by the Latin Hypercube sampling technique. These specific parameters (carbody mass (m_{cb}), carbody inertia along y ($I_{cb,y}$) and vertical stiffness of the suspensions ($k_{1,z}$)) were previously identified as being correlated to at least one of the modal responses being used as the baseline for the calibration. Only those sufficiently correlated parameters can be estimated accurately during the model calibration. Then, a numerical modal analysis is performed and the natural frequencies and mode shapes exported to MATLAB® [10] through text files.

In MATLAB® [10], firstly, the numerical modal responses are paired with their experimental correspondent based on the highest Modal Assurance Criterion (MAC) value. Then, a Genetic Algorithm is used to minimize an objective function composed by the sum of the natural frequencies and MAC residuals. Finally, based on this minimization step, a new set of numerical parameters is generated and the processes is repeated until a prescribed maximum number of generations is reached. The set of numerical parameters that led to the lowest value for the objective function is taken as the optimal one.

Due to the stochastic nature of many operators in genetic algorithms, the stability of the optimization process was attested by conducting four (GA1-GA4) completely independent optimization runs. Fig.4 presents the values and their ratio with respect to the prescribed bounds, obtained for the numerical parameters in the four optimization runs. The stability of the processes is confirmed by the small fluctuations of the values which were restricted to no more than 7 %.

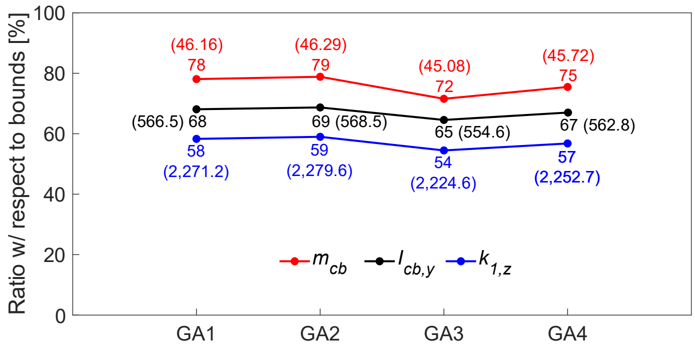


Figure 4. Ratio with respect to bounds for the optimization cases GA1 to GA4

Fig.5a presents the relative error between numerical and experimental natural frequencies before end after optimization. The mean error reduced from 8.71 % to only 0.03 %, which represents almost a perfect agreement for the natural frequencies. Fig.5b compares the MAC values before and after calibration, which were already high and remained almost unchanged after calibration.

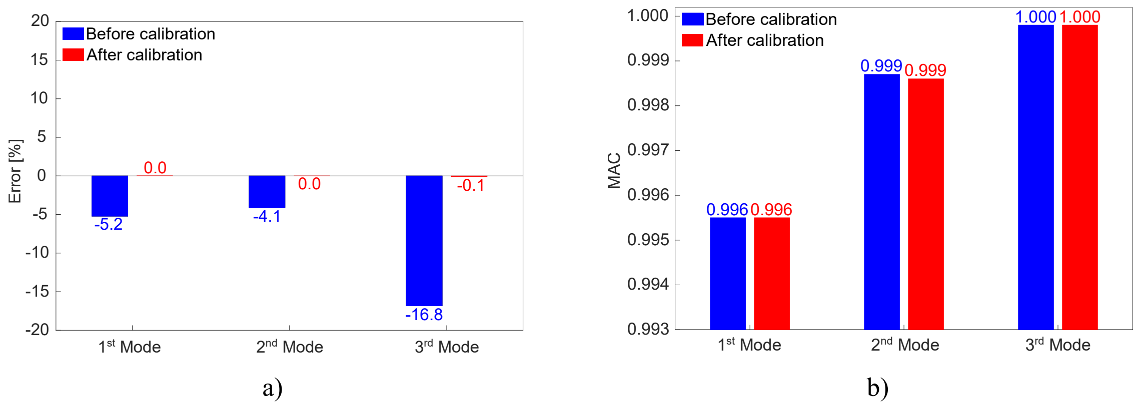


Figure 5. Errors between experimental and numerical modal responses, before and after calibration: a) Natural frequencies; b) MAC

5 Validation

The validation of the dynamic response of the wagon model was performed by the direct comparison between measured and simulated time histories of accelerations on the carbody and relative displacement between carbody and wheelset. The simulations were performed by means of the "VSI – Vehicle Structure Interaction Analysis" numerical tool developed by Neves et al. [11] and later improved by Montenegro et al. [3] to account for lateral interaction on the wheel-rail contact model. Fig.6 presents a framework of this numerical tool, which is implemented in MATLAB® [10], and imports the structural matrices, from both vehicle and structure, directly from the numerical models developed in ANSYS® [7].

