THE ANNALS OF "DUNAREA DE JOS" UNIVERSITY OF GALATI
FASCICLE XI – SHIPBUILDING. ISSN 1221-4620, e-ISSN 2668-3156
DOI: https://doi.org/10.35219/AnnUgalShipBuilding/2025.48.07
2025

THE ANALYSIS OF INDUCED VIBRATIONS BY PROPULSION SYSTEM EXCITATION OF AN 8000 TDW CHEMICAL TANKER, BY A FEM APPROACH

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ABSTRACT

Nowadays, the ships become larger and more flexible, requiring that their design be evaluated also according to the vibration criteria, being induced by the onboard propulsion system. The vibration onboard comfort addresses the conditions ensured for the humans, passengers, and crew, setting up the reference quality for living during navigation conditions. Also, excessive levels of vibrations can cause fatigue issues, damage structural elements, and also various ship's equipment during exploitation. This paper uses in study an existing operating vessel, for a preliminary vibration analysis stage, by using the FEM approach for the natural modes and the propulsion system-induced vibrations.

Keywords: induced vibrations, natural modes, FEM analysis, chemical tanker.

1. INTRODUCTION

This analysis involves the structural vibration response induced by the propulsion system's dual-shaft-line excitation of an 8000 tdw Chemical Tanker, based on a FEM approach, utilizing a structural model that incorporates the ship's hull and superstructure.

The analysis also involves a global modal analysis by the FEM approach for the chemical tanker maximum cargo loading in full case condition.

Case study is the "*Wappen von Hamburg Chemical Tanker*" vessel [6], [7], delivered by Damen Shipyards Galati in 2002.

Table 1 summarizes the main data for the vessel involved in the modal and induced vibration analyses and conform evaluation.

Table 1 Chemical tanker 8000 tdw main design characteristics [6],[7]

Characteristics	Values	Units
L_{OA} overall length	116.9	m
L_{PP} between perpendiculars ref. length	110.4	m
B max. breadth	18	m
H main deck height	9.4	m
T design draught	7.4	m
DWT deadweight	8150	t
GT tonnage	5095	-
v_s vessel speed	15	knots
Complement (crew)	11	no.

2. THE 8000 TDW CHEMICAL-TANKER FEM MODEL

As the ship study case is an already-built one, the FEM vessel model uses the information included in documents like: general arrangement, top, frame, and longitudinal views of the design drawings, shell expansion, arrangement in engine rooms, etc., provided by GLO Marine Galati Company in collaboration with DAMEN Galati Shipyard.

The FEM elements used are:

• *plate-type elements (quad shell)*, for hull panel shells, and primary longitudinal and transversal elements;

• *beam-type elements*, all ordinary longitudinal and transversal profile elements, using equivalent sectional properties for each profile type;

• *mass type elements*, for significant equipments, the propellers, the main engines, the mast navigation equipments, and others;

• *rigid type elements*, to ensure the link between base structure and mass elements, for master nodes, specific constraints definition.

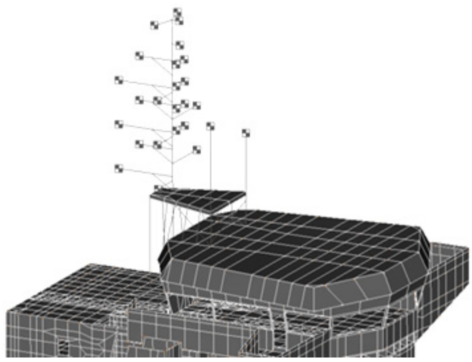

Fig. 1 Types of elements, FEM model of the chemical-tanker superstructure

To ensure the accuracy of the FEM model, special care is provided for the propulsion system interaction components' idealization (propeller, shaft, engine). The propellers are modeled as lumped mass elements at the shaft-lines' free end; the shaft-lines with variable sections, according to the technical drawing, are modeled as linked beam elements; the engine idealization is done with equivalent thickness shell elements, having a total mass of 250 t according to the engine data.

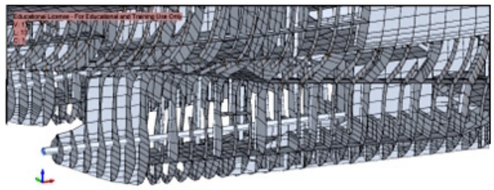

Fig. 2 Shaft arrangement-FEM model

For global vibration FEM analyses, the rules [3] recommend a typical coarse mesh type model, so we have selected for the vibration analysis an average mesh size of 700 mm, corresponding to the reference regular design distance.

