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STUDY OF SUPERSTRUCTURE AND MAST VIBRATIONS / FREQUENCIES REDUCTION SOLUTIONS FOR A MARITIME TUG

Alin Marius POHILCĂ
"Dunarea de Jos" University of Galati,
Faculty of Naval Architecture, Galati, Domneasca Street, No. 47, 800008, Romania,
E-mail:alin.pohilca@ugal.ro

Daniela Mihaela LOICĂ
"Dunarea de Jos" University of Galati,
Faculty of Naval Architecture, Galati, Domneasca Street, No. 47, 800008, Romania,
E-mail: daniela.0911@yahoo.com

Florin Renardo TEODOR
"Dunarea de Jos" University of Galati,
Faculty of Naval Architecture, Galati, Domneasca Street, No. 47, 800008, Romania,
E-mail: florin.teodor@ugal.ro

Costel Iulian MOCANU
"Dunarea de Jos" University of Galati,
Faculty of Naval Architecture, Galati, Domneasca Street, No. 47, 800008, Romania,
E-mail: costel.mocanu@ugal.ro

ABSTRACT

Onboard vessel heavy rotating equipment and environmental conditions (sailing in waves) are a major source of onboard vibrations that may affect the vessel structure and equipment. Vibrations are propagated through the vessel structure from the engine room all the way up to the superstructure affecting the installed masts. All the equipment installed on the superstructure and the mast, such as antennas, navigation lights, radars etc, are affected by these vibrations which can lead to malfunctioning equipment with major implications in the sailing fitness of the vessel itself. This paper presents the analysis of a tug superstructure and mast vibrations problems caused by the main engine and the propeller rotation. Solutions to optimize the structure and to improve the transmission model of these vibrations are also described.

Keywords: mast, vibration, structural analisys.

1. INTRODUCTION

The proposed research has been carried out for the ASD Tug 3212, built by Damen Shipyards Group in accordance with the requirements of push-pull, escorting, towing and fire-fighting operations. The ASD design takes advantage of all Damen shipyard experience building these technical ships, allowing the constructor the really innovate the vessel concept.

The Damen Tug 3212 design main operational capability is to assist LNG and oil terminals in different offshore conditions. The tug has been also successfully used by owners in the mining industry for loading operations at sea, specifically in Northern Australia.

The tug class notation and its capabilities have been chosen to allow the vessel to operate worldwide in significant wave heights of up to 3 m. In Figure 1 an ASD tug 3212 is presented.

In Table 1 the main characteristics of this tug are presented.

Figure 1

Table 1

Length overall [m]	32.7
Breadth overall [m]	12.82
Depth at side [m]	5.35
Draught aft [m]	5.35
Full displacement [ton]	774
Power [kw]	5050
Bollard pull [ton]	80
Speed [knots]	15.3

In Figure 2 and 3 the superstructure and the installed mast are presented.

Figure 2

The equipments in figure 2 and Figure 3 can be affected by the vibrations of superstructure.

The aim of this analysis is to evaluate the natural frequency for the superstructure of the tug 3212. Hotspots with vibration con-concentrators have been identified with the purpose of providing solutions to address potential issues.

It is important to be able to calculate the specific structural natural frequencies to facilitate the avoidance of resonance phenomena. In case resonance occurs, specific structural solutions are available in the industry shipbuilding standards that should be employed as applicable.

The vessel propulsion system comprising of two propellers, two shaft lines and an elastic wheelhouse structure that supports the radar and the mast, indicates the necessity of a vibration analysis to identify any unwanted effects that may cause damage to navigational equipments.

