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FINITE ELEMENT MODELS FOR THE ANALYSIS OF THE EFFECTIVENESS OF THE VERTICAL VIBRATION REDUCTION METHOD OF THE RAILWAY VEHICLE CARBODY WITH ANTI-BENDING SYSTEM - APPLICATIONS FOR AN EXPERIMENTAL SCALE MODEL

Mădălina DUMITRIU, Ștefan SOROHAN, Ioana Izabela APOSTOL

Abstract: *In the paper are presented two finite element models of an experimental scale model of the carbody (ESMC) of the railway vehicle, used to analyse the effectiveness of a new method to reduce the vertical bending vibration of the carbody. The method is based on the use of an anti-bending system composed of two bars, mounted on the longerons of the carbody underframe. The effectiveness of the method is assessed based on two criteria - the increase of the frequency and the reduction of the amplitude of the first vertical bending mode, using the results of modal analysis and frequency analysis on the dynamic response of the ESMC with/without anti-bending system.*

Key words: *railway vehicle; bending vibration; anti-bending system; finite element models; experimental scale model.*

1. INTRODUCTION

During travel, the railway vehicle is permanently subjected to the action of vibrations which, under certain conditions, may have unfavourable effects on the safety, ride comfort, ride quality, or may affect the integrity of the running gear or the track. In the current context, in which the general trend in railway transport is to increase speed, the problem of maintaining vibrations at a level required by ensuring the dynamic performance of the railway vehicle represents an important problem.

The railway vehicle is characterized by a vibration regime with specific features. The vibrations are of low frequency and appear in the vertical, lateral, and longitudinal direction as independent or coupled vibrations. Vibration modes are both simple modes, respectively rigid modes, and complex modes, namely structural modes [1].

The increase of the speed implies the adoption of an important criterion in the design stage of the railway vehicle - the reduction of weight. Ensuring this requirement also has other advantages: reduced energy consumption, reduced vibrations transmitted through the soil and savings in production costs.

In the case of the carbody, the reduction in weight is achieved by significantly reducing the structural rigidity. This allows for easy excitation of structural vibrations, especially in the long carbodies of high-speed vehicles. Several theoretical and experimental studies have demonstrated that the vibration level of the carbodies of high-speed vehicles is strongly influenced by the structural vibration modes [2, 3]. Ride comfort is affected by the first structural vibration modes of the carbody whose frequency falls within the frequency range in which the vibration comfort is affected. Ride comfort deteriorates more in the middle of the carbody. Consequently, at high speeds, the critical point of the vibration level of the car body, which is usually at its extremities (above to the bogies), is repositioned in the middle of the carbody [4]. Relevant from this point of view is the first vertical bending mode of the carbody, whose frequency is naturally located in the range of 6 ... 12 Hz [5].

In this context, the topic of structural vibrations is a research topic of interest, which is why it has been approached in different directions. One of these aims to identify the way in which structural vibrations influence the dynamic performances of the railway vehicle [6, 7]. Another direction aims to reduce or control structural vibrations to improve ride comfort, this can be found in numerous research, approached from the perspective of vibration isolation or vibration damping, based on both passive and active concepts [8, 9]. As a rule, in a first stage, to verify the new theoretical concepts introduced or the efficacy of the proposed methods, results obtained with software applications for simulation of the vibration behaviour of the railway vehicle are used. These software applications are based on simplified theoretical models of the railway vehicle. The next stage involves the validation of the theoretical results based on the experimental results. As a rule, this stage includes several phases of experimental research: laboratory testing on scaled experimental systems [10]; laboratory testing with special full-scale test vehicles [11]; on-line testing with full-scale test vehicles [12].

