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VIBRATION FINITE ELEMENT ANALYSIS ON A PASSENGER VESSEL'S SELF-SERVICE

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Abstract

After a vibration problem on a car and passenger ferry *Lobo Marinho* built on *Estaleiros Navais de Viana do Castelo* (ENVC) in 2003 several vibration measurements were done for further analysis and identification of the problem.

This 1150 passenger's capacity vessel had as a contractual demand the certification in Bureau Veritas' *COMF-G* class. This class limits noise and vibration levels on board during the vessel's normal operation conditions.

In this article a finite elements modal analysis of the most problematic area of the ship – the self-service – is made. In this analysis is possible to validate a finite element model against the vibration measurements taken during sea trials.

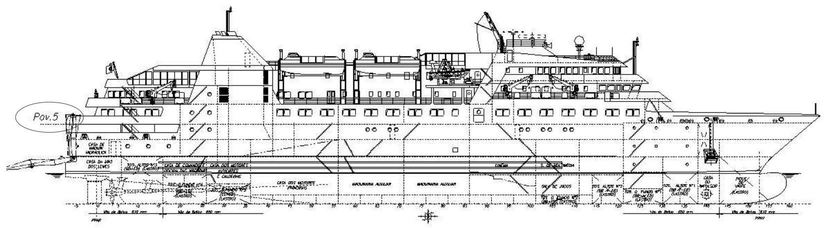
INTRODUCTION

Vibrations and noise on ships are a consequence from structural excitations caused by dynamic forces originated by propellers, main engines/turbines, shaft propellers, auxiliary machinery and waves. It is considered a good design principle to keep the main natural frequencies of the ship structure out of 15%-20% range from the main excitation frequencies (predominantly propellers and main engines).

In passengers' vessels is usual the existence of large open areas: leisure areas, restaurants, etc. It is convenient to analyse these large floors areas in the point of view of vibrations during the design development, using for instance a finite element analysis (FE). This analysis allows detecting any problem and changing the structure before sea trials, before everything is practically finished. In this article one of these large open areas with vibrations problems is analyzed using finite elements (FE).

LOBO MARINHO'S SELF-SERVICE DESCRIPTION

Lobo Marinho's profile and main characteristics are shown in figure 1.



LBP	98.20 m	Numb. of main engines	2 × MAK 16VM32
LOA	112 m	Maximum Continuous Rating (MCR)	16000 kW
B	20 m	RPM	750 RPM
D (main deck)	7 m	Numb. of propellers (CPP)	2
T (design)	4.5 m	Numb. propeller blades (each)	4
V (service)	21 nós	Rotational velocity of shaft	246.6 RPM

Figure 1- Lobo Marinho*'s profile and main characteristics.*

The main excitation frequencies of the ship and its origin are presented in table 1.

FREQUENCY EXCITATION ORIGIN	**FREQUENCY**
Propeller (*Blade Propeller Frequency* harmonics)	n × BPF = 16.44 Hz, 32.88 Hz, etc.
Main Engines (half order RPM harmonics)	n/2 × RPM/60 = 6.25 Hz, 12.5 Hz, 18.75 Hz, 25 Hz, etc.
Shaft	n × RPMshaft/60= 4.11 Hz, 8.22 Hz, etc.

Table 1- Expected Main Excitations

The self-service room is 25.4 m length and 12.8 m wide with a service and a rest area (figure 2). The structure of this area floor has 2 longitudinal girders at 3.2 m from centre line, profiles L400×120×11.5×23. This profile is also used to the web frames from aft to the frame 36 (#36), *i.e.*, practically in all the self-service. Longitudinals are bulb profiles BP100×6 with a space of 0.8 m between themselves. Other structural details were considered on the FE model.

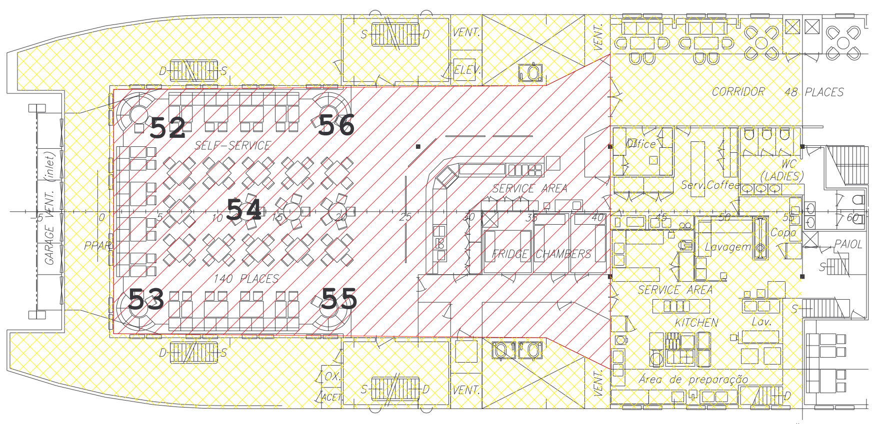


Figure 2- Deck 5, aft area until frame 60 (#60)