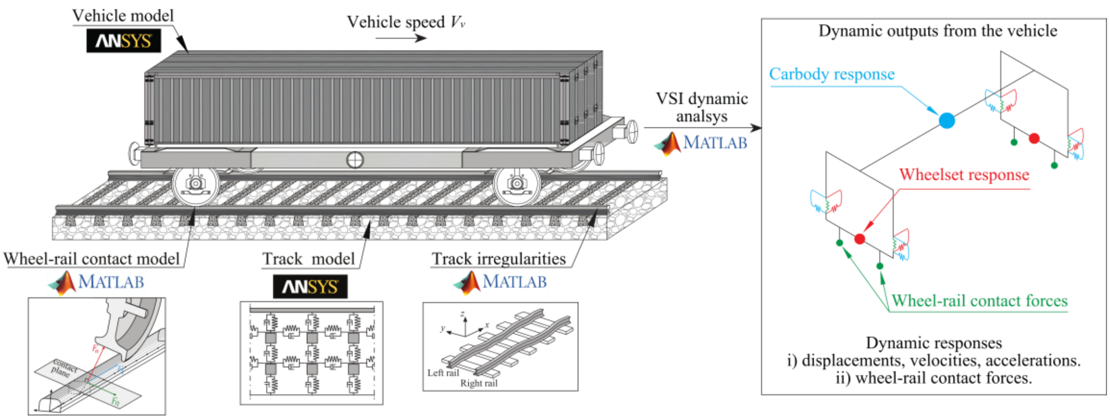


Figure 6. Framework of the VSI numerical tool.

The track section chosen for validation was a straight alignment with 400 m length in which the track irregularities were measured two weeks before the experimental campaign and were considered in the simulations. In this section, the wagon maintained an approximate constant speed of 84.5 km/h and the experimental data was collected by means of the same experimental setup presented before.

Fig.7 presents the measured and simulated, both prior and after calibration, acceleration responses at the carbody in time and frequency domains. The results are band filtered between 0.4 and 10 Hz by a 4th order Butterworth filter, following EN14363 [12] recommendations. As can be clearly seen, the time history reposes improved a lot after calibration and are much more similar to the measured ones not only in terms of magnitude, but also in terms of the shape of the signal. In frequency domain the improvement is also notorious, especially under 3.5 Hz were most of the energy is concentrated.

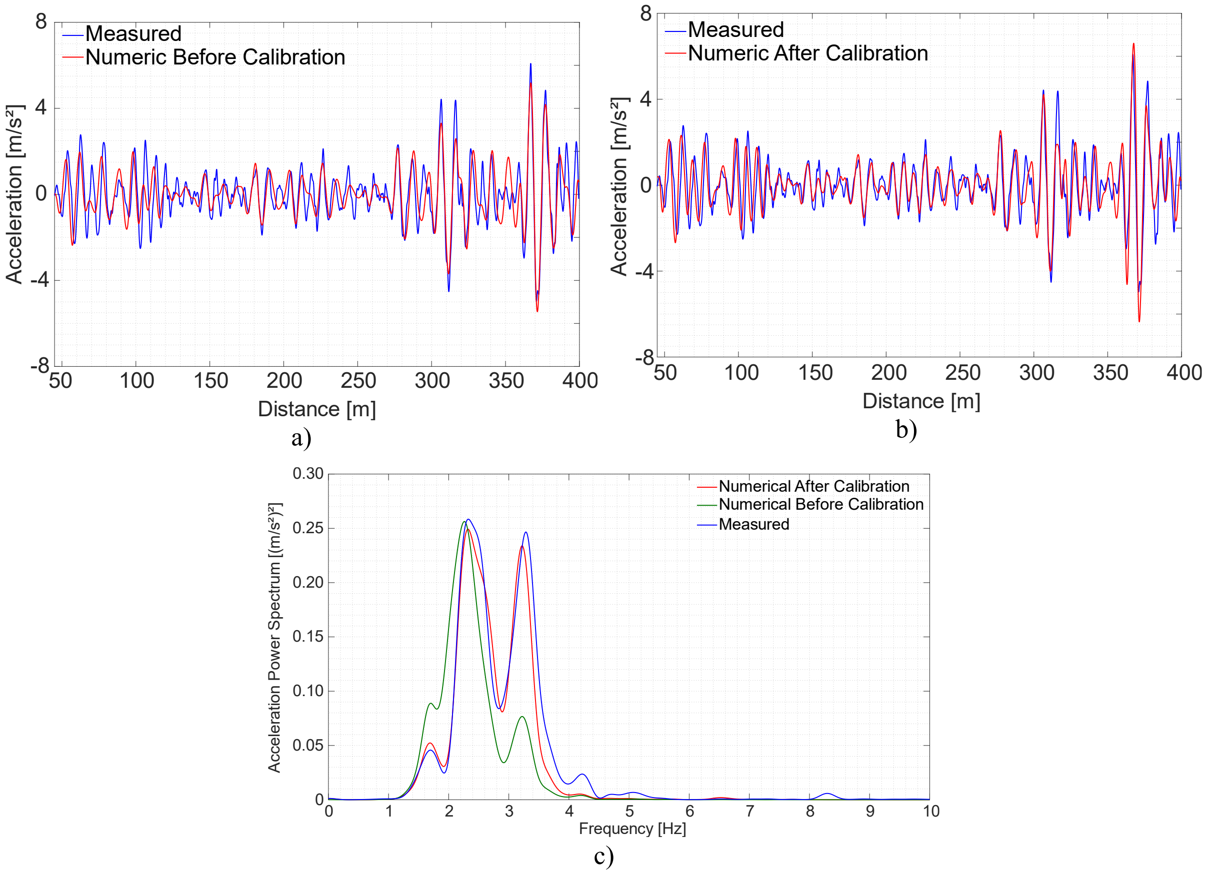
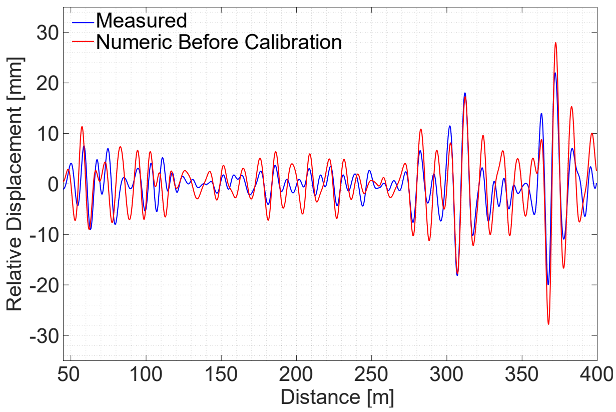
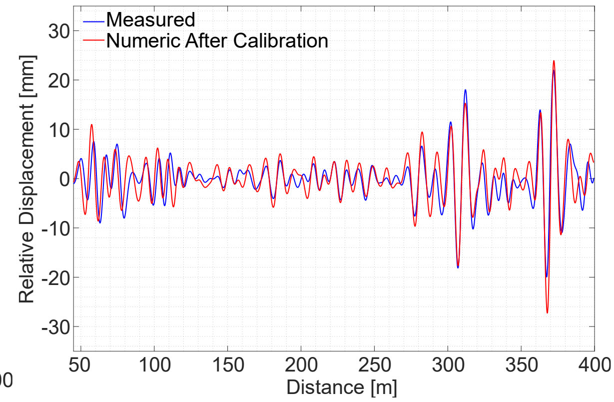


