STRUCTURAL CHARACTERIZATION OF OVERHEAD CRANE BY ITS MODAL PARAMETERS

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Abstract. *The aim of this work is the structural characterization of a overhead crane, with capacity of 120 tons in the main car, 30 tons in the auxiliary car and 22.5 m of span, by its modal parameters. The experimental modal parameters of the crane structure were obtained from the recording of its vibrations in operation, by force-balance accelerometers. Its structural characterization was obtained by adjusting the modal values of a numerical model based on the Finite Element Method to the experimental values. In order to validate the technique, the results of the numerical simulation were also compared to those of the static load test. The errors identified are small, which validates the use of modal analysis in the characterization of large industrial structures.*

Keywords: *Operational modal analysis, Stochastic modal identification, Calibration of numerical models, Static load test.*

1 INTRODUCTION

Within the industrial context, the overhead cranes play a fundamental role in the elevation and transport of loads. According to Brasil (1988), this type of structure has the objective of horizontal and vertical movement of equipment and materials in the industry and in the construction sites. In a simplified way, a overhead crane can be considered as a lifting machine composed of main beams, trucks with wheels and winch car, where the load lifting system is mounted. However, there are numerous variations on the constructive form of these equipments, depending on their application. For example, in the case of overhead cranes used in steel mills, two cars are required to move the transport container or ladle, the main car being the one that supports the ladle and the auxiliary car that carries out the tipping of this container. Fig. 1 shows a lower view of the overhead crane, object of this study, where the main and auxiliary beams can be seen.

Figure 1. Overhead crane (bottom view - main and auxiliary beams). Source: Author

Because it is an equipment with wide mobility, ie, translation, transverse displacement and possibility of turning your car, as well as braking, lifting loads, inertial forces, among others, the overhead cranes are subject to a complex set of requests which need to be precisely determined. Such requests vary over time and are often cyclical, so it is necessary to perform not only their static analysis, but also simulation of the fatigue phenomenon of their structure, for greater accuracy in their design.

With the popularization of computers and increasing their processing capacity, numerical models have been used in large scale, both in the initial design and in the life of the equipment, to monitor their behavior or to prevent failures. Thus, it is necessary to consider numerical models that represent as accurately as possible their structural behavior.

For this, the numerical characterization of a overhead crane by its modal parameters (natural frequencies, vibration modes and damping rates) consists of an efficient procedure for the construction of models that allow reliable simulations and contribute to the refined study of its structural behavior.

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According to Cismasiu *et al.* (2013), the increase in the complexity and size of the structures, together with the technological and computational progress, has led to the creation of numerical analysis tools that allow, in a simple way, a good approximation to the real behavior of the structure. However, numerical models are not always faithful to the existing structural response, and even the degradation of the structure itself must be taken into account. Therefore, it is necessary to perform the calibration and validation of the numerical models, which led to the development of modal identification techniques in order to obtain the characteristics of the structure under study.

Cassola and Pereira (1999), affirm that the knowledge of the modal parameters is fundamental for the design of a structure adapted to the conditions of use, thus becoming essential for its development. Also, according to Martins *et al*. (2016) there is a close relationship between the experimental modal parameters of a structure and the rigidities, masses, and bonding of its structural elements.

Malveiro *et al.* (2011) performed a work of experimental identification of the modal parameters of a railway viaduct with prefabricated tray and found differences between the numerical finite element model and the real structure, for natural frequencies, in case of global modes of vibration, between 9.9% to 24.8% and, for local modes, from 0.2% to 14.8%, reinforcing the need of calibration of the numerical models by the modal parameters of the real structure.

Varum *et al*. (2016) made the dynamic characterization in a historical build of the 19th century, called *Igreja Nossa Senhora das Dores* – Ceará – Brazil, identifying, through a calibrated model, points of greater probability of cracks and structural vulnerabilities, thus serving, this calibrated model, as a support tool for decision about interventions focused on the preservation and maintenance of historical buildings.

The procedure for identifying the dynamic characteristics of a structure is called Experimental Modal Analysis, which basically consists of the application of a known excitation in the structure and measurement of the response. However, for applications in large structures this procedure becomes costly, that is, for these cases, it is used the Operational Modal Analysis that consists in the identification of the dynamic characteristics of the structure when it is submitted to the excitations in normal use, without application of proof loads (Martins *et al*., 2016).

This work presents the comparative results of the natural frequency and displacement values obtained by the numerical simulation of a computational model of a steel crane, before and after calibration, by means of its modal parameters. To do so, a numerical model was first developed in finite elements, respecting the same characteristics of the real structure, then this model was submitted to the load test and its results compared to the real results previously obtained by topography. Through the operational modal test, the dynamic characteristics of the overhead crane were found and, based on these data, the finite element model was calibrated and the load test performed again. The results were compared and discussed, from which it was concluded that the dynamic test based on the operational modal analysis was validated from the static load test results.