In this paper we will refer to the first phase of experimental research – laboratory testing on experimental systems, from the perspective of designing a ESMC of the railway vehicle. It will be used to analyse the efficacy of a new method of reducing the vertical bending vibrations of the railway vehicle carbody. The method is based on an anti-bending system consisting of two bars, mounted on the longerons of the carbody underframe, which have the role of limiting the rotation of the cross-sections of the carbody caused by the vertical bending vibration. The efficacy of the method was verified based on the results of numerical simulations, following two criteria - reducing the amplitude of the first vertical bending mode of the carbody and increasing the frequency of this mode of vibration so that it is outside the range in which vibration comfort is affected [13]. Preliminary specifications for the design of the ESMC and anti-bending bar system were previously established using an analytical method [14].

To verify these specifications, to develop the final design of the ESMC with anti-bending system, it is necessary to use much more complex models with finite elements. Two finite element models of the ESMC are presented in the paper. These two models are used to verify the results of the analytical model and, implicitly, the effectiveness of the proposed method. Modal analysis and frequency analysis are used as analysis methods.

2. DESCRIPTION OF THE ESMC

The ESMC that will be used for the experimental testing of the effectiveness of the method of reducing vertical bending vibrations based on an anti-bending bars system is schematically represented in figure 1.

The vehicle carbody is reduced to an aluminum beam that rests on rubber elastic elements. Steel anti-bending bars are attached to either side of the ESMC. The ESMC is excited at one end with a harmonic force.

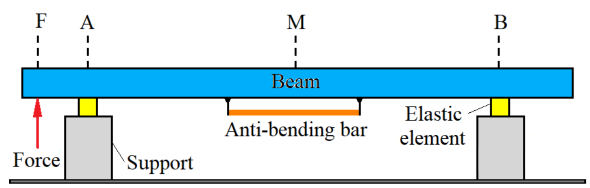

Fig.1. Scheme of the experimental carbody scale model

To develop the theoretical model of the ESMC without/with anti-bending system by the finite element method, the specifications contained in Table 1 are considered. They were established with an analytical method based on the Euler-Bernoulli beam model. In the dimensioning of the ESMC and the anti-bending bars, several conditions were involved: the advantageous adoption of the scaling factor of the carbody dimensions so that the scale model can be practically realized and ensuring stability on the elastic elements of the ESMC, the realization of the eigenfrequency of the vertical bending of the real carbody and decoupling the vibration modes of the ESMC - bounce and vertical bending [14].

The eigenfrequencies of bending vibration of the ESMC corresponding to the parameters in Table 1 are: 7.90 Hz – without anti-bending system and 20.42 Hz – with anti-bending system.

Table 1

Specifications for the development of the ESMC and anti-bending bars.

Specifications for the ESMC	
Material characteristics (aluminum)	
Modulus of longitudinal elasticity [N/m²]	$0.7 \cdot 10^{11}$
Density [kg/m³]	$2.7 \cdot 10^{3}$
The dimensions of the ESMC	
Length [m]	2.6
Width [m]	0.4
Height [m]	0.013
Specifications for the anti-bending bars	
Material characteristics (steel)	
Modulus of longitudinal elasticity [N/m²]	$2.1 \cdot 10^{11}$
Density [kg/m³]	$7.85 \cdot 10^{3}$
The dimensions of the anti-bending bars	
Length [mm]	900
Diameter [mm]	8
Other specifications	
Distance from the longitudinal axis of the anti-bending bar to the neutral axis of the ESMC [mm]	50
Longitudinal distance between elastic elements [m]	1.9
The lateral distance between the anti-bending bars [m]	0.32
Rigidity of elastic elements [kN/m]	324.252

3. THEORETICAL MODELS OF THE ESMC

3.1 Model with Shell, Beam and Spring finite elements - Model 1

Based on the data in Table 1, a preliminary finite element model of the ESMC was designed. For these data, the mass of the ESMC without anti-bending system and without springs is 36.504 kg. The structure of the model was discretized with Shell elements with eight nodes for the carbody and the flanges for fixing the anti-bending bars, Beam elements with three nodes for anti-bending bars and Spring elements for elastic supports (figure 2).