VIBRATION MEASUREMENTS

ENVC contracted a Danish company, *Ødegaard & Danneskiold–Samsøe* (ØDS), to perform the vibrations measurements in order to achieve the contracted COMF-G class.

The trial conditions were according Bureau Veritas (BV) classification rules: service speed (21 knots), normal draft (4.7 m), depth water under the keel (70 m), auxiliary systems and machinery operating, main engines at 90% of MCR (2×7200 kW) at 750 rpm, pitch propeller at 90% with a shaft rotation of 246.7 rpm, wind Beaufort 3, sea state force 3. It was agreed with BV the measurement of the 5 points in figure 2. The measured values as well as the admissible values are in table 2.

POINTS	VELOCITY (*Peak Hold*) [mm/s]	FREQUENCY [Hz]	ADMISSIBLE VELOCITY LIMITS (*Peak Hold*) [mm/s]
52 (AFT PS)	3.8	11.75	5
53 (AFT SB)	4.4	27	5
54 (CL)	5.6/5.1/6.7	8.3/8.0/8.3	5
55 (FORE SB)	4.7/4.1	8.5/16.5	5
56 (FORE PS)	5.9/4.3/4.2	17.0/16.0/18.5	5

Table 2- Measured points according BV Comfort class

The 3 measurements performed in point 54 exceeded the 5 mm/s admissible limit (already considering the maximum tolerances). The over limit in one point is sufficient for the refusal of the attribution of *COMF-G* class to the ship.

The measurements on the 5 points reveal problems in the vibrations levels but they were insufficient to allow any conclusion concerning the origin of the problem. ØDS continued with an experimental modal analysis of the area measuring 30 points. They determined the first 2 modes of vibration in the self-service (figure 3).

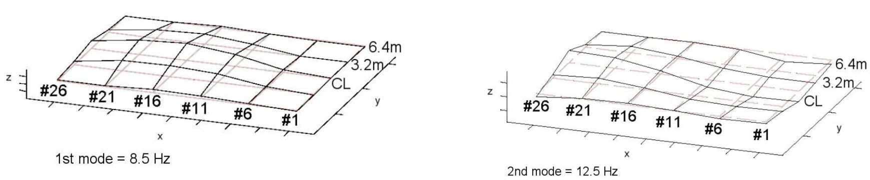


Figure 3- 1st and 2nd vibration mode in self-service area. Results from Modal Analysis.

The first natural frequency of self-service's floor occurs at 8.5 Hz and the second at 12.5 Hz. These two frequencies are very close to the second harmonic of propeller shaft and also to one of the main engines frequencies (1×RPM/60). Thus the floor was in resonance because of the existence of relevant excitations inside a 20% margin.

SELF-SERVICE MODELLING USING FINITE ELEMENTS

Usually FE modelling involves simplifications relatively to the real model (due to limitations on the transition reality/model, to reduce modelling time, etc.). Sometimes these simplifications are responsible for large differences in the results. Two models with different simplification levels were tested using Ansys FE software. In figure 2 is signed the used area to construct the models (red area). To acquire some sensitivity to these type of modelling 2 different models were used:

- Model 1: short model, detailed, where all of the elements are shell elements. It is called “short” model because, as it will be seen later, it was needed a bigger model. It is called “detailed” for being the most similar model to the reality, where all of the reinforcements are drawn in 3D (figure 4).
- Model 2: short model, simplified, where beams are now represented as lines (beam elements). This model is simpler, with a much shorter modelling time than model 1 (figure 5).

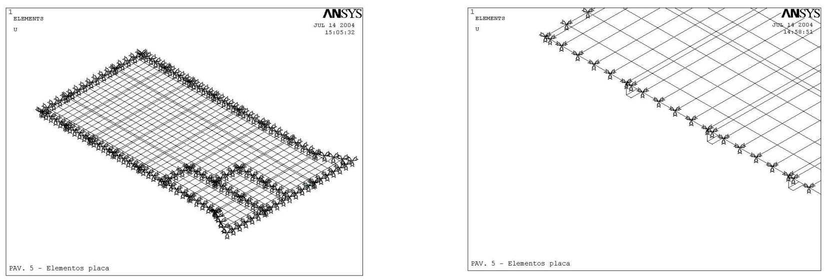


Figure 4- Structure and boundary conditions of model 1. In the right a model detail.

