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APPLYING THE EXPERIMENTAL MODAL ANALYSIS TO VALIDATE THE RAILWAY FRAME BOGIE DESIGN

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Abstract. A bogie subsystem is an essential component of a train car. During bogie lifetime several external forces, both exceptional and normal service loads, act on the bogie frame, coming from the wheel-rail contact points and from the interfaces with the carbody. This paper presents the validation of the bogie frame type CO-CO numerical model (finite element model). This type of bogies is used for asynchrony electric locomotive type LEMA 5100kW (under construction in Softronic Craiova-Romania). The finite element model validation is based, in according to EN 15227/2010, on experimental modal analyses [2].

***Key word**:* modal parameters, experimental modal analysis, finite element analysis.

1. INTRODUCTION

Bogie subsystems are complex equipments with a vital role in the railway vehicle operation. This bogie subsystem is designed to carrying the carbody, ensuring the traction and braking forces transmission and taking transported loads, as well as of vibratory isolation of the carbody [1].

During bogie lifetime (approximately 30 years), several external forces, both exceptional and normal service loads, act on the bogie frame, coming from the wheel-rail contact points and from the interfaces with the carbody.

These forces are generated from: double sprung masses, including payload; track irregularities; lateral accelerations caused by curve riding; longitudinal accelerations caused by traction and braking [1] as well as other typically exceptional events: exceptional payloads; buffer impacts; minor derailments.

Because the bogie has a very important role in road safety rail (all loads and mechanical stresses from the track are transmitted to the carbody through the bogie and act in the bogie frame), the European standardization body CEN give a special attention to design and assessment procedures of the new railway bogie frames.

The actual European standard EN 13749/2011 [3, 4], specifies the methods to be followed to achieve a satisfactory design of bogie frames including the design procedures and assessment methods. According to EN 13749/2011, the aim of the acceptance program is to show that the behavior of the bogie frame will give satisfactory service without the occurrence of defects such as catastrophic rupture, permanent deformations and fatigue cracks.

The acceptance program shall demonstrate that there is no adverse influence on the associated bogie components or subassemblies.

The procedure for acceptance of the mechanical strength shall be established on the basis of: calculations; static tests; fatigue tests; on-track tests.

For a new design of bogie frame destined for a new type of application, all four-validation stages shall be used. A reduced programmer could be accepted.

Manufacturers of railway material must line up to the normative requirements, so that at the present the realization of a railway bogie involves:

- Computer aided design, using dedicated programs when are established the bogie frame physical model, geometric and execution details;
- Structural analysis, using dedicated programs when it is validated physical model in terms of structural strength and are highlighted the high mechanical stressed areas, that need to be carefully monitored during the tests;
- Static and fatigue tests, with priority monitoring of the mechanical stress in areas indicated by structural analysis;
- Validate structural analysis by calibrating numerical model with experimental data;
- Include the numerical model of the bogie into the general numerical model of the railway vehicle;
- Validate the railway vehicle numerical model for the traffic (including the crashworthiness requirements).

The EN 13749/2011 normative highlights, but don't imposes, taking into account the effects of flexibility on the bogie frame dynamics. It is left to the manufacturer to consider the dynamics of bogie frame and their influence on the ride quality of railway vehicle as a whole [3, 4].

Given that a significant percentage of the weight of railway vehicle is concentrated in the bogie structure and its associated devices, a detailed bogie dynamic structural analysis is needed to assess the railway vehicle to crash, according to European standard EN 15227/2010 [1].

In terms of structural dynamics, the bogie frame can be characterized by a set of modal parameters: modal frequency, modal damping and mode shape. This set of modal parameters completely characterizes the dynamic properties of a structure and also are refereed as a modal model of the structure. When a structure is subjected to extern vibration state, from the frequency spectrum, the structure absorbs energy mainly of their eigenfrequency and in the structure vibration state will be found mainly vibrations corresponding to eigenmodes [5, 7].