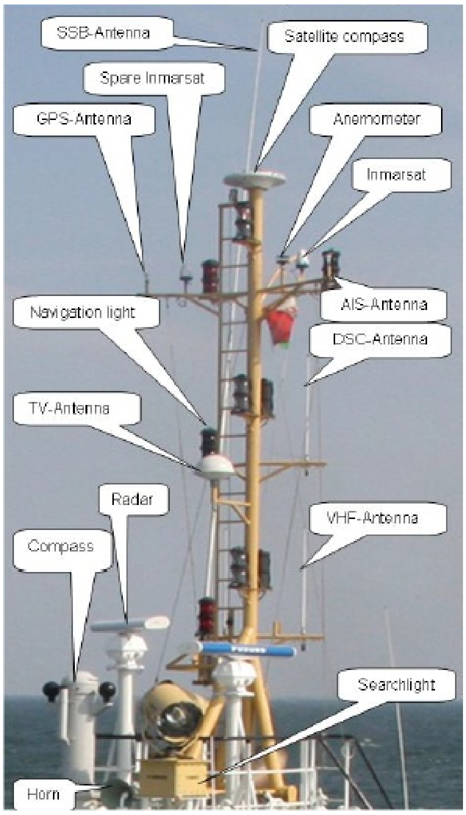


Figure 3

The calculation has been performed using NX Siemens using the finite element method.

Main excitators considered for this analysis were the main engines and the propulsion system (the two propellers more exactly).

The superstructure of the tug contains also the mast construction, which plays a prominent role in supporting the navigation lights and the aerial antennas.

In order to get accurate results, the weight of the equipments need to be taken into account during the frequency analysis.

2. MODAL ANALYSIS THEORY

Modal analysis is a technique that can provide us with an overview of a certain system amplitude response for a particular input.

Modal analysis allows the user to calculate the natural frequency of any given shaped object with respect to free vibration phenomena. Finite element analysis is usually used to perform such calculation due to the infinite variations of vessel shapes, in our case, method that returns recognized and acceptable results.

The classical approach to solve the vibration equation usually takes into consideration 4 elements : the excitation force, the stiffness or the rigidity of the system, the report between the natural frequency of the systems and the excitation frequency and the critical damping factor.

In accordance with the 4 above mentioned elements, there is a possibility to intervene in a consistent approach to improve the vibration model as described in Figure 4:

- **Excitation force** – the most efficient way is to improve the flow through the propeller
- **Stiffness** – usually its preferred to increase the system stiffness rather than decrease it. This solution is rather less efficient than others.
- **Resonance avoidance –** changing the number of propeller blade or propeller RPM
- **Damping –** increasing the damping is the less efficient method due to the fact that the vessel damping is usually fixed.

The goal of such solutions is to improve the behaviour of the structure and its performance in a given set of parameters.[1]

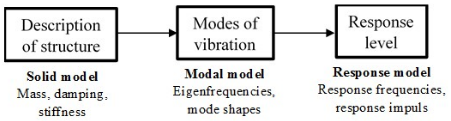


Figure 4

The solving of the vibration system as per Figure 5 shall be done for both the free vibration scenario (usually the first 5 modes are sufficient to be identified) and forced vibrations. [5].

The vibration of the mast can be approached with the excitation of the base from a mass - spring system. The base corresponds to the top of the wheelhouse where the mast is fitted. The material of the mast has properties which provide a spring-damper system. A graphical representation of such a system is shown in Figure 5.

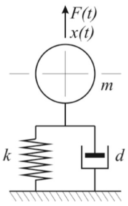


Figure 5

Figure 5 describes a one-degree system that is comprised of :

- Rigid body mass, **m**
- **k** – spring rigidity
- damping ratio , **d**

The system is characterised by linearity for the spring rigidity, **k** and damping ratio coefficient, **d** and the equation describing the system is as per below :

$$m\ddot{x} + d\dot{x} + kx = F(t)$$

with the solution which has the form:

$$x(t) = e^{(-\delta t)}\left[C_1 e^{(i\omega_d t)} + C_2 e^{(i\omega_d t)}\right]$$

where:

- $\delta = \dfrac{d}{2m}$;
- $\omega_d = \sqrt{\omega_n^2 - \delta^2}$;
- $\omega_n = \sqrt{\dfrac{k}{m}}$.

The principal natural frequency analysis usages are the following:

- Superstructure vibration analysis;
- Mast vibration analysis;
- Full model vibration analysis and optimization.

