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EVALUATION AND CONTROL OF INDUCED VIBRATIONS FOR A PASSENGER CAR FERRY

Octavian Neculeţ
NASDIS Consulting SRL
6 Tecuci Street, Bl. V5, Et. 1
800120 Galati, Romania,
E-mail: octav.neculet@nasdis.ro

Liviu Crudu
"Dunarea de Jos" University of Galati,
Faculty of Naval Architecture,
47 Domneasca Street, 800008, Romania,
E-mail: liviu.crudu@ugal.ro

ABSTRACT

The aim of this paper is to present some examples related to the evaluation of the level of the global forced vibrations and to highlight the possible vibration problems of the ship. The calculation of the vibration level is based on Lloyd's Register's guidelines and other international standards. The hull structure has been considered using 3-D global FEM model. The dynamic linear analysis (displacements, velocities and accelerations) has been performed by NX Nastran for Windows using FEMAP as interactive graphic software program, pre and postprocessor designed for calculation codes using the finite element method. The calculations have been performed for a passenger and vehicle ferry, double-ended vessel, with boarding ramps at both ends.

Keywords: vibrations, forced vibrations, passenger-car ferry.

1. INTRODUCTION

The dynamic structural analysis has been carried out for a double ended passenger – car ferry type double-ended with boarding ramps at both ends, able to operate in all seasons in a designated service area.

Some structural particularities of this ship have led to the necessity of an evaluation of the global vibration level.

The aim of this analysis is to evaluate the level of the global vibrations and to highlight the possible vibration problems of the ship, taking into account the client's requirements regarding the level of vibration.

The present calculation of the vibration level is based on the guidelines provided by Lloyd's Register and other international standards.

The investigation of vibration is practically necessary due to some particularities of the above mentioned type of ships:

- large overhanging parts of the main deck (sponsons), loaded with vehicles, which may vibrate individually;
- large breadth that, together with the sponsons, may lead to a particular torsional mode shape;
- two propellers running simultaneously at both extremities of the ship;
- the special constructive solution of the superstructure;
- the elasticity of the wheel house related to the musts and radar supports;

The FEM analysis based on a 3-D FEM appears to be the most suitable method to investigate and to solve the structural vibration problems.

The most important problem to be sorted out is the connection with the resonance phenomena which have to be avoided using adequate structural solutions. The excitation sources to be taken into account in order to evaluate the global vibrations were the propellers and the engines.

Forced vibration and resonance may occur if the pressure impulses induced by the operating propeller or the engine external forces and moments are significantly large.

As a matter of fact, Roll-on/Roll-off (Ro-Ro) type double-ended vessel with boarding ramps at both ends has a main deck and a mezzanine deck for cars, an observation deck for passengers and a navigation deck. The starboard side mezzanine deck is accessible from the car deck by means of access ramps located at each end.

The vessel is propelled by azimuth fixed pitch thrusters at both ends and is equipped with modern and flexible machinery fuelled by either LNG or MDO.

At the level of the tween deck are placed the engine rooms, the auxiliary machinery compartments, the control room, the propulsion compartments, and the fore and aft peaks, located symmetrically in relation to the centerline of the vessel.

2. THE FINITE ELEMENT MODEL

The calculations have been carried out using two different mesh sizes (see Fig. 1):
- mesh size of one frame spacing in longitudinal, transverse and vertical directions for the deckhouse model;
- mesh size of ½ of frame spacing in longitudinal, transverse and vertical directions for the hull model.

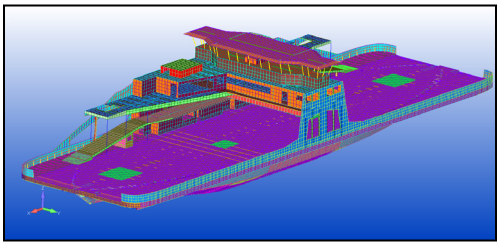

Fig. 1 FE 3D model

The following types of elements have been used for the model structure:
- plate elements for decks, sides, bottom, double bottom, longitudinal and transverse bulkheads, and for the primary member webs;
- bar elements for secondary stiffeners and the face plates of primary members.

3. STRUCTURE EVALUATION

The evaluation has been carried out in two stages:
- the first one deals with the evaluation of the mode shapes and natural frequencies in order to be able to find solutions to avoid the resonance phenomenon;
- the second one refers to the forced vibrations impact.

3.1 Mode shapes and natural frequencies

During the vibrations analysis process, both ship masses together with equipment and added masses have to be considered according to classification societies requirements. Theoretical aspects of the hull girder mode shapes are presented in Fig. 2.