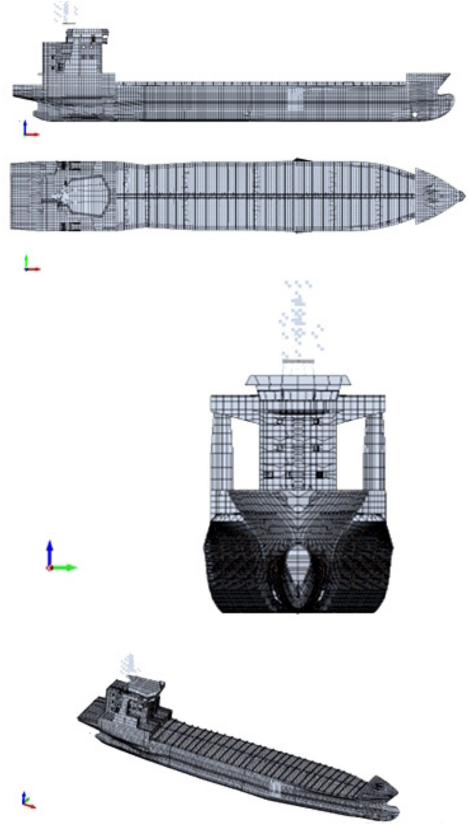

Fig. 3 Chemical tanker 8000 tdw, global FEM model views

The demand of adapting the computational resources to the induced vibration analysis extended running resources requirements, a partial FE model is selected from the global model (Fig.3), including the aft vessel part up the fore engine room bulkhead (Fig.4). So, the structural model covers the domain where influence of the propulsion system vibration is the most relevant on aft hull and superstructure.

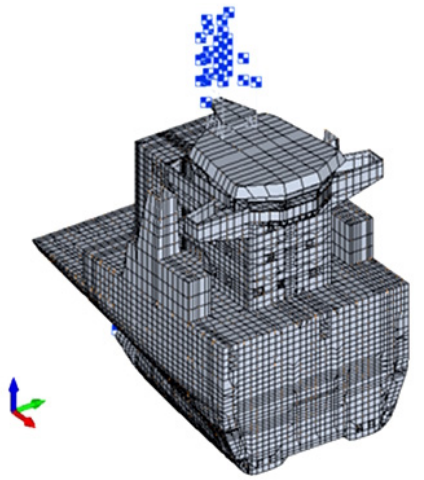

Fig. 4 Chemical tanker 8000 tdw, FEM ship aft and superstructure model

To achieve the scope of proper vibration analysis, the masses have to be specifically distributed on the FEM model. Besides the structural and equipment masses (defined as lightship), the cargo and added hydrodynamic (fluid effect) masses have been considered. Besides the lumped masses, also non-structural masses formulation is applied on the exterior aft bottom shell (Fig.6) and cargo tanks' bulkheads (Fig.5).

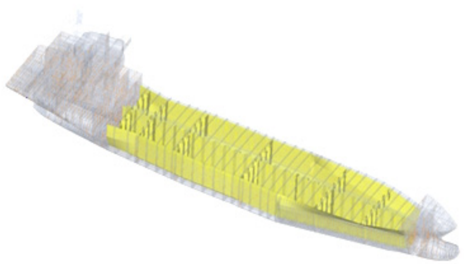

Fig. 5 Cargo tanks non-structural mass

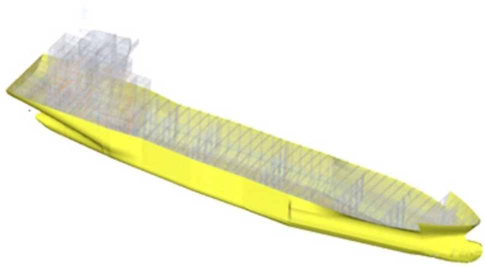

Fig. 6 Outer shell non-structural mass

3. THE 8000 TDW CHEMICAL-TANKER NATURAL VIBRATIONS ANALYSIS

A Normal modes/Eigenvalues analysis is defined in FEMAP [5]. For the modal analysis is requested a FEM fully developed model, side to side, aft to fore, with or without the effect of the added hydrodynamic masses. No constraints are necessary, using the shifting eigenvalue approach to avoid singularity. The following two modal analysis results:

- Dry analysis - no cargo or added hydrodynamic masses (lightship case);
- Hyd.& cargo analysis - with cargo and added hydrodynamic masses (full cargo case);

The modal analysis returned the following outputs for the hyd. & cargo modal analysis:

Table 2 Hyd. & cargo modal analysis results

No.	Frequency [Hz]	Global mode
1	1.04	vertical 1
2	1.59	horizontal 1
3	2.12	torsion 1
4	2.32	vertical 2
5	3.12	horizontal 2
6	3.79	vertical 3
7	3.84	torsion 2
8	4.59	horizontal 3
9	5.12	torsion 3
10	5.23	vertical 4
11	5.86	horizontal 4
12	6.31	vertical 5

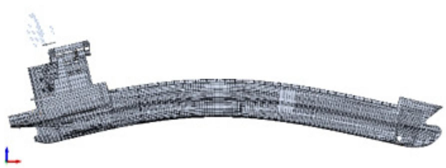

Fig. 7 FE model- Hyd. & cargo, Global vertical mode 1: 1.04 Hz

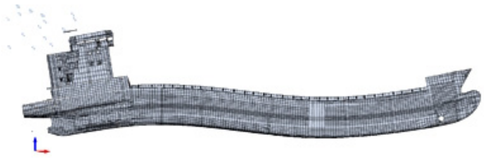

Fig. 8 FE model- Hyd. & cargo, Global vertical mode 2: 2.32 Hz

Fig. 9 FE model- Hyd. & cargo, Global horizontal mode 1: 1.59 Hz

Fig. 10 FE model- Hyd. & cargo, Global horizontal mode 2: 3.12 Hz

Fig. 11 FE model- Hyd. & cargo, Global horizontal mode 3: 4.59 Hz

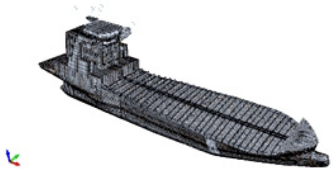

Fig. 12 FE model- Hyd. & cargo, Global torsion mode 1: 2.12 Hz

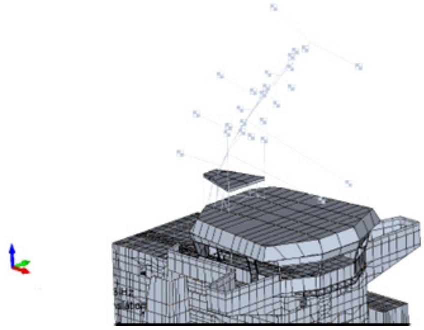

Fig. 13 FE model- Hyd. & cargo First local mast vibration mode: 2.61 Hz

The modal analysis returned the following outputs for the dry (lightship case) modal analysis:

Table 3 Dry modal analysis results

No.	Frequency [Hz]	Global mode
1	3.91	vertical 1
2	6.10	horizontal 1
3	6.26	torsion 1
4	8.05	vertical 2
5	10.68	torsion 2
6	11.57	horizontal 2