3. NATURAL FREQUENCY ANALYSIS USING FINITE ELEMENT METHOD

3.1 Basic

On board the ASD tug, it was considered that the main excitators frequencies are:

- wave frequency
- propeller rpm i.e blade passing frequency
- main engines ignition frequency

The wave frequency is caused by the waves bashing against the hull of the ship. The blade passing frequency is caused by the propellers which rotate at a certain speed with a certain number of blades. The blades pass the casing of the vertical driveshaft in combination with a different flow of water at the extremities of the propeller (top and bottom). This causes a difference in load on the propeller and results in a vibration phenomena. The ignition frequency is caused by the engine which ignites fuel a certain number of times per second [4].

3.1 Modeling with NX NASTRAN

For the analysis of the own vibration modes of the superstructure-mast assembly, the NX NASTRAN finite element analysis program is used.

Although there are a lot of applications in the Siemens NX package, only three of these applications will be used during this study:

- Modelling;
- Advanced Simulation;

Modelling application is used to create and edit the geometry. Modelling provides tools for creating and editing the CAD model made with solids or surfaces. The geometry created using this application is contained in the **.prt** files. The entities of the **prt** file can be any kind of solids, surfaces or curves and construction (additional) geometry. Two different ways can be used in NX to create a model: History Mode (parametric model) and History-Free Mode (without parameterisation).

The goal of the analysis is to calculate the natural frequencies of the mast and wheelhouse of ASD Tug 3212 and compare them with excitation frequencies range of shaft and blades frequency. Natural frequency analysis was done in Siemens NX9.

In order to accomplish the proposed study, the superstructure of the tug was generated in the modelling aplication, starting from the general construction plan drawings. As input data it's also important to have provided information about the propeller and engine to be able to compare the results.

Model extension is shown in Table 2.

Table 2

Direction	Extension
Longitudinal	FR28-150 – Fr48+100
Transversal	4300 mm PS–4300 mm SB
Vertical	Above main deck

Figure 6 shows a transversal section on body tug at frame 30.

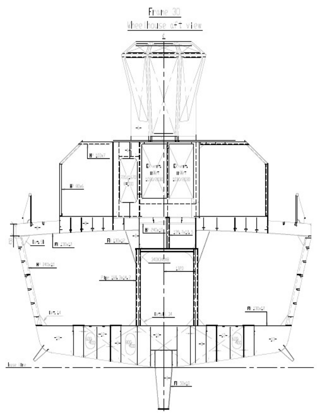
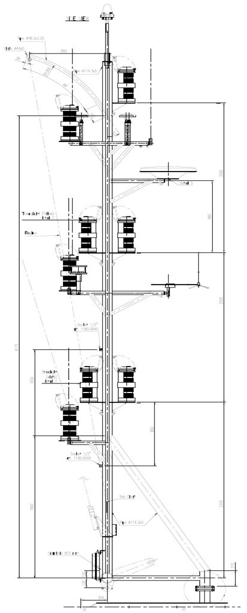
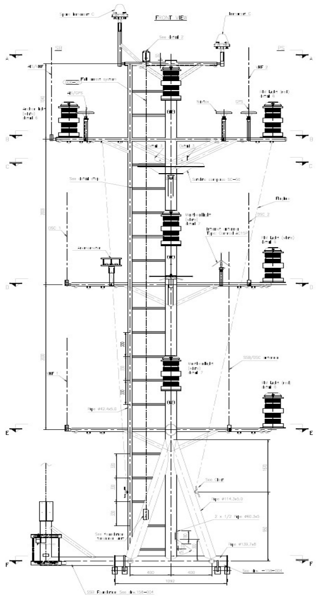


Figure 6

In Figure 7 the side (7a) and front (7b) view of the arrangement mast are presented.