2 DESCRIPTION OF THE STRUCURE

The structural characterization, object of this study, occurred in a overhead crane used for the handling of liquid steel ladle, with a 22.5 m span, manufactured in 1977 by the company PHB (PohligHeckel from Brazil), in operation for approximately 40 years.

The overhead crane is composed of two winch cars, considering as the main car the one that realizes the elevation of the pan, with capacity of 120 t, and auxiliary which performs the tilting of the ladle to pour the steel, with capacity of 30 t.

The structural part of the overhead crane consists of two main beams, where the main carriage rails are located, two auxiliary beams, so called because they support the rails of the auxiliary car and two head beams that have the purpose of making the connection between the beams and the wheel set for the translation of the overhead crane as a whole. Fig. 2 schematically shows the cited components.

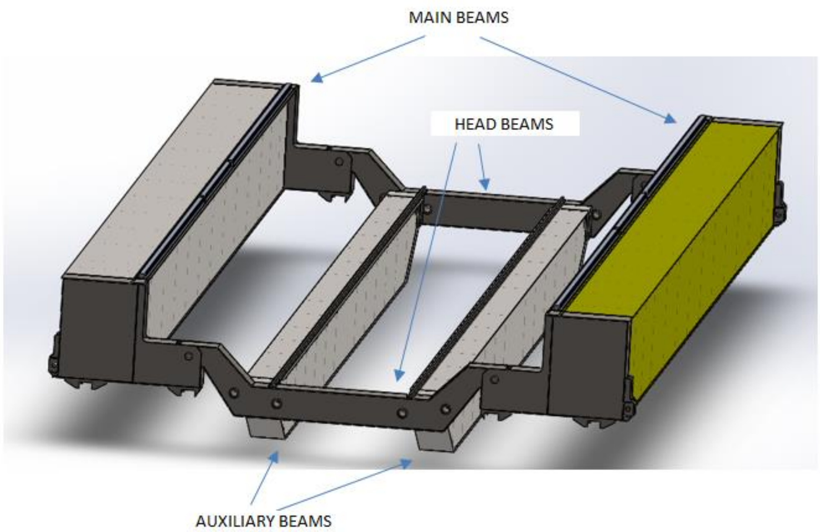

Figure 2. Schematic representation of the crane structure. Source: Author (2016)

The main beams have a total length of 22980 mm, made of sheet steel ASTM A 36, forming a box beam, having as main dimensions: width of 1600 mm and height of 2032 mm, with internal diaphragms. The tables have a thickness of 16 mm and the cores 11 mm. Table 1 shows the mechanical properties of the material of the beams.

The auxiliary beams have a total length of 21994 mm, also made of sheet steel ASTM A 36, forming a box beam with internal diaphragms, having the following main dimensions: height 1476 mm and width 700 mm. Their tables have a thickness of 13 mm and the cores 8 mm.

Finally, the head beams are composed of a closed profile, made of ASTM A 36 steel plate. These beams are rigidly interconnected with the auxiliary beams and by means of pivoting axes with the main beams.

In order to facilitate the mapping and recording of the results, in the case of the main beams, the designation was denominated as beams "A" the beams positioned to the left side and "B"

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beams located on the right side, respectively VPA to main beam side "A", VPB the main beam side "B", VAA auxiliary beam side "A", VAB the auxiliary beam side "B". In the case of head beams, the VCA was the head beam "A" and VCB the head beam "B". Fig. 3 demonstrates this convention.

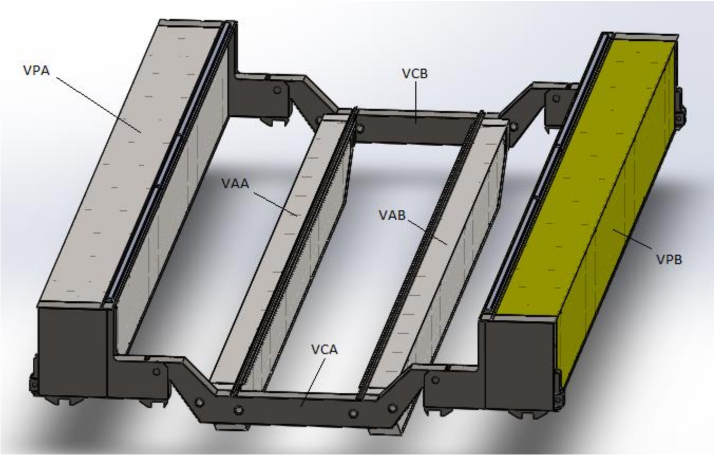

Figure 3. Designation of beams. Source: Author (2016)

Table 1 Mechanical properties of beams material

Steel	Modulus of Elasticity (GPa)	Coefficient of Poisson	Yield point (MPa)	Breaking point (MPa)
ASTM A36	200	0.3	250	400

3 DESCRIPTION OF THE NUMERICAL MODEL

The crane was modeled initially in AutoCAD software (2016), using the 3DFace entity, based on its structural executive design. All the internal ribs and diaphragms of the beams were made in order to have a model as close to reality as possible.