This model was developed in ANSYS APDL. For discretization, 1952 finite elements with 6035 nodes were used. To verify convergence, the finite element dimension was halved, resulting in a model with 7408 finite elements and 22583 nodes.

Table 2 shows the eigenfrequencies of the first six eigenmodes of the ESMC obtained with model 1, to which the two types of discretization were applied. It can be seen that the eigenfrequencies calculated with the finer discretized model are lower, but the differences are insignificant.

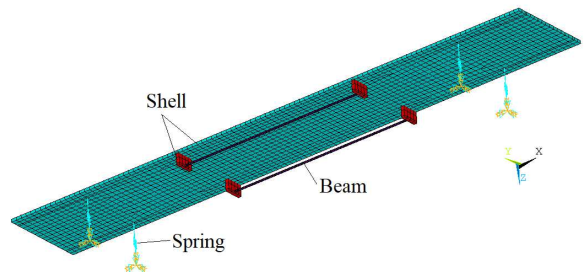

Fig.2. Model 1 for ESMC with anti-bending system

Table 2

Convergence verification of the preliminary finite element model of the ESMC.

Vibration mode	Model structure	
	6035 nodes; 1952 finite elements	22583 nodes; 7408 finite elements
	Eigenfrequency [Hz]	
I	12.284	12.276
II	24.452	26.450
III	30.945	30.492
IV	32.778	32.776
V	33.551	33.549
VI	45.265	45.243

3.2 Model with Solid type finite elements - Model 2

A second model was created in Ansys Workbench discretized with 98957 nodes and 50374 finite elements TETRA10. The CAD model is presented in figure 3, and the discretization of the model in figure 4.

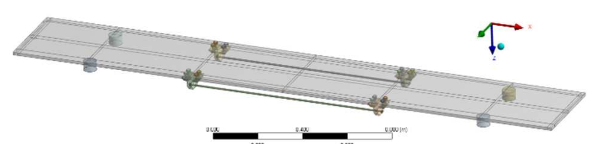

Fig.3. CAD model for the ESMC with anti-bending system

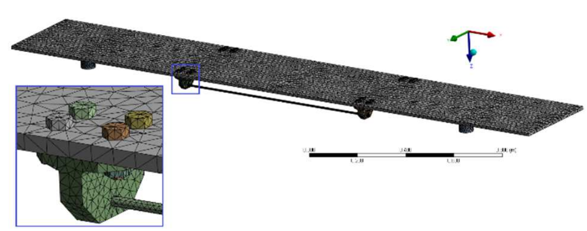

Fig.4. Model 2 of the ESMC with anti-bending system

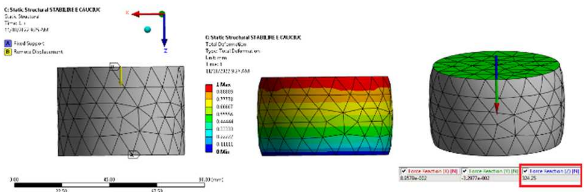

Fig.5. The model used for the elastic rubber element (diameter of 60 mm; height of 42 mm)

Figure 5 shows the model used for the elastic rubber elements. To achieve the stiffness of the elastic elements in Table 1 (324.25 kN/m), the transverse contraction coefficient 0.48 was adopted for rubber. The longitudinal modulus of elasticity is 3.6733 MPa.

4. DYNAMIC RESPONSE OF ESMC WITHOUT/WITH ANTI-BENDING BARS

4.1 Results obtained with Model 1

Figure 6 shows the main vibration modes of the ESMC without anti-bending system, and in figure 7 for the ESMC with anti-bending system.

It is found that the value of the eigenfrequency of the first vertical bending mode (7.89 Hz) is very close to the one obtained with the help of the analytical model [14].