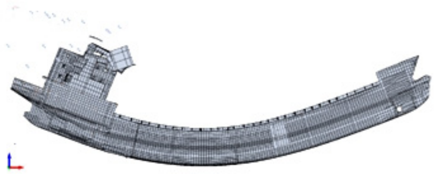

Fig. 14 FE model- Dry, Global vertical mode 1: 3.91 Hz

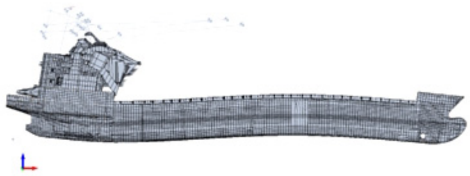

Fig. 15 FE model- Dry, Global vertical mode 2: 8.05 Hz

Fig. 16 FE model- Dry, Global vertical mode 3: 14.26 Hz

Fig. 17 FE model- Dry, Global horizontal mode 1: 6.10 Hz

Fig. 18 FE model- Dry, Global horizontal mode 2: 11.57 Hz

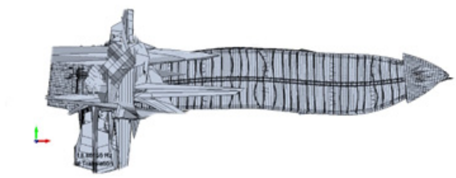

Fig. 19 FE model- Dry, Global horizontal mode 3: 18.86 Hz

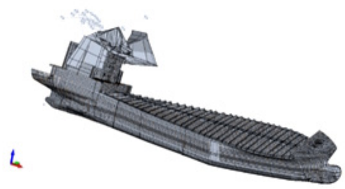

Fig. 20 FE model- Dry, Global torsion mode 1: 6.26 Hz

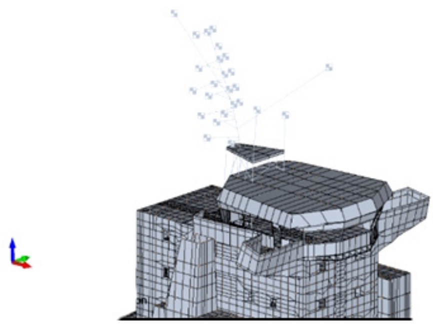

Fig. 21 FE model- Dry, Local mode of the main mast: 3.91 Hz

The hyd. & cargo analysis represents the real ship hull free vibration response. Comparing the two modal analysis cases, the dry analysis has significant higher frequencies, due to neglecting the cargo and added hydrodynamic (fluid effect) masses.

Table 4 Comparative natural frequencies with or without model cargo and added hydrodynamic masses, FEM

Global modes	Hyd & cargo case	Dry case	Diff.
	[Hz]	[Hz]	[-]
V	1.04	3.91	0.266
	2.32	8.05	0.288
H	1.59	6.10	0.261
	3.12	11.57	0.271
T	2.12	6.26	0.339
	3.84	10.68	0.360

Table 5 Chemical tanker global vibration and propeller reference frequencies

Global modes	[Hz]	Frequency range	[Hz]
propeller	-	1/3 frequency	0.76
vertical 1	1.04	-	-
propeller	-	1/2 frequency	1.13
horizontal 1	1.59	-	-
torsion 1	2.12	-	-
propeller	-	frequency	2.27
vertical 2	2.32	-	--
propeller	-	110% freq.	2.49
horizontal 2	3.12	-	-

From Table 5, results that direct resonance conditions can occur between the hull structure and the propeller, for the propeller frequency 2.27 Hz (136 RPM) on the vertical mode 2 (diff. 0.05 Hz) and for half (50%) of the propeller frequency on the vertical mode 1 (diff. 0.09 Hz), thus being the most sensitive operational conditions.

For preliminary vertical global frequency prediction, the statistical Kumai`s [3] expression can be used. It requests only the main vessel data, takes into account the cargo and added hydrodynamic masses, and is suitable for validation of the FEM vertical modal analysis approach.

Comparing the FEM vertical modal analysis frequencies with the statistical Kumai`s values, the differences are in the practical range, especially for the first vertical vibration mode (Table 6).

Table 6 FEM and Kumai comparison of the vertical natural global frequencies

FEM [Hz]	Kumai [Hz]	[%]
1.04	0.97	6.7%
2.32	1.94	16.5%

4. THE 8000 TDW CHEMICAL-TANKER INDUCED VIBRATIONS TIME DOMAIN ANALYSIS

In order to introduce constraints closer to reality on the local model, aft ship and superstructure, extracted from the global one, all the nodes of the engine room bulkhead are connected by a rigid element to a master control node. All displacements at the master node are restrained.

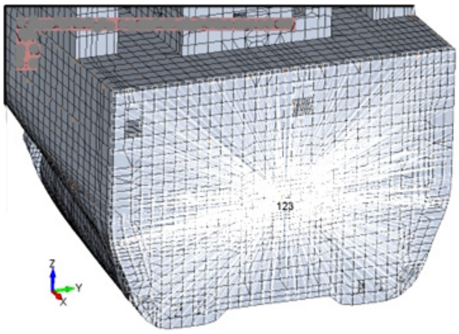

Fig. 22 Local, aft ship and superstructure, FEM model-constraints

For the propulsion system induced vibrations analysis, with no waves influence, the following loads on the vessel structure are considered for the time domain analysis:

- Propeller pressure on aft hull surface
- Forces induced in shaft-lines
- Still water hydrostatic pressure on aft hull shell, at design draught.

The propeller pressure acting on the hull shell is obtained by statistical rules based on, Holden approach ABS [3], useful for practical preliminary propeller-induced vibrations. The statistical Holden formulation includes both cavitating and non-cavitating pressure components, delivering the maximum design pressure reference value.