After this phase, the three-dimensional model was exported to SAP2000 software (16), based on the Finite Element Method. The overhead crane model consists of 2934 nodes and 3720 thin-shell elements, which respect exactly the thickness of the plates with which the overhead crane was built

The trucks and wheel sets were not modeled because they were not directly object of the analysis. Instead, links were created so that the most accurate calibration of relative rigidity between the structure and the rails could be performed.

The connection between the auxiliary beams and head beams was modeled in a solidarity manner, according to design and, to represent the pinned connection between the main beams and the head beams, 8 links were used, in order to have flexibility in the calibration of the model regarding to the stiffness of these connections. Fig. 4 shows a three-dimensional view of this model.

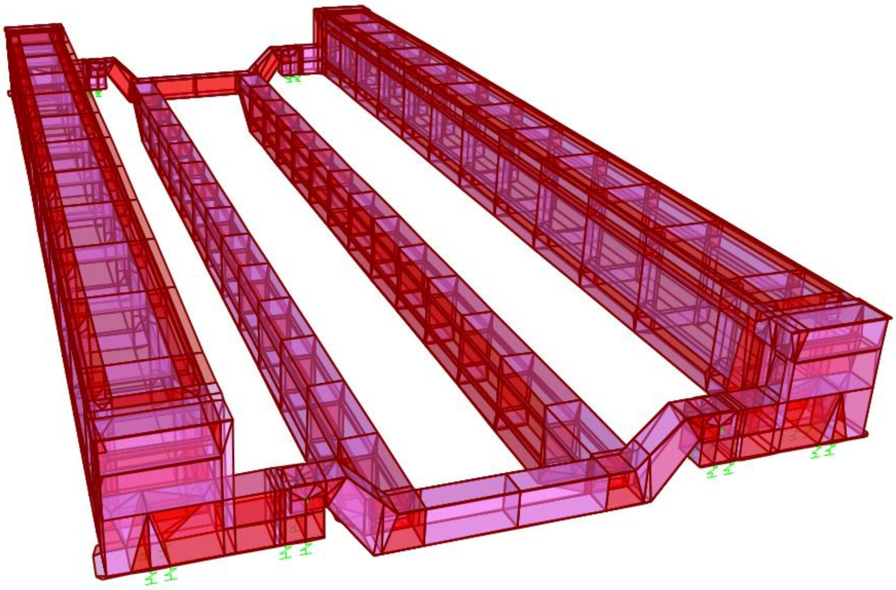

Figure 4. Model of the overhead crane in finite elements. Source: Author (2016)

The materials adopted in the model followed those specified in the project, namely ASTM A 36 steel. With regard to the actuating actions, we have basically the two cars winch (main and auxiliary), bench of resistances and electric panels in the VPA and, because it is an equipment subject to accumulation of dust, the distributed load equivalent to the powder in the upper part of any structure was considered.

Thus, the applied loads, in the form of mass, were:

- Main car: mass equivalent to the total mass of the car divided by the number of wheels (4) and applied respecting the wheel spacing of 4300 mm, equal to 12000 kg / wheel.
- Auxiliary car: mass equivalent to the total mass of the car, divided by the number of wheels (4) and applied respecting the wheel spacing of 2850 mm, equal to 3000 kg / wheel.
- Electrical cabling mass and resistance bank distributed on the top table of the VPA equal to 6200 kg
- Mass of panels and electrical components within the VPA distributed in its side panels equal to 1500 kg.
- Dust mass, the density of which was measured by means of local sampling (640 kg / m^3), distributed over the whole surface of the overhead crane, equal to 20 kg / m^2.

4 DESCRIPTION OF THE MODAL TEST

Two SYSCOM MR3000C equipment from Syscom Instruments were used for the modal test. These devices integrate triaxial accelerometers (orthogonal axes), with a sensitivity of 1.5V / g, pickup range of accelerations between -2g and +2g, acquisition with 24-bit resolution and battery power. The summary datasheet can be seen in Fig. 5.