The effect of the anti-bending system is observed in the modification of the eigenfrequencies of the vibration modes of the ESMC, as follows: the frequency of the first vertical (symmetric) bending mode increases to 12.28 Hz from 7.89 Hz; the frequency of the second vertical bending mode (antisymmetric) decreases to 26.45 Hz from 27.83 Hz; the frequency of the bounce vibration decreases to 32.78 Hz from 32.86; the frequency of pitch vibration decreases to 33.55 Hz from 34.20 Hz.

The results show the effectiveness of the anti-bending system from the point of view of increasing the eigenfrequency of the first vertical bending mode of the ESMC. The frequencies of the other modes of vibration do not change appreciably.

Comparing the above results with those obtained with the analytical model, it is found that it overestimates the eigenfrequency of the first vertical bending mode of the ESMC. This is explained by the simplifications on which the development of the analytical model is based.

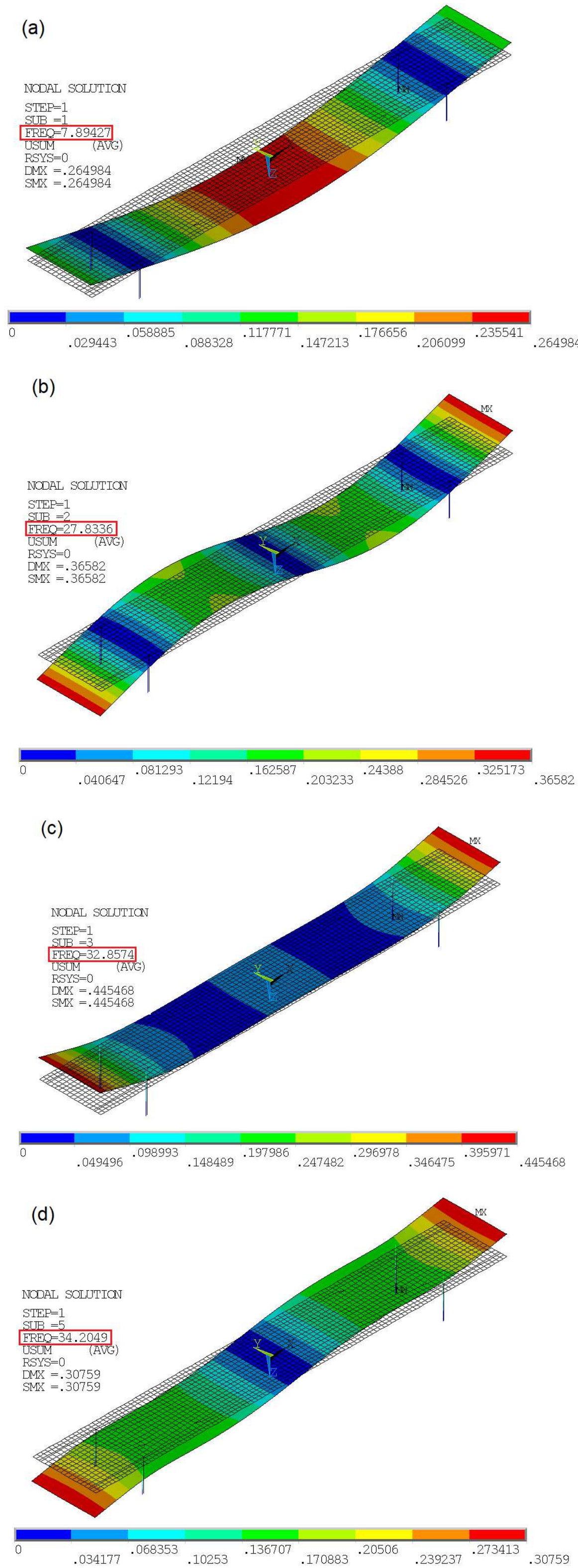

Fig.6. The main modes of vibration of the ESMC without anti-bending system: (a) first vertical bending mode (symmetric) (7.89 Hz); (b) second vertical bending mode (antisymmetric) (27.83 Hz); (c) bounce (32.86 Hz); (d) pitch (34.20 Hz)