The propeller dynamic pressure (as a periodic load) is applied as normal pressure over the shell bottom elements of the partial aft and superstructure FEM structural model (Fig.4), around the propellers' domain.

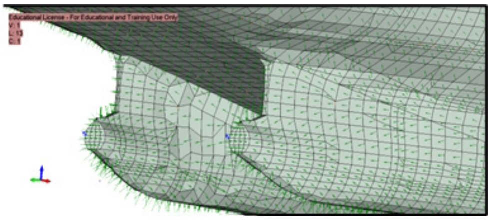

Fig.23 FE aft vessel model- total propeller impulse pressure

The shaft forces based on statistical data according to MAN [4].

For the FEM local model, the shaft loads are applied at the corresponding shaft position on the beam elements.

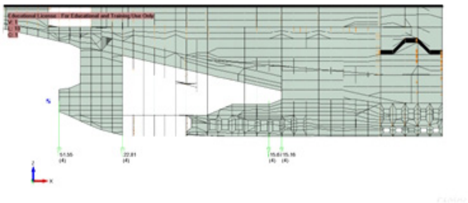

Fig.24 FE aft vessel model-shaft forces

As a loading condition, the still water at full cargo case (*T*=7.4m) is considered. The hydrostatic still water pressure is applied on external bottom shell with a specific user function in the FEM program [1],[2],[5].

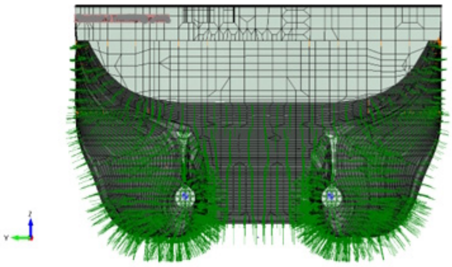

Fig.25 FE aft bottom model-hydrostatic pressure, in still water condition

Both pressures, the dynamic periodic component induced by the propellers and the hydrostatic still water component, are applied combined on the ship's aft bottom shell, using the normal pressure definition on the top and bottom plate faces.

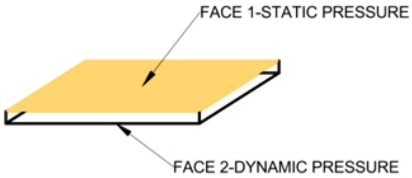

Fig.26 Pressure definition on the element

The dynamic loads are induced by the propellers, so that the excitation frequency is equal to the propeller one. To cover the transitional propeller operation, 4 reference frequencies are selected for the induced vibration analysis: 33% (0.76 Hz), 50% (1.13 Hz), 100% (2.27 Hz), 110% (2.49 Hz), from the referenced propeller frequency (136 rpm).

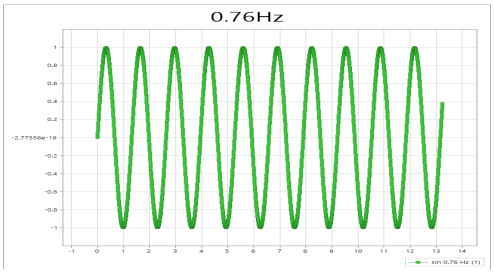

Fig.27 Time function 0.76 Hz (33%)

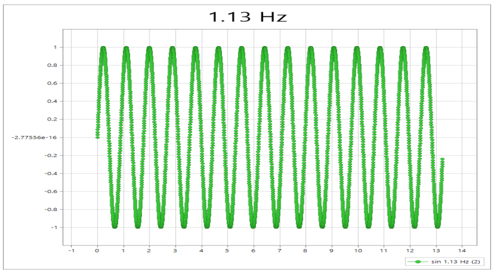

Fig.28 Time function 1.13 Hz (50%)

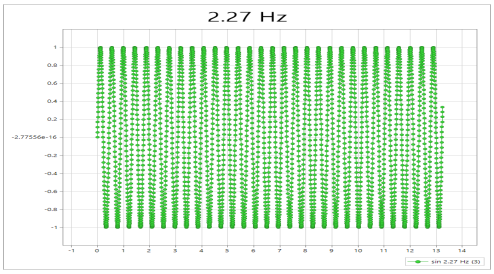

Fig.29 Time function 2.27 Hz (100%)