Principle	The sensing element is an analog force feedback accelerometer featuring a variable capacitance silicon bulk-micro machined acceleration sensor (MEMS) and a custom low-power mixed-signal integrated circuit (ASIC). The MEMS/ASIC custom design forms a DC coupled analog servo accele rometer. Micro mechanical (MEMS), force balance accelerometer (FBA)
Hysteresis	None
Dynamic range (100 Hz BW)	typ. 110 dB (±4g)
Noise (10 to 1000 Hz)	typ. 900 $ng_{rms}/\sqrt{Hz}$
Noise (0.1 to 100 Hz)	typ. 11 μg_{rms}
Natural frequency	Frequence response DC to > 1000 Hz
Measuring range	±4 g standart, ±2 g
Non-linearity	< 1.5 % of full scale
Scale factor temp. drift (± 2g)	typ. <2000ppm/ºC
Zero point offset drift (± 2g)	typ. < 400 µg/ºC
Orientation	Traxial, horizontal (floor) mounting or vertical (wall mounting)
Selftest	Test-pulse
Cross axis rejection	> 40 dB

Figure 5. Integrated system Syscom MR3000C. Source: Syscom Instruments

Nine measurement campaigns were carried out with the two devices collecting the data simultaneously, one of them always being fixed at point P1 and the second device positioned at points P2 to P10, as can be seen in Fig. 6, each campaign lasting 30 minutes. The data acquisition rate was 200Hz.

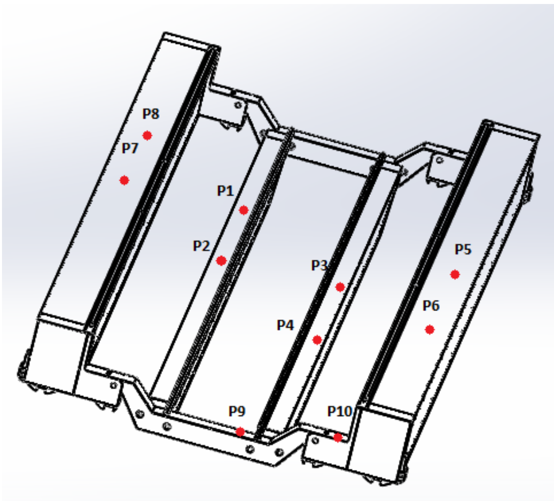

Figure 6. Identification of measuring points in the vibration test. Source: Author

As indicated in Fig. 6, the points P7, P5, P2 and P3 are at the central point of the main and auxiliary beams, respectively. The points P8 and P1 are 2/3 of the total length of the beams and the points P4 and P6 are located in the first third of them. Finally, points P9 and P10 are located on the head beam.

The acceleration measurements were taken along the overhead crane and processed using a numerical stochastic time domain technique (SSI-UPC Merged). For the processing, the software ARTeMISv3.0 was used, considering several parameter options, such as standard deviation, resolution and system order, which were omitted due to the large volume of analysis.

Figure 7 shows the main singular values of the spectral density matrix obtained. Table 2 presents the values of the real modal parameters of the structure, obtained after the data processing. It can be observed that, even with the crane stopped during the tests and without any imposed artificial excitation (the source of vibration was due to the transmission of vibration from the structure of the shed due to the wind to the overhead crane), it was possible to extract from the temporal registers the dynamic modal parameters.