The railway vehicle carbody receives vibrations and mechanical shocks from four major sources: engines, bogie, irregularities in the track, and train slack movement. Slight imbalances in reciprocating engines, rotating drive shafts, motors, and generators create vibrations which are transferred to the carbody. Bogie gears can create vibrations, particularly during dynamic braking. Track which is not perfectly level vertically and laterally creates vehicle body accelerations that results in vibration. Engine loading and vehicle speed cause the vibrations to vary in intensity. Shunting and slack run-ins and run-outs are common sources of mechanical shock. A proper locomotive design can significant reduce the vibration level on the structure and can increase the dynamic performance and vehicle ride quality [1].

From the above it follows that comprehensive process of design and assessment of railway bogies involves the dynamic structural analysis using specialized software. A good theoretical model permits the numerical simulation of all the tests in real operating conditions of the equipment. If it uses only numerical analysis cannot determine the equipment real response under dynamic conditions even if the theoretical model is known as geometric data and the material characteristics are known approximately and introduce significant errors in the results of numerical simulation. A good numerical simulation requires calibration of the numerical model by experimental tests. The validation program is therefore a combination of the experimental tests and numerical simulations [4, 5].

The combined analysis consists in calibration of numerical model based on experimental data obtained by measurements of the variables which characterize the system evolution in known condition tests. The system is excited in well-defined conditions and by determining the evolution of the system response for the know excitation evolution law it will be identified a minimal set of parameters which are the intrinsic characteristics of equipment, independent of external conditions.

Finite element analysis (FEA) is commonly used in the development of the most new machines, structures and products of all kinds. Once a finite element model was validated, it can be used for numerical simulations, calculating stresses and strains, and for investigating the effects of structural modifications on the vibration properties of a structure. Since both experimental modal analysis (EMA) and finite element analysis (FEA) yield a similar set of modes for a structure, modal parameters are used to compare experimental and analytical results and for calibration of FEA model by experimental EMA model [5].

The paper presents an application of combined analysis, experimental and numerical, on a bogie frame type CO-CO for asynchrony electric locomotive LEMA 5 100 kW, under production in SC Softronic Craiova – Romania. The numerical model, validated by experimental data, constitutes support for crash certification of LEMA 5100kW locomotive, according to EN 15227/2010 standard.

2. FE ANALYSIS OF LEMA 5 100 kW BOGIE FRAME

The main technical characteristics of LEMA 5100kW bogie:

Axle formula	CO – CO	Maximum speed	120 km/h (gearbox ratio 1:3,65) 160 km/h (gearbox ratio 1:2,66) 200 km/h (gearbox ratio 1:2,108)
Distance between the extreme axles of a bogie	4350 mm		
Wheel diameter	1250 mm (new state) 1210 mm (semi-used state)	Bogie total weight with traction motor	28449 kg
Axle load	21t±2% (120 km/h – with ballast) 20t±2% (160 km/h and 200 km/h – version – without ballast)	Bogie total weight without traction motor	19449 kg
Bogie maximum length	8384 mm	Bogie maximum width	3000 mm

The finite element analysis of bogie frame was performed to evaluate the mechanical stresses distribution on the structure to the loads applied during the homologation tests, according to EN 13749/2011 and the bogie frame response to a crash scenario, according to EN 15227/2010.

In the first step it was applied a static structural analysis in view to determine the stresses and strains, on the bogie frame structure during the static and fatigue homologation tests, covering the service and exceptional loads. In the second step it was made a modal analysis to determine the vibration characteristics, natural frequencies and mode shapes of the bogie structure according to EN 15227/2010.