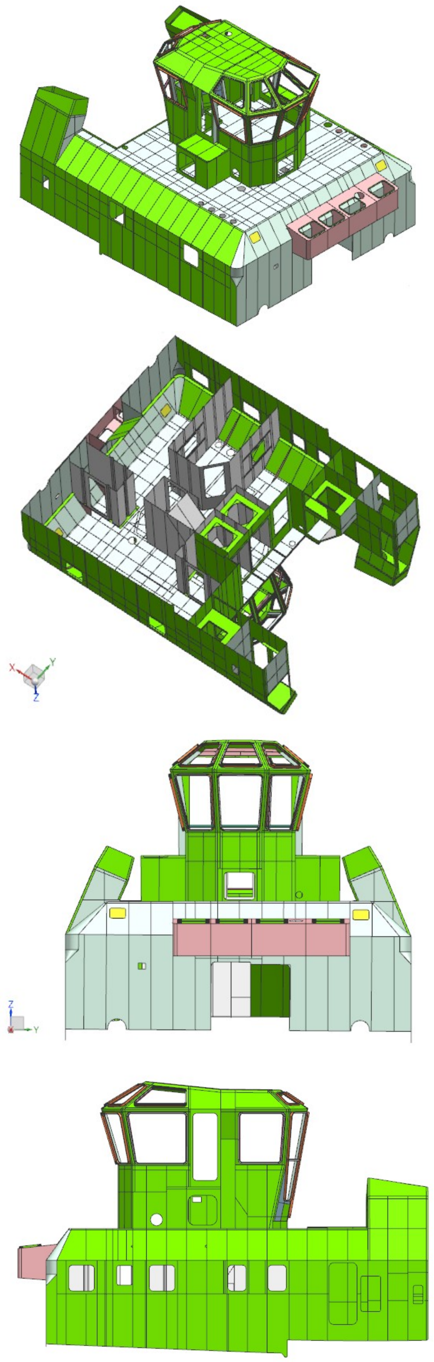
a)



b)

Figure 7

Geometry model was created by means of plate bodies according to the construction plans. Geometry model is shown in Figures 8.



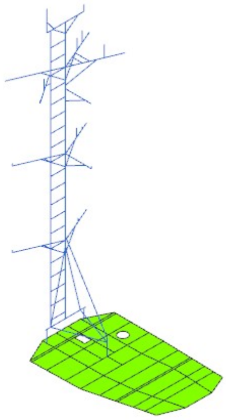
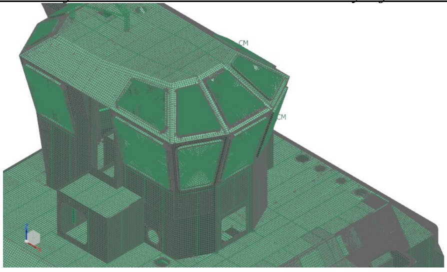


Figure 8

The material used for this analysis is steel grade A. Steel grade A is usually employed in the shipbuilding industry due to its homogenous properties (the material properties are identical in every direction)
Finite element model is shown in Figures 9. The model was created using CQUAD4 elements. The mesh has an average size of 50 mm. Secondary stiffeners were simulated using CBEAM elements. Navigation equipment on the mast was simulated using 0D CONM2 elements connected to the mast by means of RBE2 rigid elements.



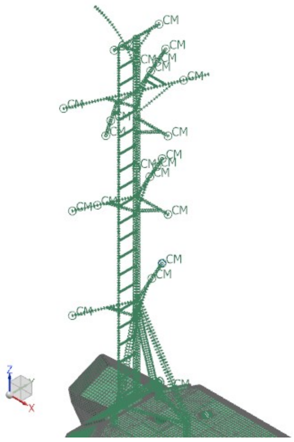
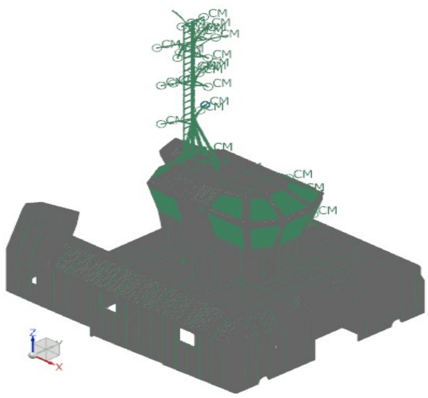


Figure 9

For the studied structure fixed constraints were considered at all points where the superstructure would normally be attached to the main deck. See Figure 10. Red lines indicate the fixed constraint.



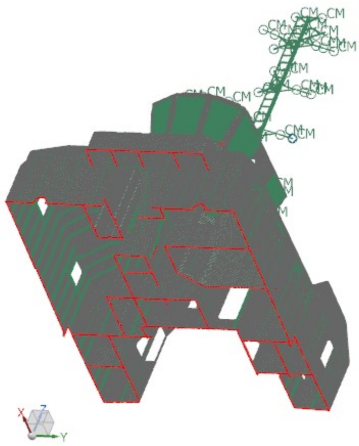


Figure 10

3.2 Numerical results

Excitation frequencies and investigating ranges are presented in Table 3. Safety margin of 20 % is used for analysis.