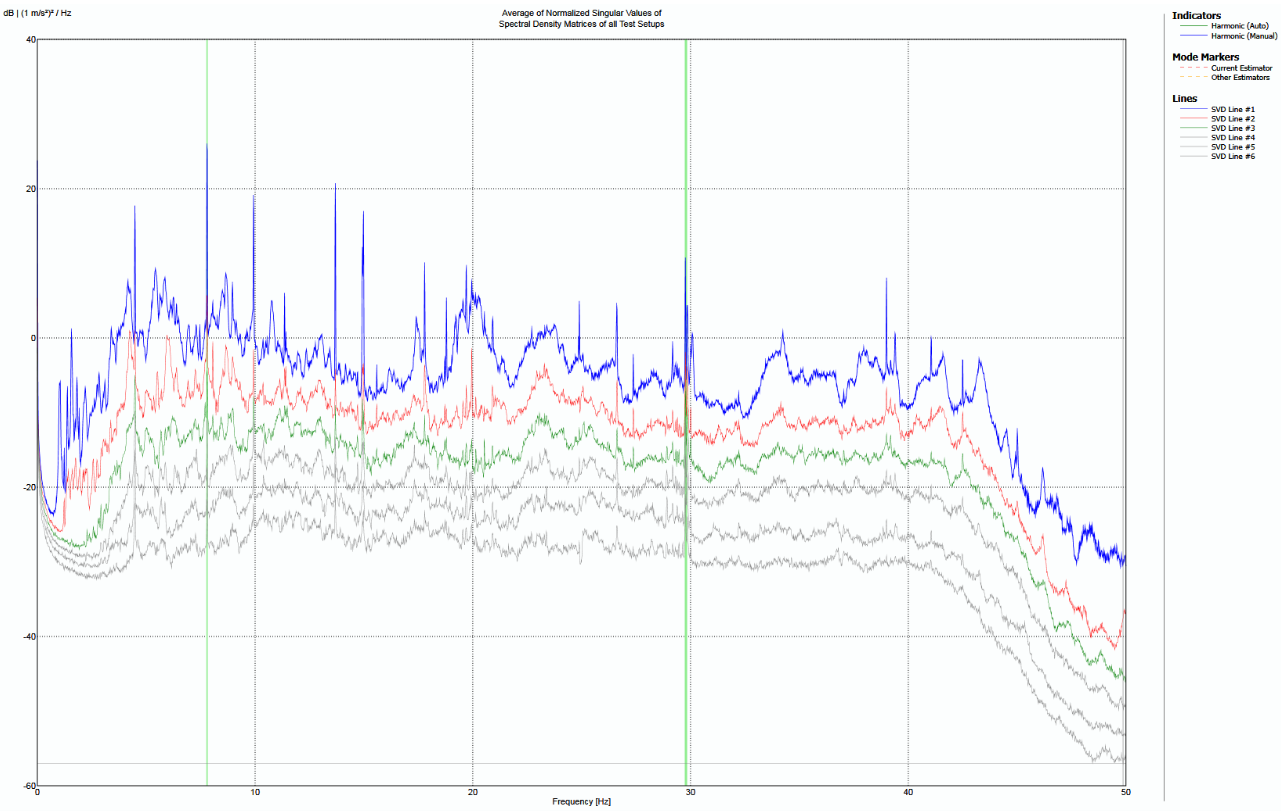

Figure 7. Amplitude x Frequency - Spectrum obtained in the vibration test. Source: Author

Table2. Experimental modal parameters

Mode	Frequency [Hz]	Damping rate [%]	Standard [%] deviation of frequency
1	4,21	3,122	0,3
2	4,56	7,828	0,2
3	5,86	3,782	0,1
4	7,58	5,971	0,1
5	7,77	1,076	0
6	8,33	5,362	0,2
7	9,09	5,972	0
8	10,56	5,717	0,1
9	12,56	6,466	0,1
10	14,92	8,252	0,2
11	15,47	9,714	0,2
12	17,48	3,217	0,1
13	17,94	4,091	0,1

14	19,60	2,832	0,1
15	19,89	4,356	0,1
16	20,04	2,792	0,1
17	22,01	5,66	0,7
18	22,93	4,263	0,1
19	23,11	7,272	0,3
20	23,75	2,477	0,1
21	26,66	4,303	0,1
22	29,57	2,747	0,1
23	29,59	6,07	0
24	33,26	6,711	0,1
25	34,09	3,751	0,6
26	35,86	8,02	0,6
27	36,29	3,093	0,1
28	38,45	9,935	0,2
29	39,03	1,807	0,2
30	39,19	4,77	0,3
31	41,25	7,45	0,4
32	41,36	3,372	0,1
33	41,84	5,128	0,4
34	42,60	6,665	0,2

5 CALIBRATED NUMERICAL MODEL

The calibration process of the model was performed in an iterative manner, with changes in the stiffness of the structure, that is, adaptation of the elasticity of the material and the bonding of the elements, to approximate the natural frequencies of the numerical model to the values of the respective experimental frequencies.

In the numerical model adopted for the development of this work, the connections representing the structure / loops connection were realized as links that had translational rigidities, in the three directions, infinite and null rotational rigidities. After calibration, the translational bonds began to present rigidities ranging from 1.10^9 to 2.10^8 N / m while rotational rigidities were kept null.

Figure 8 shows the mass participation curve of each mode of vibration. It is observed that from the mode no vibration mode mobilizes more than 5% of the mass of the structure, and therefore has little importance in the representation of this, as described in EUROCODE 8 - Part 2: 2009.

Figure 9 shows the vibration modes of the initial model, the data processing and the numerical model after its calibration. The error of the natural frequencies of the calibrated model in relation to the experimental data is also shown in Fig.10. In these figures UX, UY, UZ correspond to the translations of the structure in the directions X (longitudinal, direction of the movement of the overhead crane), Y (transversal) and Z (vertical), while RX, RY and RZ to the rotations of the structure in the directions X, Y and Z.