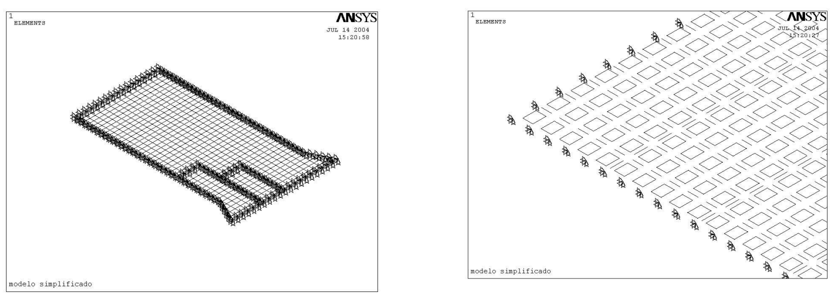


Figure 5- Structure and boundary conditions of model 2. In the right a model detail.

Model 1 uses elastic shell elements with 4 nodes and transmission capacities of bending and membrane efforts. The element has 6 degrees of freedom (DOF) in each node. Model 2 uses the same shell elements and also elastic 3D beam elements with 2 nodes and traction, compression, torsion and bending capacities. This element has the same 6 DOF in each node.

In both models boundary conditions (BC) are simply supported (SS), with all the translations restricted, substituting the existing steel bulkheads (around the whole model and in starboard also in the compartment of the refrigerating chamber and the access corridor to the kitchen). The first big difference between the models is precisely on the application of the BC. In the model 2 the conditions SS are applied restricting translations in the nodes of the plates and beams as showed in the detail of figure 5. However, to have an equivalent situation in model 1 only the 3 translations of the deck plate can be fixed (detail of figure 4). This is one of the main difficulties that happen during the BC appliance on this type of models.

The over weight of this area was estimated in 50 kg/m^2.The weight was uniformly applied on decks material by changing its density. Instead of 7.85 to the steel density was used the value 14.1.

RESULTS

Following is the comparison of the 2 models. These results were obtained using ANSYS block Lanczos algorithm.

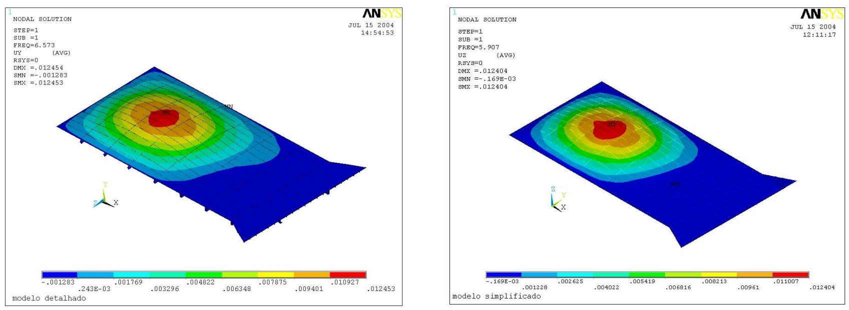


Figure 6- Model 1 (Mode 1: 6.573 Hz) vs. Model 2 (Mode 1: 5.907 Hz)

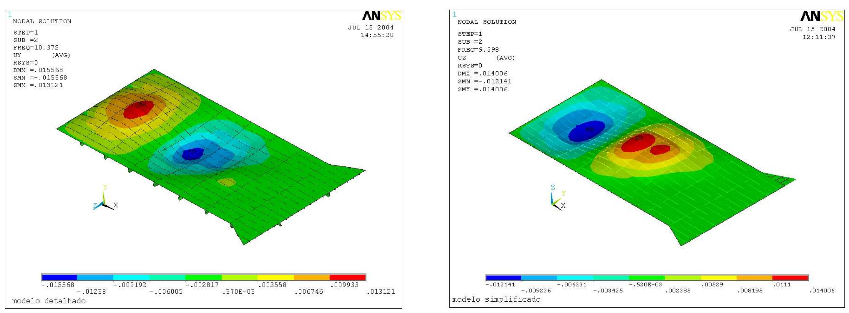


Figure 7- Model 1 (Mode 2: 10.372 Hz) vs. Model 2 (Mode 2: 9.598 Hz)