Table 3 - Excitation frequencies and investigating range

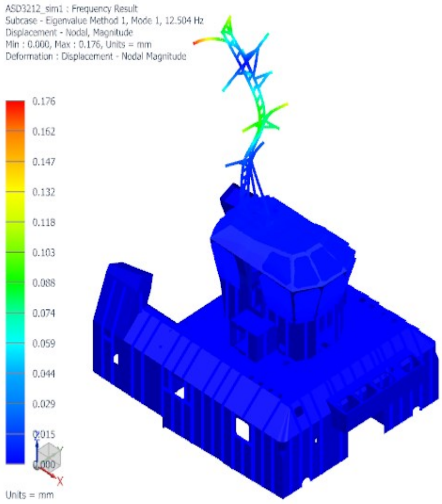
Excitation source	$f_{(at\ 100\%)}$ *[Hz]*	$f_{low\ -20\%}$ *[Hz]*	$f_{upper\ +20\%}$ *[Hz]*	$f_{upper} - f_{low}$ *[Hz]*
Propeller shaft	16.7	13.36	20.04	6.68
Propeller blade	14.5	11.6	17.4	5.8

To obtain the analysis ranges, the engine and propeller data must be known. These information are provided in Table 4.

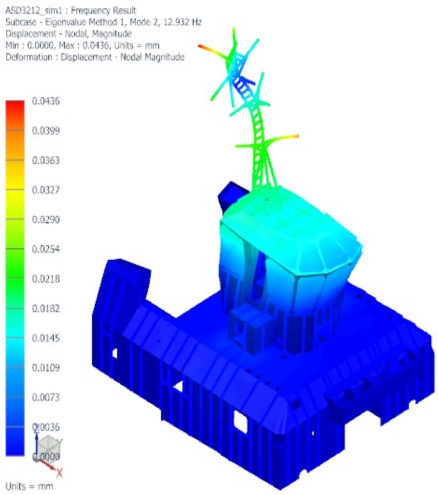
Table 4 – Engine and propeller data

Engine	Type and rating	Wartsila 8L26 / MCR
	Brake power	2610 [kW]
	Number of revs	1000 [rpm]
Propeller	Diameter	2800 [mm]
	Number of blades	4
	Number of revs	218 [rpm]

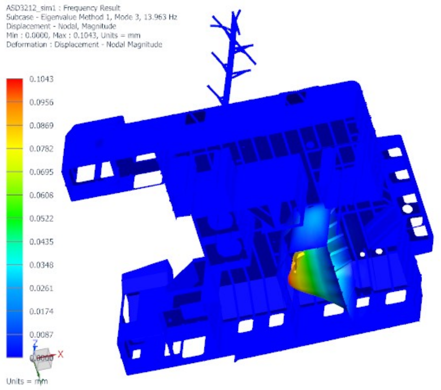
Running the natural frequency analysis, it eight frequencies have resulted in the range of 11 – 20 Hz. The solver used was NX Nastran and the solution type was SOL 103 Real Eigenvalues. Results of the analysis are presented below in figures 11.