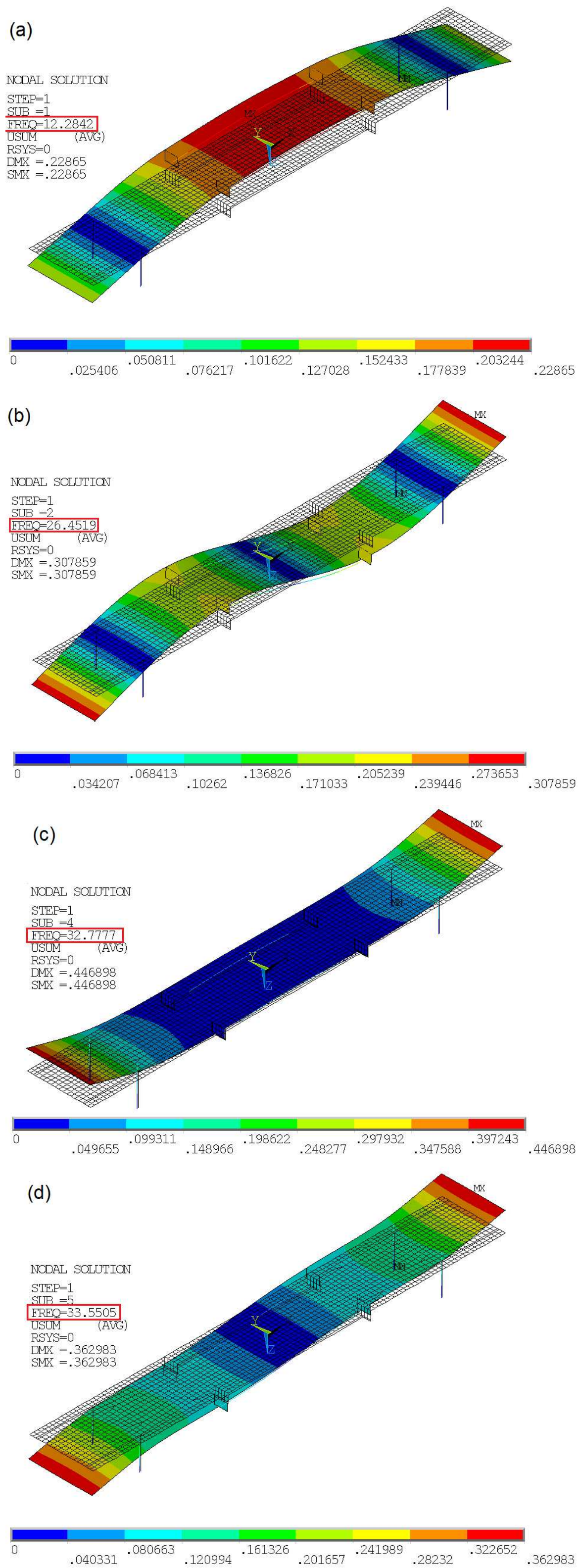

Fig.7. The main modes of vibration of the ESMC with anti-bending system: (a) first vertical bending mode (symmetric) (12.28 Hz); (b) second vertical bending mode (antisymmetric) (26.45 Hz); (c) bounce (32.78 Hz); (d) pitch (33.55 Hz)

Figure 8 shows the frequency response functions (FRF) of the acceleration of the ESMC without anti-bending system in the sections marked on figure 1: section F where the harmonic excitation force acts - diagram (a); sections A and B (above to the two elastic elements) - diagrams (b) and (d); section M (in the middle of the ESMC) - diagram (c). In all these sections, the maximum value of the acceleration of the ESMC is registered at the frequency of the first vertical bending mode.

Regarding the distribution of acceleration maxima, the highest value is in the middle of the ESMC, followed by the section where the excitation force is applied, and the lowest value is registered above to the elastic elements.