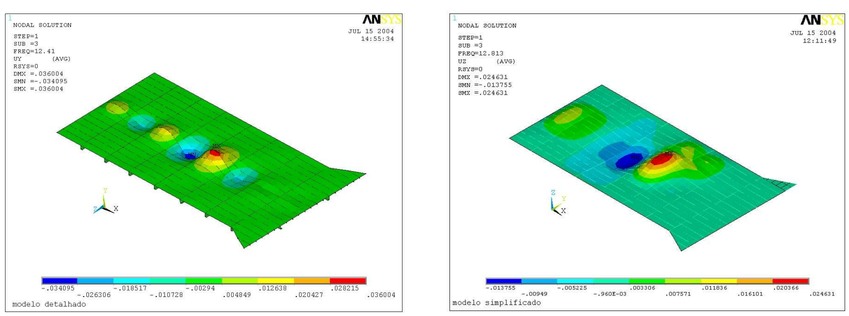


Figure 8- Model 1 (Mode 3: 12.41 Hz) vs. Model 2 (Mode 3: 12.813 Hz)

Only in the first and in the second modes (figures 6 and 7) the correspondence is evident. The mode is the same in both models. The estimated frequency by model 1 is higher than in model 2. This means that model 1 is more rigid than model 2, considering that natural frequency of the undamped structure is proportional to the relationship $\sqrt{K/M}$ (where K is the rigidity of the structure and M its mass).

In the third mode there is no longer a total correspondence between both models. In model 2 there are two web frames spacing with the same direction while in model 1 the movement changes every web frame spacing (figure 8). After the 3rd mode the direct correspondence is also inexistent.

The relative errors of these first five modes are in table 3. Seemingly the relative errors are decreasing with the modes advancing but, as seen previously, a good correspondence only exists between the first 2 modes. For this reason this conclusion in not trustworthy.

MODES **Model 1- Model 2**	**Model 1** **Freq. [Hz]**	**Model 2** **Freq. [Hz]**	**Relative Error [%]**
Mode 1 - Mode 1	6.573	5.907	10.1
Mode 2 – Mode 2	10.372	9.598	7.5
Mode 3 – Mode 3	12.410	12.813	-3.2
Mode 4 – Mode 5	13.371	13.777	-3.0
Mode 5 - Mode 5	13.561	13.777	-1.6

Table 3- Relative Errors of vibration modes (model 2 vs. model 1)

In agreement with the measurements previously described the first mode of this area occurs at 8.5 Hz (figure 3). The results from both models are far from this experimental value, being the relative error of model 1 first mode of 23%. This means that this model has less rigidity than in real one the self-service has, assuming that the uniform distribution of the over weight does not affect significantly the modes and the natural frequencies. These BC are not capable to guarantee the *clamping level* that the steel bulkheads and the remaining floor attribute to the area. After these results it was decided to rerun the two models but now with fixed BC. The obtained results for these first 2 modes with fixed BC are illustrated in figures 9 and 10.

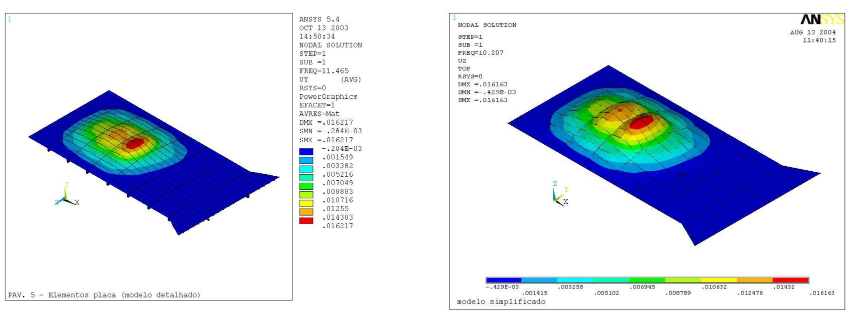


Figure 9- Model 1 (Mode 1: 11.465 Hz) vs. Model 2 (Mode 1: 10.207 Hz)

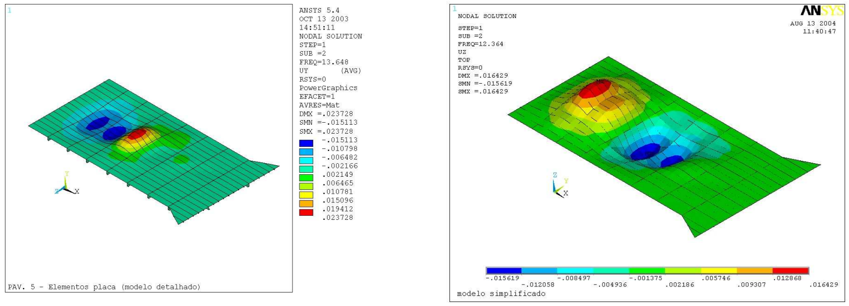


Figura 10- Model 1 (Mode 2: 13.648 Hz) vs. Model 2 (Mode 2: 12.364 Hz)