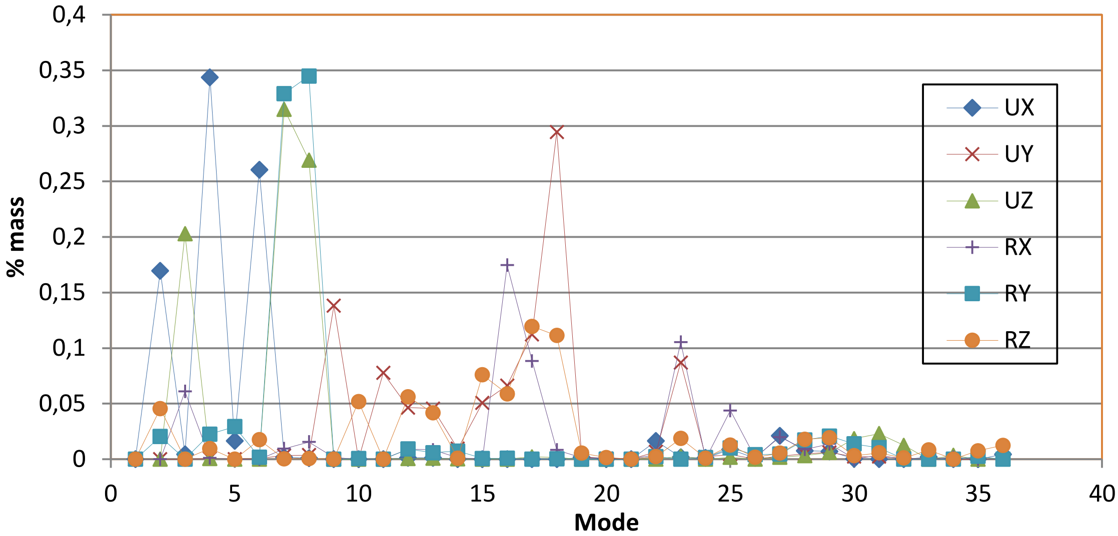

Figure 8. Mass participation x Vibration modes - Numerical model. Source: Author

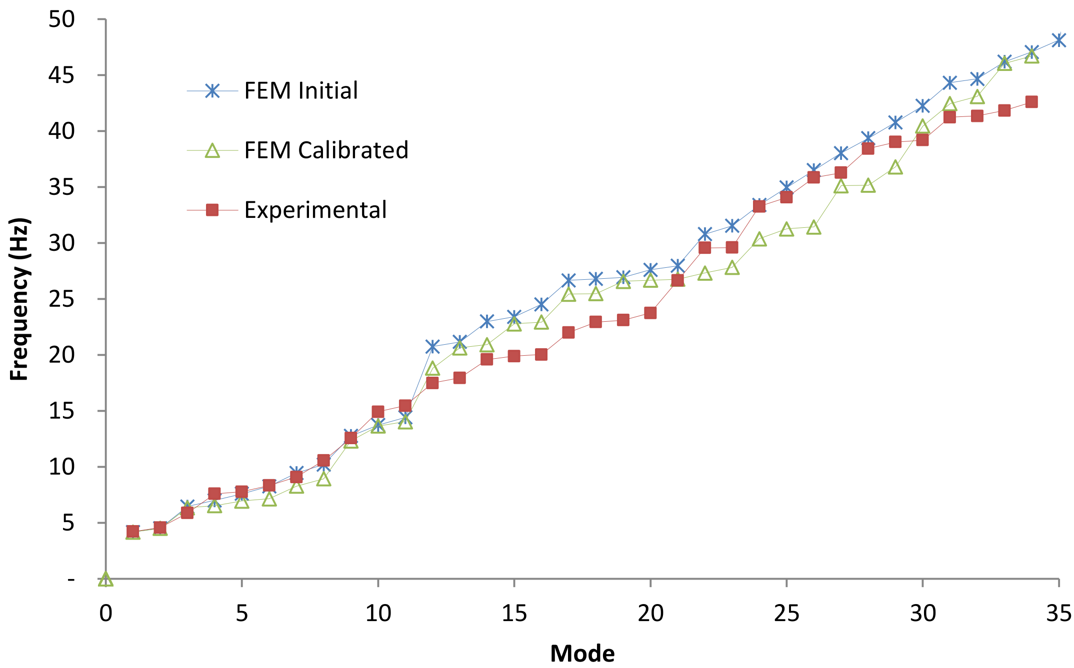

Figure 9. Frequency x Vibration mode - Numerical models and experimental results. Source: Author