Figure 9 shows the FRF of the ESMC with anti-bending system calculated in the same sections as above. Comparing the maximum values of the acceleration with those obtained in the case of the ESMC without anti-bending system, it is observed that the allure of their distribution along the model is preserved. The value of these maxima is greatly diminished, which shows the effectiveness of the anti-bending system in reducing the bending vibrations of the ESMC.

4.2 Results obtained with Model 2

In this case, we focus on the first vertical bending mode of the ESMC, as it is important for the research topic.

Figure 10 shows the shape of the first vertical bending mode of the ESMC without anti-bending system, obtained under the conditions in which the effect given by the distributed mass of the elastic elements was considered. The mass of the ESMC (37.030 kg) is higher than for of Model 1, where the elastic elements are modelled with ideal elastic elements, i.e. without distributed mass. Although Model 2 has a larger mass, the frequency of the first vertical bending mode is 8.03 Hz, higher than that obtained for Model 1 (7.89 Hz). This aspect is explained by the fact that the dynamic stiffness of the elastic elements increases by considering their distributed mass.

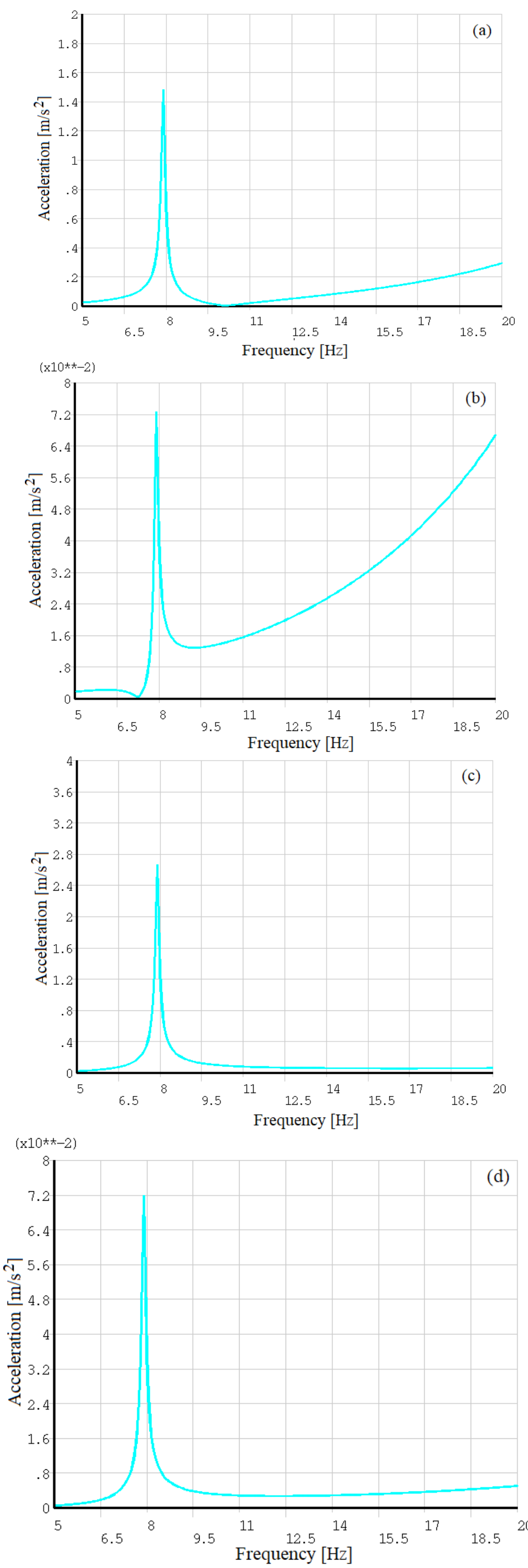

Fig.8. FRF of the ESMC without anti-bending system: (a) in section F; (b) in section A; (c) in section M; (d) in section B