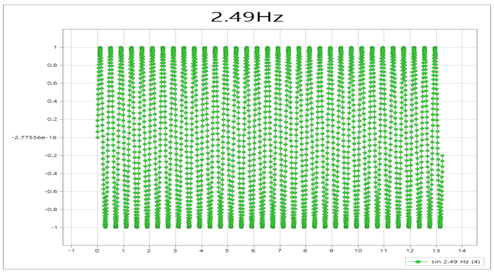

Fig.30 Time function 2.49 Hz (110%)

The dynamic pressure loads are considered periodic time records (Figs.27-30), corresponding to the reference 4 selected propeller frequencies, with the statistic values by rules ABS [3] for the pressure amplitude on the aft vessel bottom shell.

The induced vibration analysis solution is obtained by a direct time domain integration approach [2],[5], with a total of 3300 integration steps (0.004 s time step). The hydrodynamic and structural damping are very reduced on vibrations [3], so that for the numerical dynamic analysis, the limit condition without any damping is considered. The induced vibration response is assessed at 3 reference points on the FEM model (Figs. 31-33): near the shaft-line at the starboard engine room, above the engine room at the main deck, and in the superstructure at the wheelhouse.

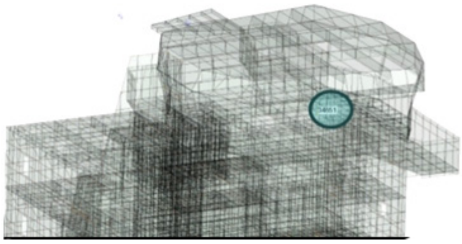

Fig.31 FEM superstructure model- wheelhouse reference point

Fig.32 FEM - main deck reference point

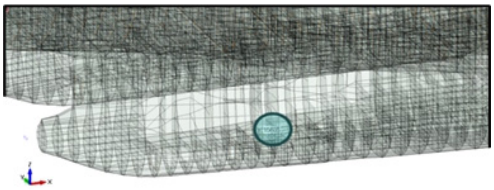

Fig.33 FEM model- engine room at starboard reference point

The dynamic response maximum amplitudes at the reference points are directly compared to the *RMS* (most probable response) weighted overall frequency range (0-80 Hz) limit values, according to the rules "*Criteria for Crew and Passenger Relating to Mechanical Vibration-ISO 6954/2000*" [3], to evaluate the vibration comfort onboard for the crew working near to the propulsion system.

For the selected 3 reference points, in Tables 7-18, the assessment by the rules criteria [3] of the maximum vibration response amplitudes is presented.

Table 7 Wheelhouse point, 0.76 Hz (33%)

Criteria	FEM	Adm. limit	Adverse limit
w [mm]	0.01	3	7
v [mm/s]	0.15	11	13.5
a [mm/s^2]	7.52	150	300
σ [MPa]	1.31	235	-

Table 8 Engine room point, 0.76 Hz (33%)

Criteria	FEM	Adm. limit	Adverse limit
w [mm]	0.04	3	7
v [mm/s]	0.39	11	13.5
a [mm/s^2]	9.34	150	300
σ [MPa]	19.2	235	-

Table 9 Main deck point, 0.76 Hz (33%)

Criteria	FEM	Adm. limit	Adverse limit
w [mm]	0.01	3	7
v [mm/s]	0.10	11	13.5
a [mm/s^2]	3.64	150	300
σ [MPa]	9.36	235	-

Table 10 Wheelhouse point, 1.13 Hz (50%)

Criteria	FEM	Adm. limit	Adverse limit
w [mm]	0.02	3	7
v [mm/s]	0.24	11	13.5
a [mm/s^2]	11.1	150	300
σ [MPa]	1.3	235	-

Table 11 Engine room point, 1.13 Hz (50%)

Criteria	FEM	Adm. limit	Adverse limit
w [mm]	0.05	3	7
v [mm/s]	0.66	11	13.5
a [mm/s^2]	15.2	150	300
σ [MPa]	19.3	235	-

Table 12 Main deck point, 1.13 Hz (50%)

Criteria	FEM	Adm. limit	Adverse limit
w [mm]	0.10	3	7
v [mm/s]	0.15	11	13.5
a [mm/s^2]	5.48	150	300
σ [MPa]	9.41	235	-

Table 13 Wheelhouse point, 2.27 Hz (100%)