Figure 7. Experimental and numerical carbody vertical accelerations: a) Before calibration; b) after calibration; c) power spectrum.

Fig.8 presents another comparison between simulated and measured time history responses, but in terms relative vertical displacement between the vehicle's carbody and wheelset. The results are band filtered between 0.1 and 4 Hz by a 4th order Butterworth filter, following EN14363 [12] recommendations. Just like for accelerations, major improvements were achieved in the time history responses, with respect to both magnitude and shape of the signal. In the frequency domain significant improvements are also notable, especially because in the model before calibration the response between 1.4 and 2.5 Hz was significantly overestimated.



a)



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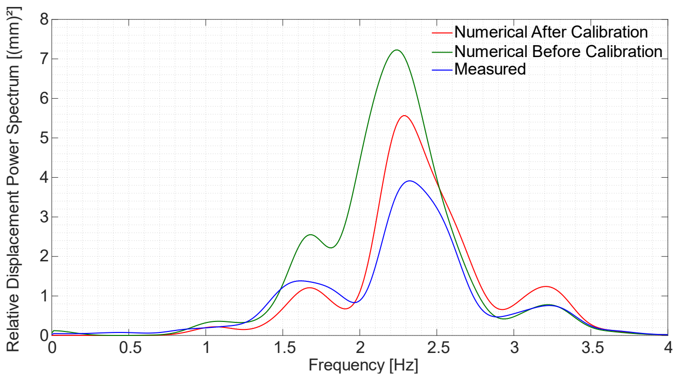

c)

Figure 8. Experimental and numerical Carbody-Wheelset vertical relative displacement: a) Before calibration; b) after calibration; c) power spectrum

6 Conclusions

This paper presented the calibration and validation the numerical model of a freight wagon based on experimental data. The acceleration time histories, collected during the vehicles operation by means of a very minimalistic experimental setup, allowed the identification of three rigid body modes of the carbody. These modes were used for calibrating the model by means of an iterative methodology based on a genetic algorithm. The calibration process was very efficient in estimating three numerical parameters of the model and significantly reducing (from 8.71 % to only 0.03 % in average) the differences between numerical and experimental natural frequencies.

Finally, the model was validated by the direct comparison between measured and simulated time histories and spectral responses of the vertical acceleration at the carbody and the vertical relative displacement between carbody and wheelset. The comparisons reveled great improvements in both responses both in time and frequency domains, especially for the frequency range covered by the three modes used during the model calibration.

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