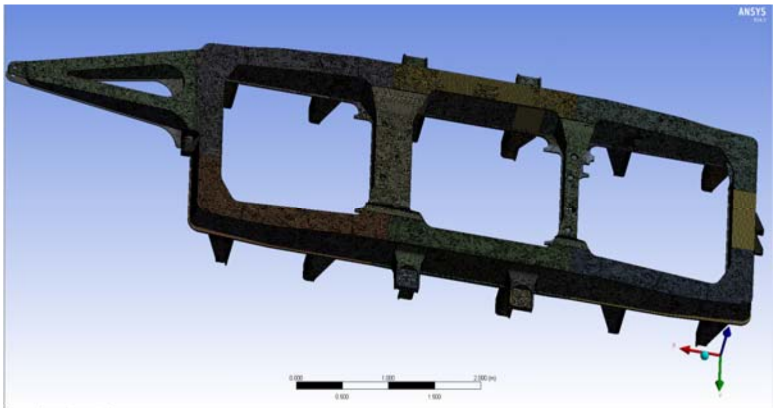

Fig. 1 – Finite element model of the LEMA 5 100 kW bogie frame.

The static and dynamic analysis was made with ANSYS using static structural module for statically analysis and modal module for dynamic analysis. Validation of analytical model for dynamic analysis was performed by comparison with experimental data obtained through experimental modal analysis. The bogie is modeled using shell and solid elements and the finite element model is shown in Fig. 1.

The bogie frame is meshed to have 2253246 nodes and 1129051 elements (rectangular shell and hexagonal solid elements). The material used in the bogie frame is the non-alloy structural welded steel type S355J2G3-STD01W03 (EN10025:1990). The material properties in according with EN10025:part 2: 2004, are [6]:

Tensile Yielding Strength	355 MPa	Poisson`s Ratio	0.3
Compressive Yield Strength	255 MPa	Bulk Modulus	170833MPa
Tensile Ultimate Strength	522MPa	Shear Modulus	78846MPa
Young`s Modulus	205000MPa		

Modal analysis was performed to determine the eigenmodes in the frequency range of 1 ÷ 100 Hz.

The modal analysis was performed considered that the bogie frame structure is an undamped system, any nonlinearities in material behavior are ignored. Stiffness was specified using isotropic and orthotropic elastic material models (for example, Young's modulus and Poisson's ratio). Mass derive from material density.

3. EMA ANALYSIS OF LEMA 5 100 kW BOGIE FRAME

3.1. THEORETICAL BACKGROUND

Any mechanical system can be modeled by a system consisting of n concentrated mass points, m^k joints by elastic elements with k^k stiffness and damping elements with c^k damping coefficient. For this damped system with n degrees of freedom, loaded by external excitation $\mathbf{F}(t)$, the motion equations are given by the following relation [7–10]:

$$[\mathbf{M}]\cdot\ddot{\mathbf{x}}(t)+[\mathbf{C}]\cdot\dot{\mathbf{x}}(t)+[\mathbf{K}]\cdot\mathbf{x}(t)=\mathbf{F}(t), \tag{1}$$

$[\mathbf{M}],[\mathbf{C}],[\mathbf{K}]$ – mass, damping and stiffness matrices; $\ddot{\mathbf{x}}(t),\dot{\mathbf{x}}(t),\mathbf{x}(t)$ – acceleration, velocity and displacement vectors and $\mathbf{F}(t)$ – force vector.

The system response to the external excitation is presented as a sum of n modal contributions due to each separated degree of freedom:

$$\mathbf{X}(\omega)=\sum_{k=1}^{N}\left(\frac{\boldsymbol{\psi}^k\cdot\left(\boldsymbol{\psi}^k\right)^T\cdot\mathbf{F}(\omega)}{a_k\cdot\left(-\mu_k+\mathrm{i}\cdot(\omega-v_k)\right)}+\frac{\boldsymbol{\psi}^{k^*}\cdot\left(\boldsymbol{\psi}^{k^*}\right)^T\cdot\mathbf{F}(\omega)}{a_k^*\cdot\left(-\mu_k+\mathrm{i}\cdot(\omega+v_k)\right)}\right), \tag{2}$$

$$\mathbf{X}(\omega)=\left[\mathbf{H}(\omega)\right]\cdot\mathbf{F}(\omega), \tag{3}$$

where: $\mathbf{X}(\omega)$ − Fourier Transform of displacement vector; $[\mathbf{H}(\omega)]$ − matrices of frequency response functions (FRF); $\boldsymbol{\psi}^k$ and $\boldsymbol{\psi}^{k^*}$ − the k order eigenvector and its complex conjugate; μ_k and v_k − the k order damping factor and damped natural frequency; a_k and a_k^* − normalization constants of the k order eigenvector and ω − frequency of the external excitation.