Criteria	FEM	Adm. limit	Adverse limit
w [mm]	0.10	3	7
v [mm/s]	1.64	11	13.5
a [mm/s^2]	40.5	150	300
σ [MPa]	1.31	235	-

Table 14 Engine room point, 2.27 Hz (100%)

Criteria	FEM	Adm. limit	Adverse limit
w [mm]	0.28	3	7
v [mm/s]	4.50	11	13.5
a [mm/s^2]	75.5	150	300
σ [MPa]	19.2	235	-

Table 15 Main deck point, 2.27 Hz (100%)

Criteria	FEM	Adm. limit	Adverse limit
w [mm]	0.06	3	7
v [mm/s]	0.98	11	13.5
a [mm/s^2]	20.5	150	300
σ [MPa]	9.46	235	-

Table 16 Wheelhouse point, 2.49 Hz (110%)

Criteria	FEM	Adm. limit	Adverse limit
w [mm]	0.58	3	7
v [mm/s]	3.75	11	13.5
a [mm/s^2]	70.4	150	300
σ [MPa]	1.63	235	-

Table 17 Engine room point, 2.49 Hz (110%)

Criteria	FEM	Adm. limit	Adverse limit
w [mm]	1.90	3	7
v [mm/s]	10.3	11	13.5
a [mm/s^2]	172	150	300
σ [MPa]	19.5	235	-

Table 18 Main deck point, 2.49 Hz (110%)

Criteria	FEM	Adm. limit	Adverse limit
w [mm]	0.28	3	7
v [mm/s]	2.21	11	13.5
a [mm/s^2]	40.7	150	300
σ [MPa]	10.5	235	-

In the case of excitation frequency below or equal to the propeller frequency (136 rpm) (Tables 7-15), the vibration response, displacement, velocity, and acceleration, satisfy the rules admissible criteria [3], with the minim safety level of 1.98 at the engine room reference point for the acceleration criterion ($75.5 < 150$ mm/s^2).

In the extreme case with 10% higher excitation frequency (Tables 16-18) (a hypothetical case), the vibration maximum response has increased. The significant values are recorded at the engine room reference point, where the displacement and velocity criteria become close to the admissible limits, and the acceleration criterion is no longer satisfied ($150 < 172$ mm/s^2). On the acceleration criterion, the safety factor is $0.87 < 1$, making it possible to record vibration adverse comments concerning the crew comfort working in the engine room.

For the excitation frequencies range (33%-110%), the yielding stress limit criterion is very well satisfied in still water navigation conditions.

5. CONCLUSIONS

The modal analysis of the 8000 tdw chemical tanker has included an extended range of global modes (vertical, horizontal, and torsional, Fig.7-21). These points out the significant influence of the inclusion of the cargo and added hydrodynamic masses for the numerical model, otherwise the resulting natural frequencies would be several times higher (Table 4) and not realistic.

The vibration resonance condition assessment, between the hull global modes and the propeller reference frequencies, for a range between 0.5 to 3 Hz, has proven that resonance occurs for 2.27 Hz and 1.13 Hz (50%) propeller frequencies, in the domain of global vibration vertical modes 1 and 2.

The propulsion system induced vibration analysis results have revealed that for the excitation below or equal to the propeller frequency (136 rpm) all the vibration limit criteria [3] for the crew are satisfied (Tables 7-15). If the excitation exceeds by 10% the propeller frequency, the dynamic response amplitudes are increase, leading to exceed the safety limits on the acceleration criterion at the engine room (Tables 16-18), and adverse comments can occur. The stress criterion is safe for the excitation frequencies range.

A comparative analysis of the maximum dynamic responses (Tables 7-18) points out that as the position of the reading points is increasing in reference to the shaft-line, the recorded values are reducing, with the minimum effect at the wheelhouse and maximum at the engine room.

These results prove that for the still water navigation conditions, for the rules-based propulsion loads [3], without exceeding the propeller frequency, the crew onboard comfort is ensured.

Further studies shall include on-site measured vibration excitations, the influence of the structural damping, and the wave navigation scenarios, for an extended induced vibration analysis of the 8000 tdw chemical tanker.

Acknowledgments

We thank very much to GLO Marine Galati in collaboration with DAMEN Galati Shipyard, for providing us with the 8000 tdw chemical tanker technical data. This research was developed for academic purposes at the Naval Architecture Research Center, Dunarea de Jos University of Galati.

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Paper received on October 7th, 2025