MODES **Model 1- Model 2**	**Model 1** **Freq. [Hz]**	**Model 2** **Freq. [Hz]**	**Relative Error [%]**
Mode 1 – Mode 1	11.465	10.207	11.0
Mode 2 – Mode 2	13.648	12.364	9.4

Table 4- Relative errors now with fixed BC (model 2 vs. model 1).

The relative errors are similar to the obtained with the SS boundary conditions. However, if with the SS boundary conditions the model with the closest 1st mode to the measured values was the model 1 now, with fixed boundary conditions, the closest model is model 2. The smallest relative error of the 1st natural frequency compared to the measured values is about 32%, again a very high error.

There is an important conclusion to take: the BC of the self-service of the ship are between the SS and the fixed. It is necessary an intermediate degree of rigidity, difficult to quantify and variable from ship to ship nad from model to model.

Model 2 was therefore extended, in the aft part and borders till the side shell and in the forward part until the web frame 56 (yellow part in figure 2). The intention of this extension is to perform more realistic boundary conditions to the model, conditions between the SS conditions and the fixed conditions once the additional plate increases the rotational rigidity. Figure 11 shows the results.

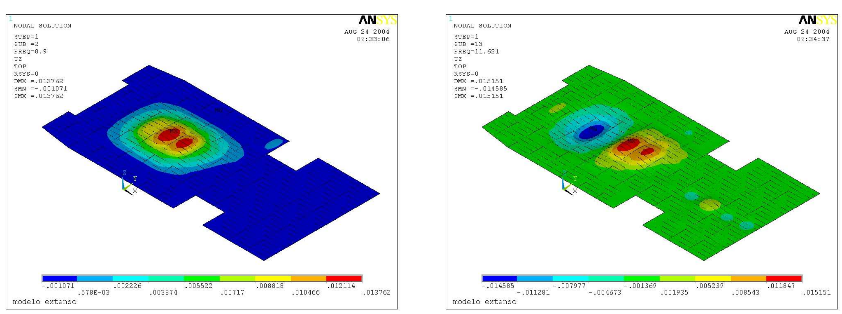


Figure 11- Extended Model (Mode 2: 8.9 Hz and Mode 13: 11.621 Hz)

The 2nd vibration mode of the all model (left side of figure 11) has higher amplitudes in the area of the self-service and it corresponds to the 1st mode of global vibration of the “rectangle” self-service. This is one of the modes corresponding to the modal analysis which determined value was of 8.5 Hz. The obtained value in ANSYS for this mode was of 8.9 Hz, what corresponds to a difference of 4.7%. This allows saying that for the 1st natural frequency of the model is quite well adjusted with the measurements.

Several local modes without interest to this analysis appear successively until the 13th mode (right side of figure 11), where the 2nd global mode of the "rectangle" self-service occurs. This mode is the other that is possible to compare with the modal analysis 2nd mode. In the following table are the results of the 2 global modes of the "rectangle" self-service compared with the measured values.

MODES	Measured Values Freq. [Hz]	Extended Model Freq. [Hz]	Relative Error [%]
Mode 1	8.5	8.90	4.7
Mode 2	12.5	11.621	-7.0

Table 5- Relative errors of extended model vs. measured values

The 2nd mode presents a difference of 7%. It is not so well adjusted as the 1st mode but, even so, this value stills inside the limits for the acceptable error's order for FEA (10%).

In a preliminary design phase of a ship is enough to obtain trustworthy information of the 1st mode and an indication of the greatness order of the 2nd mode to apart these modes around 20% or more of the main excitation frequencies.

FINAL COMMENTS

The models with a reduced dimension are not sufficiently accurate to calculate the first natural frequencies. It means that is difficult to model the restrictions imposed by the bulkheads.

The extension of the finite elements model of the deck until the side shell allows to attainment trustworthy results for the 1st and the 2nd natural frequencies.

It was not noticed substantial differences between the detailed model (model 1) and the simplified (model 2) being possible to choose the second, with the biggest easiness on modelling.

ACKNOWLEDGEMENTS

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REFERENCES

Bureau Veritas, (April 2001), "Comfort on Board (COMF)", BV rules, Part F, Chapter 6.