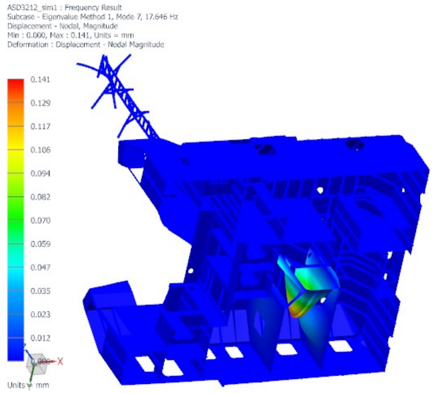
a) Mode 1 - 12.5 Hz



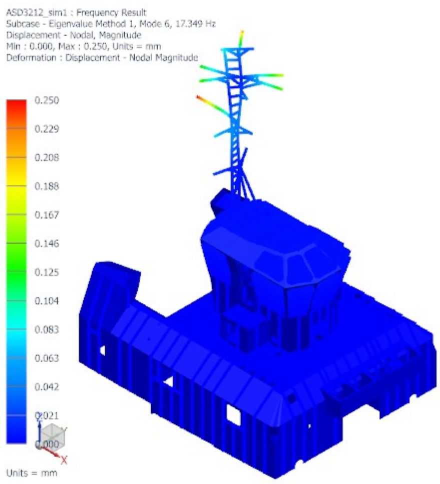
b) Mode 2 – 12.93 Hz



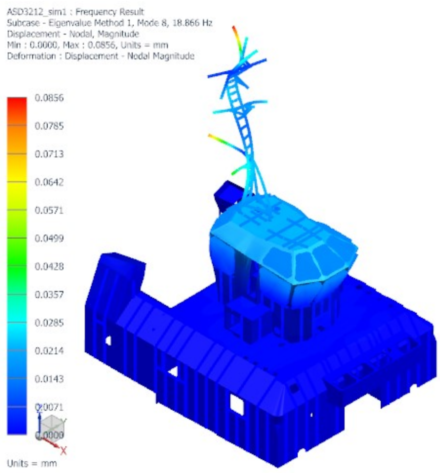
c) Mode 3 – 13.96 Hz



e) Mode 7 - 17.65 Hz



d) Mode 6 – 17.35 Hz



f) Mode 8 -18.87 Hz

Figure 11

4. PROPOSED MODIFICATIONS

In order to reduce the displacement of the bulkheads that occurs in Modes 3, 5, 7, some changes have been done. The proposed modifications are displayed in figure 12.

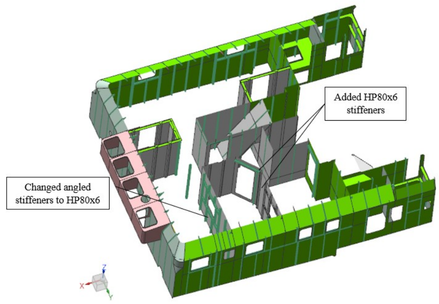


Figure 12

The optimized structure was analysed again obtaining different results. The results of both initial and updated analysis are summarized in Table 5 and 6.

Table 5

MODE	Initial results (case 1)		Updated results (case 2)	
	Disp [mm]	Freq [Hz]	Disp [mm]	Freq [Hz]
1	0.176	12.5	0.168	12.31
2	0.043	12.93	0.04	12.93
3	0.104	13.96	0.251	15.42
4	0.327	15.64	0.208	15.85
5	0.116	16.25	0.221	16.25
6	0.25	17.35	0.226	17.35
7	0.141	17.65	0.076	17.65
8	0.085	18.87	0.325	18.87

Table 6

MODE	Diferencies between cases		
	Disp [mm]	Freq [Hz]	How is case 1 compare with case 2
1	4.76%	1.54%	lower
2	7.50%	0.00%	lower
3	-58.57%	-9.47%	bigger
4	57.21%	-1.32%	bigger
5	-47.51%	0.00%	bigger
6	10.62%	0.00%	lower
7	85.53%	0.00%	lower
8	-73.85%	0.00%	bigger

Adding the stiffeners on the bulkhead leads to smaller displacement of the structure and the deformation of the bulkheads are not presented anymore. For some modes, there is a slight increase in the frequency range.

5. CONCLUDING REMARKS

Due to the fact tugs are the most utilized technical ships in each shipyard all over the world, it is important to take interest in their optimized design, as research and improvements could reduce the costs for the shipyard, improve the confort on board, optimize the power of each ship.Natural frequency analysis of the mast of ASD 3212 tug was performed. The lowest global mode of the mast is about 12 Hz and the higher mode is about 19 Hz. None of the global modes of the mast fall into shaft frequency range. None of the global modes fall into blade passing frequency range.

The results obtained indicate toward a local vibrations problem. The displacement results as shown in Table 5 and Table 6 allows us to highlight all critical areas. One of the structural solutions proposed is to add sttiffeners on the problematic bulkheads.

The vibration levels of the navigation mast and wheelhouse structures shall be also measured and verified during sea trials.

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