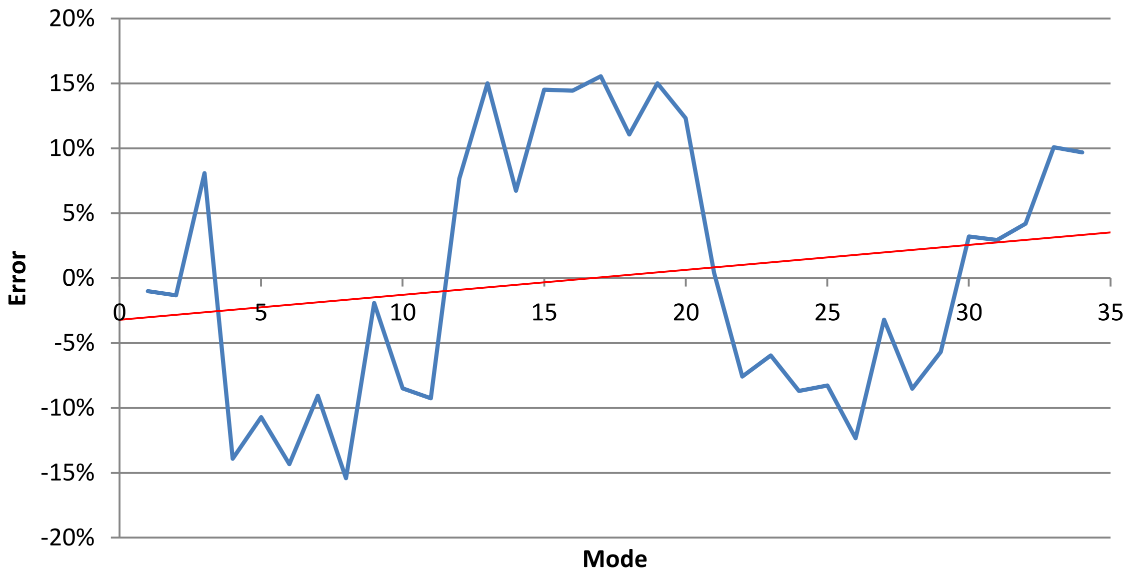

Figure 10. Error natural frequencies. Source: Author

It is observed that after the calibration procedure the resulting errors are always lower than 16.00% and the trend line (red line in Fig. 10) shows values lower than 4.00% up to 34th mode. These values indicate that the calibrated numerical model has enough precision to represent the dynamic behavior of the real structure.

6 STATIC LOAD TEST

In order to evaluate and quantify the errors to which the numerical model is subjected before and after calibration by its modal parameters, the static load test, to obtain the values of displacement at maximum capacity on the crane, were previously simulated. In this test a standard weight of 120 tons was used erected by the main car and through topography, obtaining of the values of displacement was realized, being the car in the middle of the span of the overhead crane.

The same load was applied to the numerical model, before and after calibration, Fig. 11 and 12 show the displacement values found in the static load test for VPA and VPB. The value of the bend, in any position of the span, can be obtained by subtracting the elevation of the beam without load and with load. The values of the maximum deflection for VPA and VPB, as well as the values of the bends found in the numerical model, before and after the calibration are shown in Table 3. It can be seen that the uncalibrated model presents superior rigidities compared to those of the real structure, while calibrated model indicates a good approximation of the static response of the real structure.