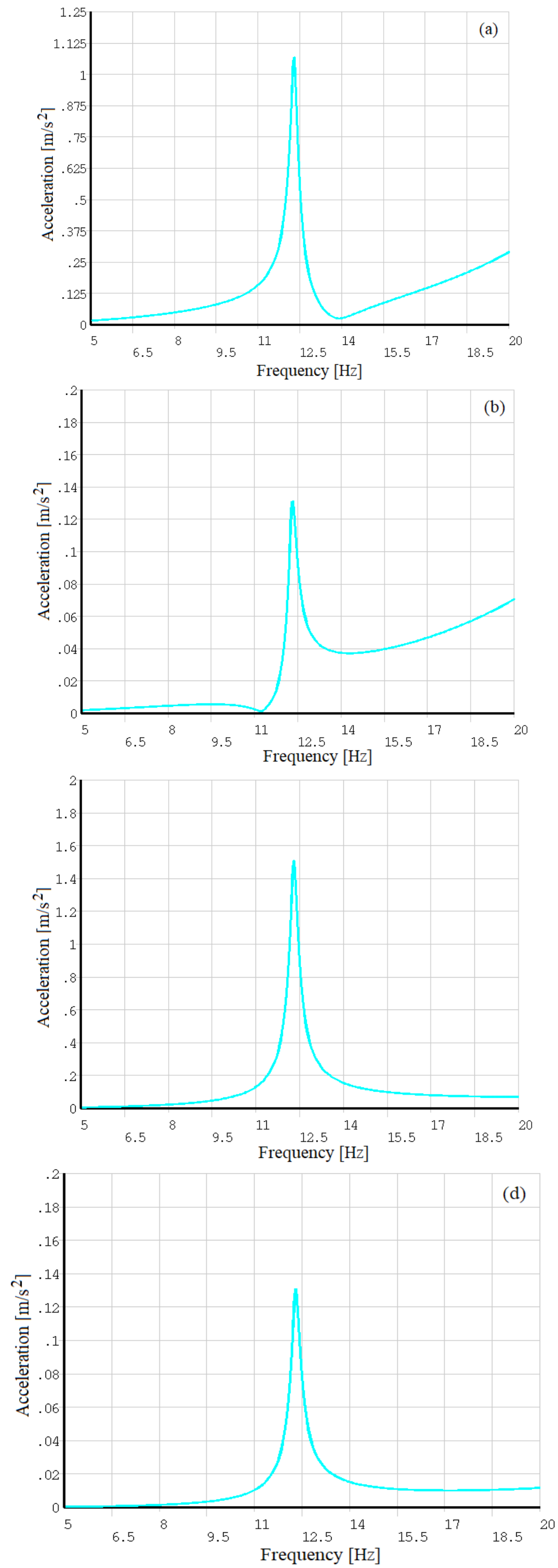

Fig.9. FRF of the ESMC with anti-bending system: (a) in section F; (b) in section A; (c) in section M; (d) in section B

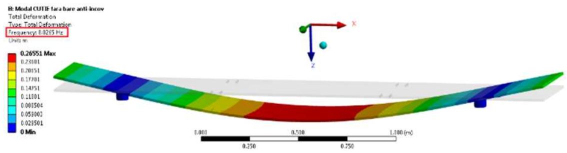

Fig.10. The first vertical bending mode of the ESMC without anti-bending system

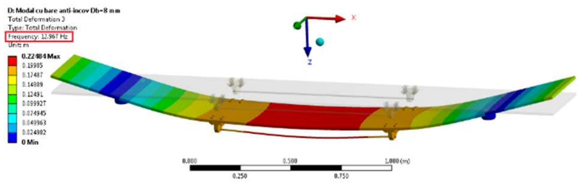

Fig.11. The first vertical bending mode of ESMC with anti-bending system

The influence of the anti-bending system on the first vertical bending mode of the ESMC is illustrated in figure 11. In this case, the mass of the model is 42.058 kg. The frequency of the first vertical bending mode of the ESMC is 12.97 Hz, higher than that calculated with Model 1 due to the superior refinement of the finite element structure in the case of Model 2.