In the fast experimental modal analysis, the modal vectors are replaced by two modal constants U_{ij}^k and V_{ij}^k defined by:

$$\frac{\psi_i^k\cdot\psi_j^k}{a_k}=U_{ij}^k+\mathrm{i}\cdot V_{ij}^k \quad\text{and}\quad \frac{\psi_i^{k^*}\cdot\psi_j^{k^*}}{a_k^*}=U_{ij}^k-\mathrm{i}\cdot V_{ij}^k. \tag{4}$$

For a good approximation of the real system through the discrete system, it must have $n\to\infty$ bat this is not possible. In practical, the frequencies domain is limited to a reasonable width determined by the major resonances of the analyzed equipment and the frequency domain of the application goal. The contributions of inferior and superior modes are included in two correction factors known as “inferior modal admittance” $-\dfrac{1}{M'_{ij}\cdot\omega^2}$ (for inferior modes) and “residual flexibility”, S'_{ij} (for superior modes). The frequency response function elements will be written as:

$$H_{ij}(\omega)=-\frac{1}{M'_{ij}\cdot\omega^2}+\sum_{k=1}^{N}\left(\frac{U_{ij}^k+\mathrm{i}\cdot V_{ij}^k}{-\mu_k+\mathrm{i}\cdot(\omega\text{-}v_k)}+\frac{U_{ij}^k-\mathrm{i}\cdot V_{ij}^k}{-\mu_k+\mathrm{i}\cdot(\omega+v_k)}\right)+S'_{ij}. \tag{5}$$

So, an eigenmode can be defined by a set of $4n+2$ parameters: $\mu_k; \nu_k; U_{ij}^k; V_{ij}^k; -\dfrac{1}{M'_{ij}}; S'_{ij}$.

The global response of the system can be viewed as a sum of contributions of individual modes of vibration. The current approach in experimental modal identification involves using of numerical techniques to separate the contributions of individual modes in measurements such as frequency response functions. The concept involves estimating the individual single degree of freedom (SDOF) contributions to the multiple degree of freedom (MDOF) measurements. Equations (5) are nonlinear in terms of the unknown modal parameters.

In this approach the system dynamic is described [9, 12, 13] in terms of complex – valued modal frequencies λ_k, modal participation matrices $[\mathbf{L}]$, mode shapes matrices $[\boldsymbol{\psi}]$ and residue matrices $[\mathbf{A}^k]$. In these terms the equations (5) can be found in the following forms:

$$[\mathbf{H}(\omega)] = \sum_{k=1}^{N} \left(\frac{[\mathbf{A}^k]}{\mathrm{i}\omega - \lambda_k} + \frac{[\mathbf{A}^{k^*}]}{\mathrm{i}\omega - \lambda_k^*} \right), \tag{6}$$

or

$$[\mathbf{H}(\omega)] = [\boldsymbol{\psi}] \cdot \left[\frac{\mathbf{1}}{\mathbf{i}\boldsymbol{\omega} - \boldsymbol{\lambda}} \right] \cdot [\mathbf{L}]^T. \tag{7}$$

The combination of the modal participation vectors $\mathbf{L}^k$ and the modal vectors $\boldsymbol{\psi}^k$ for a given mode k gives the residue matrices $[\mathbf{A}^k]$ with elements given by relation: $A_{ij}^k = L_i^k \cdot \psi_j^k$.

Another important concept in modal parameter estimation originates in unique relationship between the time and frequency domains of the same system. Knowing that time impulse response function is inverse Fourier Transform of frequency response function, the frequency defined equations (5–7) can be transposed in the time domain by the following relation:

$$[\mathbf{h}(t)] = [\boldsymbol{\psi}] \cdot [\mathbf{e}^{\lambda t}] \cdot [\mathbf{L}]^T. \tag{8}$$

Based on relations (5) ÷ (8), were implemented several algorithms to identify modal parameters based on data measured in either time or frequency domain. The actually available methods help us to calculate the system modal parameters (modal damping, frequency, mode shape, etc.,) as well as modal participation vectors and residue vectors.