In order to be able to take into account both the influences of blade frequencies and engine exciting frequencies respectively, a range of frequencies between 1 Hz and 36 Hz has been considered for the calculations.

The natural frequencies of the ship are illustrated in Fig. 3 for a range between 1 Hz and 8 Hz, only.

The additional masses were calculated according to reference [4]. The ship was divided into 8 transoms with the same length. In this stage of the calculations the additional masses were introduced in the model together with the ship masses, by the correction of the density.

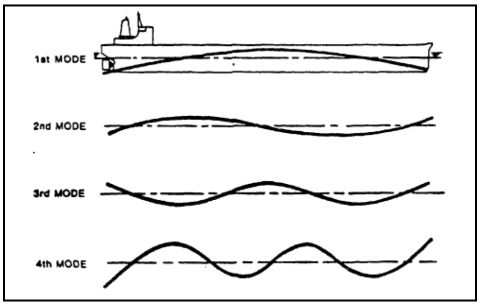

Fig. 2 Vertical hull girder of 2-5 nodes

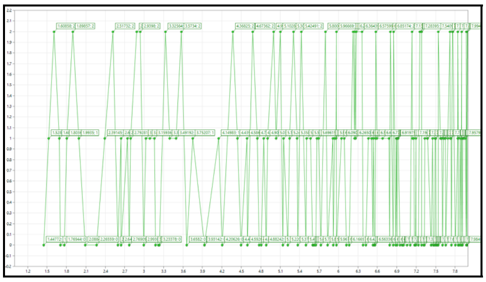

Fig. 3 Modal frequency between 1-8 Hz

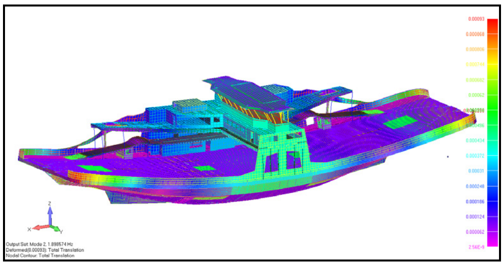

Fig. 4 The first (1st B) bending mode shape of the hull

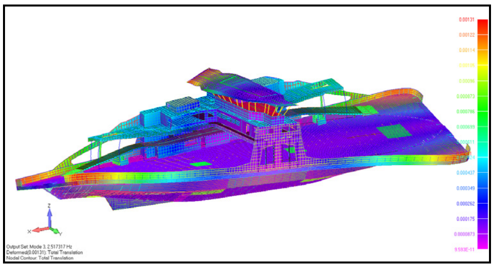

Fig. 5 The first (1st T) torsional mode shape of the hull

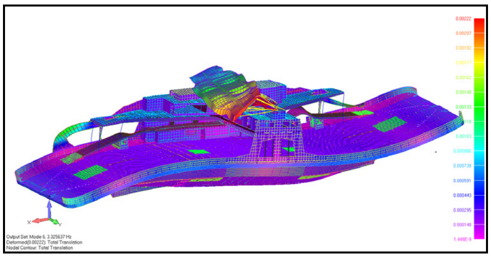

Fig. 6 The second (2nd B) bending mode shape of the hull

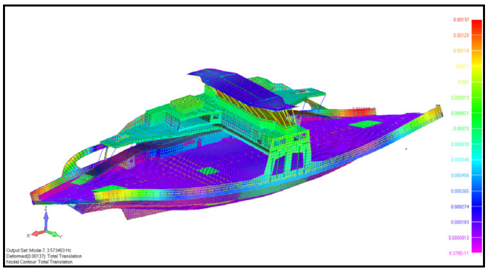

Fig. 7 The (2nd T) torsional mode shape of the hull

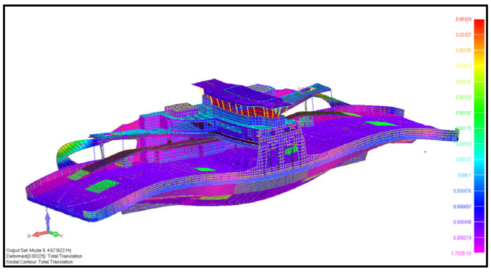

Fig. 8 The third (3rd B) bending mode shape of the hull

3.2 Forced vibrations

The source of forced vibrations is the pressure impulses induced by the propeller as well as the external forces and moments due to the engines.