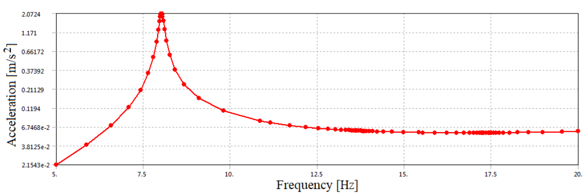

Fig.12. FRF of middle of the ESMC without anti-bending system

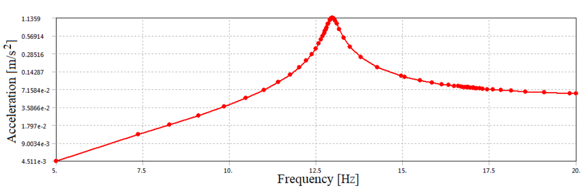

Fig.13. FRF of middle of the ESMC with anti-bending system

Figures 12 and 13 show the FRF at the middle of the ESMC without anti-bending system and with anti-bending system, respectively. Comparing the maximum values with those obtained with Model 1 (diagrams (c) from figures 8 and 9) it is observed that it overestimates the acceleration in the middle of the ESMC. The effect of the anti-bending system is highlighted by the significant reduction of the maximum acceleration value, which decreases from 2.07 m/s^2 to 1.14 m/s^2.

5. CONCLUSION

To verify the effectiveness of any new method for reducing the vibrations of the railway vehicle carbody, it is necessary to validate the theoretical results based on the experimental results. In a first phase of the experimental research, the validation of the theoretical results can be done based on the results obtained in the laboratory with scale experimental systems. In the present case it is an experimental demonstrative system for testing the effectiveness of a new method of reducing the vertical bending vibrations of the railway vehicle carbody based on a system of anti-bending bars. The main element of this system is the ESMC to which the anti-bending bars are attached.

The paper presents two finite element models of the ESMC without/with anti-bending system, developed based on the design specifications established with an analytical method. The first model is a simpler model with Shell, Beam and Spring finite elements, and the second model is a Solid finite element model. In the case of both models, two analysis methods are used - modal analysis and frequency analysis. The obtained results demonstrate the effectiveness of the method based on the use of anti-bending bars for both evaluation criteria:

- the eigenfrequency of vertical bending vibration increases from 7.89 Hz to 12.28 Hz in the case of the first model and from 8.03 Hz to 12.97 Hz for the second model;

- the acceleration at the eigenfrequency of the vertical bending decreases significantly in the middle of the ESMC; for example, in the case of the Model 2 there is a reduction in maximum acceleration of almost 45%.

The following conclusions can also be summarized: the anti-bending system have no effect on the eigenfrequency of the other vibration modes of the ESMC; the analytical model overestimates the value of the eigenfrequency of the first vertical bending mode of the ESMC.

6. ACKNOWLEDGEMENT

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Modele cu elemente finite pentru analiza eficacității metodei de reducere a vibrațiilor verticale a cutiei vehiculului feroviar cu bare anti-încovoiere - Aplicații pentru un model experimental la scară

În lucrare sunt prezentate două modele cu elemente finite ale unui model experimental la scară al cutiei vehiculului feroviar, utilizate pentru analiza eficacității unei noi metode de reducere a vibrațiilor verticale de încovoiere ale cutiei. Metoda presupune utilizarea unui sistem pasiv format din două bare anti-încovoiere, montate pe lonjeroanele laterale ale șasiului cutiei. Eficacitatea metodei se apreciază pe baza a două criterii - creșterea frecvenței și reducerea amplitudinii primului mod de încovoiere verticală a cutiei, utilizând rezultatele analizei modale și analizei în frecvență privind răspunsul dinamic al modelului experimental la scară al cutiei cu/fără bare anti-încovoiere.

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