3.2. MODAL IDENTIFICATION METHODS

The most common methods in worldwide for modal identification are [8–10]:

• *Polyreference Frequency* (MDOF method): A low order method where the iterations (successive solutions) are based upon increasing spatial dimension. The more DOFs (FRFs) that has been measured the more iterations will be available for the Stability Diagram

• *Polyreference Time* (MDOF method): A high order method, where the iterations are based upon increasing order of the underlying polynomial equation

• *Rational Fraction Polynomial-Z* (Advanced MDOF method): A high order method, where the iterations are based upon increasing order of the underlying polynomial equation

• *Eigensystem Realisation* (Advanced MDOF method): A low order method where the iterations are based upon increasing spatial dimension. The more DOFs (FRFs) that has been measured the more iterations will be available for the Stability Diagram

• *Least Squares Global Partial Fraction* (SDOF method): Modes are identified from peaks or the valleys in the selected modal indicator function (or FRF) and modal parameters fitted over the specified number of frequency lines around each identified mode

• *Quadrature* (SDOF method): Extracts the imaginary part of the accelerance FRF's at the selected frequencies. This corresponds to the traditional "Quadrature Picking" technique where the peak of the imaginary part is extracted

• *Alias-Free Polyreference* (Advanced MDOF method): A frequency domain Laplace method that uses orthogonal polynomials and accounts for out-of-band poles.

For bogie frame presented in this paper, after experimental measurements, the validation of the numerical model parameters was performed using *Rational Fraction Polynomial-Z* method

4. EMA APPLICATION

4.1. TEST PROCEDURE

Modal identification tests were performed in SC Softronic Craiova, on the LEMA 5 100 kW bogie frame in the finite state, before the mounting stage. Bogie frame was placed in the normal operating position on elastic suspension consists of four helicoidally springs, each with elastic characteristic k = 519 N/mm.

Fig. 2 – EM analysis of LEMA 5 100 kW bogie frame.

Bogie frame geometry created under Ansys, for finite element analysis, was imported in Pulse Reflex [15] and decimated in order to achieve an acceptable geometry for testing and processing [14]. Structure excitation was made by impact method using an impact hammer of 25 kN. Impact was succesively applied in points 1 ÷ 10, according to Fig. 2, on vertical and horizontal – transverse directions. The acceleration response was measured in the same points and in the same directions. Tests were automatically conducted by LabShop, after each force impact visualizing the signals time evolution and frequency response functions.

4.2. DATA PROCESSING

Data processing was made under Pulse Reflex, Advanced Modal Analysis Package. In Fig. 3 is presented the Pulse Reflex-Advanced Modal Analysis Package screen [15].