Due to the lack of symmetry of the superstructures both travelling directions of the ship have been considered. Moreover, in order to consider the different power allocations of the propellers, several combinations were used: 50%-50%; 60%-40%; 70%-30%, 85%-15% and 100%-0%.

Based on CFD calculations, the areas where the induced pressures by the propeller are acting have been identified. These areas are presented in Fig. 9 and Fig. 10.

The damping was considered as a combination between the structural contribution, the hydrodynamic one, lack of cargo, etc. According to classification societies rules the damping is in the range of 2% - 4%.

The forced vibrations evaluation was carried out using the 1st and the 4th blade harmonics.

The main engines do not induce strong vibrations due to the special dampers which have been modelled as springs elements and their masses have been modelled using mass type elements placed in the centre of gravity. The evaluation was performed in the range of 1 Hz to 36 Hz with an interval of +/- 2,5 Hz around the 1st and 4th blade frequency and engine harmonics (see Table 4).

The total pressure acting on the aft and fore part of the hull is a combination between components given by the cavitating and non cavitating conditions of the propeller:

pz = (po + pc), where

po – non cavitating propeller,

pc – cavitating propeller.

The four harmonics of the main engines which have been taken into account, illustrated in Table 2, are: N/2; N; 3/2N; 2N

Table 1 The pressure impulse at different percentages of the power thrusters

%	N [rpm]	p_0 [kPa]	p_c [kPa]	p_z [kPa]	1-st order Freq. [Hz]	4-th order (Blade freq.) [Hz]
70	179.9	1.422	2.827	3.165	3	12
30	77.1	0.261	0.519	0.581	1.28	5.13

Table 2 Engines major harmonic frequencies

	rpm	N/2 [Hz]	N [Hz]	3/2N [Hz]	2N [Hz]
Aft	N = 700	5.83	11.66	17.49	23.32
Fore	N = 300	2.5	5	7.5	10

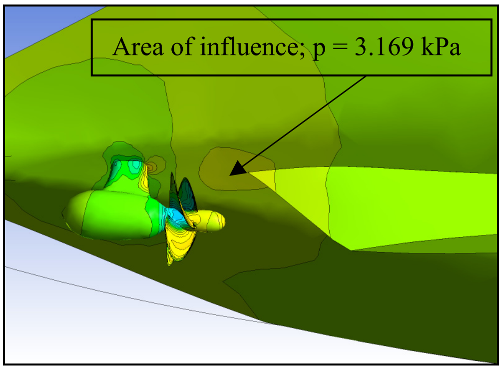

Fig. 9 Aft pressure on the shell area

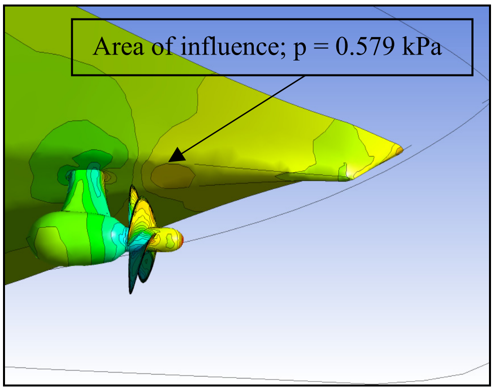

Fig. 10 Fore pressure on the shell area

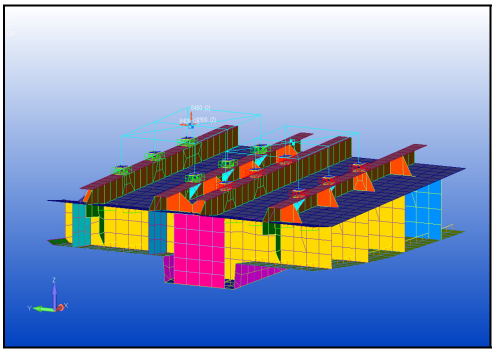

Fig. 11 Engines - External loads

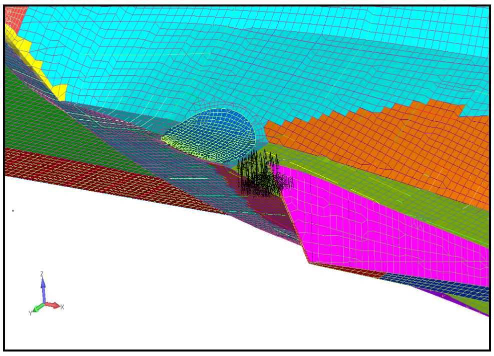

Fig. 12 Aft pressure on the shell

4. RESULTS

The maximum vibration levels for passengers and crew spaces according to ISO standards are presented in Table 3.