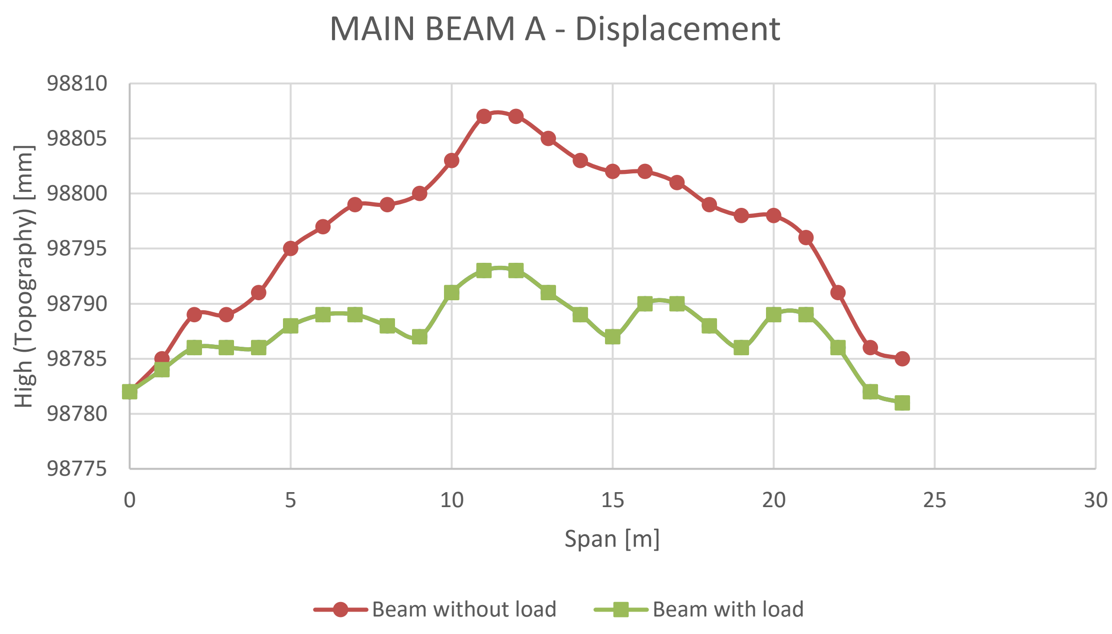

Figure 11. Static load test - Beam bend A. Source: Author

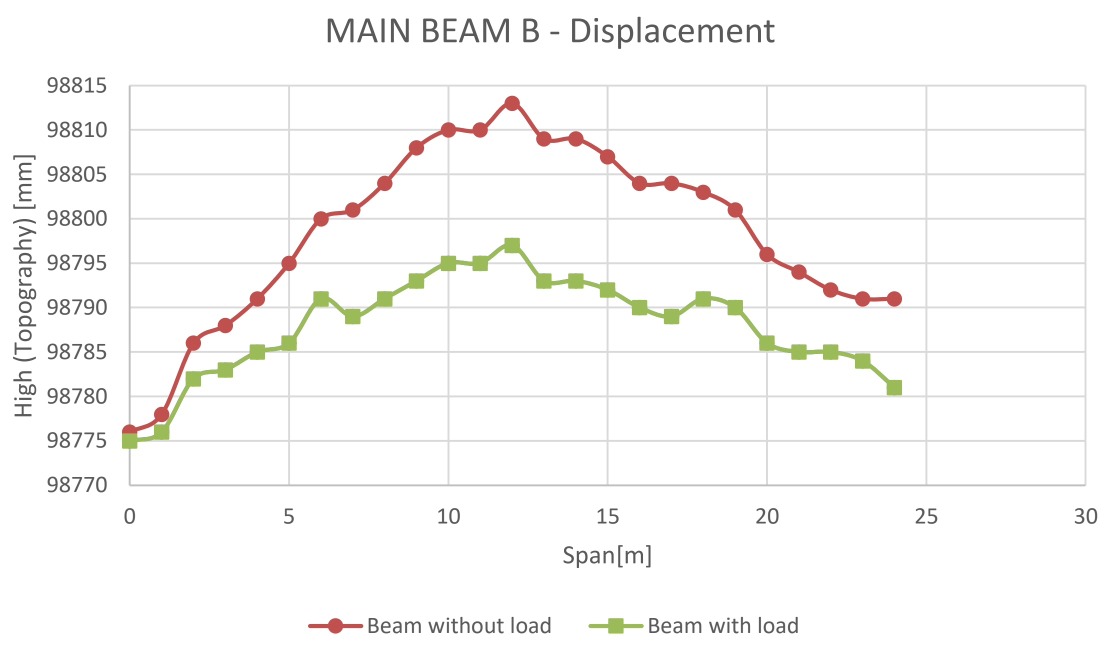

Figure 12. Static load test - Beam bend B. Source: Author

Table 3. Displacements of the main beams

	Maximum displacement VPA (mm)	Error %	Maximum displacement VPB (mm)	Error %
Static load test	15,0	-	16,0	-
Uncalibrated model	11,3	24,7	11,4	28,8
Calibrated model	14,4	4,0	14,6	8,8

7 CONCLUSIONS

This work presented the results of the structural evaluation of a numerical model, based on finite elements, of a steel overhead crane with a capacity of 120/30 t. The model was elaborated based on the executive project of the overhead crane and included in it the masses referring to all the installed equipment.

In order to perform the calibration of the numerical model, a vibration test was performed on the crane, considering unknown operational sources, thus achieving the modal parameters of the structure. In order to obtain these results, high-sensitivity accelerometers and data acquisition system with high resolution were used, which allowed the obtaining of records of vibrations in the time and later obtaining the modal parameters, even with the crane out of operation during the performance of the tests.

Once the values of the natural frequencies of the structure were obtained, the calibration of the model was started, adjusting the numerical modal parameters to those obtained in the modal test. The result of the calibration generated a numerical model with average errors of less than 4%, being able to evaluate more precisely the structural behavior of the overhead crane.

Finally, based on the results of the static load test on the crane, it was possible to compare the relative error between the bend values obtained in the numerical model before and after the calibration. It was shown that the error was reduced from 24.7% to 4.0% for beam A and from 28.8% to 8.8% for beam B, ie the uncalibrated model presented rigidity higher than the actual structure, which, depending on the case, can generate analysis contrary to security.

Therefore, the need of using accurate models for the static and dynamic study of complex structural systems was evidenced, being the calibration by its modal parameters an efficient method of adjustment of numerical models.

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