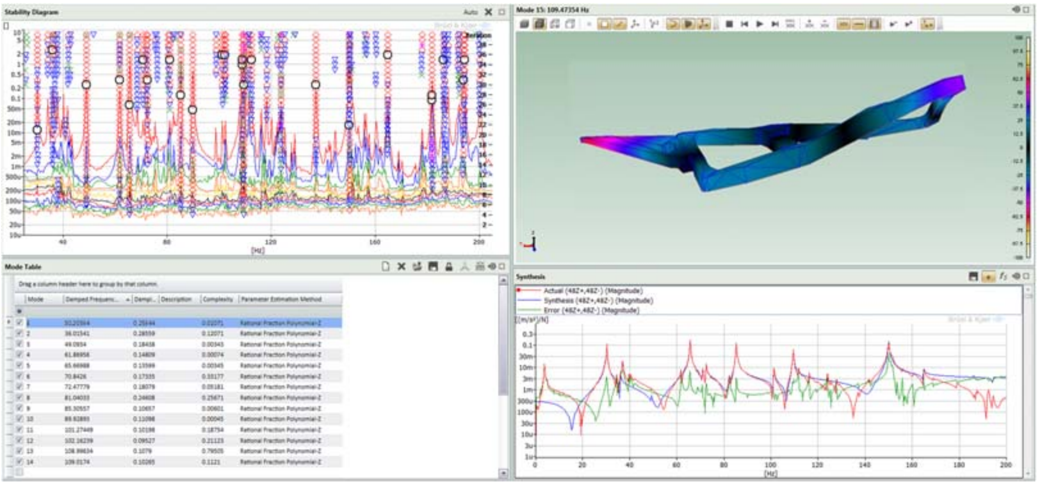

Fig. 3 – Stability diagram for EM analysis, the corresponding mode table, mode shape graphs (in figure is illustrated the first mode) and synthesis diagram.

5. COMPARATIVE FE-EM ANALYSIS

Table 1 presents the overall eigenfrequencyes obtained by both EM and FE analysis. Table shows the mode direction and frequency error between the damped eigenfrequency, determined by EMA and undamped eigenfrequency evaluated by FEA. Except the 3rd eigenmode at frequency of 49.09 Hz by EMA and 44.14 Hz determined by FEA, it can be considered that frequency error is fewer than 5%, which is a very good result for task proposal.

Figures 4 ÷ 6 comparative shows the bogie frame in the vibration eigenmodes determined by both methods, experimentally, by EMA, in the left side and numerically, by FEA, in the right side. From the examples presented in figures it can be observed a very good correspondence between the analytical and experimental determined vibration eigenmodes.

Table 1

Comparative eigenfrequencyes of the LEMA 5100 kW bogie frame obtained both from EMA and (FEA)

Mode	Mode Direction	EMA			FEA	Error [%]
		Damped Frequency (Hz)	Damping (%)	Complexity	Undamped Frequency (Hz)	
1	OZ	30.20	0.25	0.010	29.30	2.98
2	OZ	36.01	0.28	0.120	37.89	-5.22
3	OY	49.09	0.18	0.003	44.14	10.08
4	OY	61.86	0.14	0.001	61.04	1.33
5	OZ	65.66	0.13	0.003	66.43	-1.17
8	OZ – OY	81.04	0.24	0.256	83.41	-2.92
10	OY	89.92	0.11	0.001	88.86	1.18

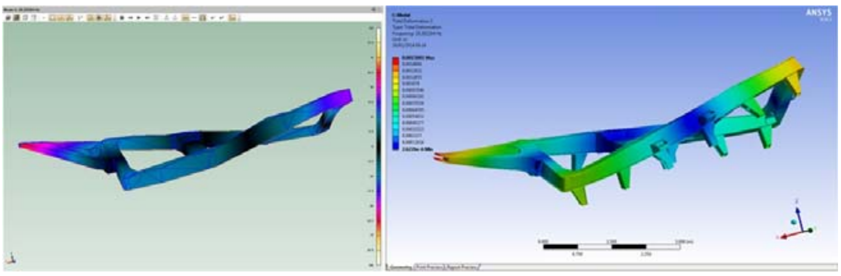

Fig. 4 – The 1st eigenmode, OZ direction, Fq = 30.20Hz by EMA, Fq = 29.30Hz by FEA.