Table 3 Maximum vibration level for passengers and crew spaces (human reaction)

	ISO 6954: 1984 Peak velocity (5–100 Hz), mm/s	ISO 6954:2000 Frequency weighted (1–80 Hz) velocity mm/s rms
Passenger cabin Luxury	2.5	2.1
Passenger cabin Standard	4.0	2.4
Public spaces	4.0	3.0
Open recreation decks	5.0	3.5
Accommodation and navigation spaces	5.0	3.5
Work spaces	6.0	5.0

Table 4 Range of frequencies

Thruster power	70% aft		30% fore	
	1-st order blade freq. [Hz]	4-th order blade freq. [Hz]	1-st order blade freq. [Hz]	4-th order blade freq. [Hz]
lower limit	**0.5**	**9.5**	**0**	**2.6**
excitation freq.	**3**	**12**	**1.2**	**5.1**
upper limit	**5.5**	**14.5**	**2.7**	**7.6**

The results are graphically presented in Figure 13 for a power allocation of 70% and in Figure 14 for a power allocation of 30%, both cases showing that the values are within the accepted limits already mentioned in Tab. 3.

It has to be underlined that the results presented in the paper are the final ones, obtained after the evaluation of several solutions regarding local reinforcements.

The velocity amplitudes and the global deformation of the ship structure on z direction are shown in Fig. 15.

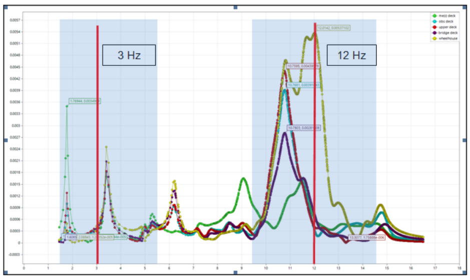

Fig. 13 Aft propeller first harmonic and 4-th order harmonic Velocity amplitude in z direction

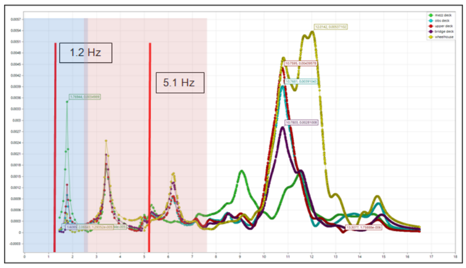

Fig. 14 Fore propeller first harmonic and 4-th order harmonic Velocity amplitude in z direction

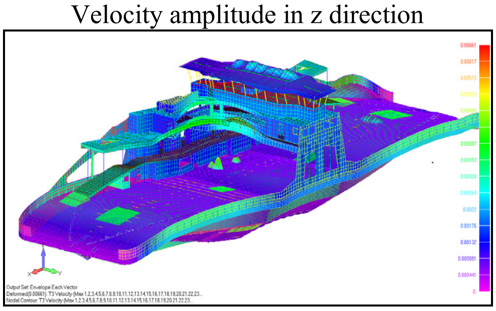

Fig. 15 Forced response – Velocity amplitude in z direction

5. CONCLUSIONS

As first step, the analysis of the results has led to the conclusion that the power allocation percentages 85%-15% and 100%-0% cannot be considered because of the much extended area of the required reinforcements and the velocity amplitude values which exceed the allowable values of the rules.

Thus, the cases 50%-50%, 60%-40%, 70%-30% were considered to be acceptable in both directions. The recommended power allocation for propulsion in both navigation conditions is considered to be 70% being acceptable from the vibration point of view. Moreover, the hydrodynamic tests have suggested the same power allocation from the point of view of the propulsion efficiency. Any other power allocation exceeding 70% will lead to discomfort due to the velocity amplitudes on decks which are higher than the allowable values provided by the rules.

REFERENCES

[1]. **Lloyd's Register,** *Rules and Regulations for the Classification of Ships*, July 2012, incorporating Notice No. 1, 2, 3, 4 & 5.

[2]. **Bureau Veritas** - *Building and Operation of Vibration-Free Propulsion Plants and Ships*, NR 207 SMS E, November 1987.

[3]. **DNV** – *Prevention of harmful vibration in ships*, July 1983.

[4]. **The Society of Naval Architects and Marine Engineers** - *Guidelines for Prevention of Excessive Ship Vibration*, November, 1980.

[5]. **ABS** – *Guidance Notes on Ship Vibration,* April, 2006.

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