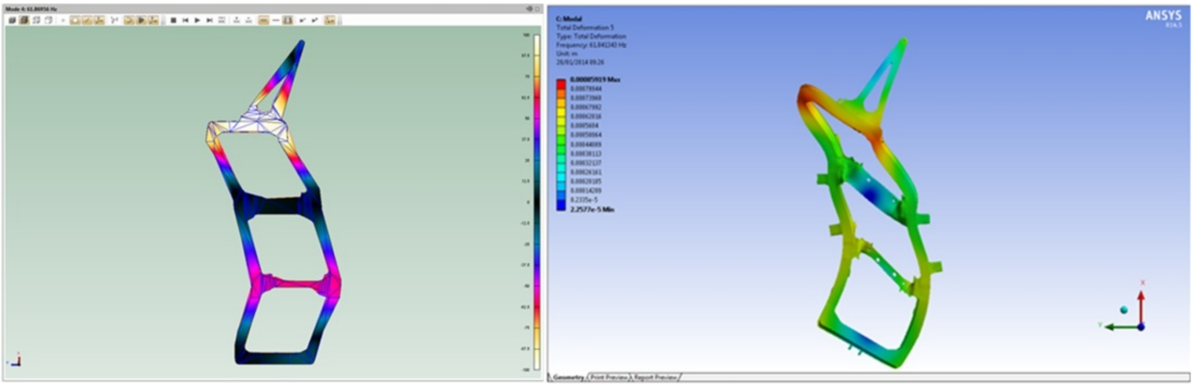

Fig. 5 – The 4th eigenmode, OY direction, Fq = 61.86Hz by EMA, Fq = 61.04Hz by FEA.

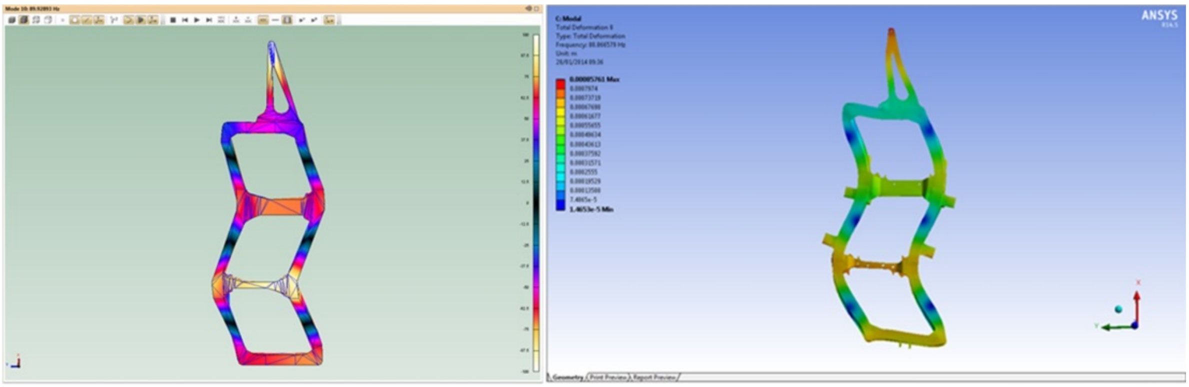

Fig. 6 – The 10th eigenmode, OY direction, Fq = 89.92Hz by EMA, Fq = 88.86Hz by FEA.

6. CONCLUSIONS

The goal of the work presented in this paper is focused on develop a methodology for calibration and validation of a FEM for complex structures using EMA. For the EM analysis the tests were automatically conducted by LabShop and the measurement data validation and analysis was performed automatically by the Pulse Reflex platform.

This methodology has resulted in a significant reduction in person number employed in performing the tests and in the time required for measured data processing.

In this paper the experimental modal analysis (EMA) was used to calibrate and validate the railway bogie frame FE model (and bogie frame design).

Analyzing data from the Table 1, modal shapes presented in Figs. 4 ÷ 6 and general data obtained from the two EM and FE analysis (data that for reasons of space could not be presented in article) it can be considered that overall error in eigenfrequencyes determination by the two EMA and FEA methods is fewer than 5%. This is a good result for the task proposal of analytical FE model calibration by experimental determined data. Modal shape in eigenmodes is the same in the two type of analysis.

Given the foregoing it can be concluded that the analytical model, developed under ANSYS, is correct and can be used in both applications, to evaluate the mechanical stresses distribution on the structure to loads applied during the homologation tests, according to EN 13749/2011 and to evaluate the bogie crashworthiness to a crash scenario, according to EN 15227/2010.

The calibrate bogie frame FE model will be used for a locomotive FE model.

With the FE model of the locomotive the virtual tests will be performed both to study the behavior and to evaluate the locomotive crashworthiness to a different crash scenarios.

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