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ORIGINAL ARTICLE

Free and forced vibration analyses of ship structures using the finite element method

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Abstract With increases in ship size and speed, shipboard vibration becomes a significant concern in the design and construction of vessels. Excessive ship vibration is to be avoided for passenger comfort and crew habitability. In addition to the undesired effects on humans, excessive ship vibration may result in the fatigue failure of local structural members or malfunctioning of machinery and equipment. The propeller induces fluctuating pressure on the surface of the hull, which induces vibration in the hull structure. These pressure pulses acting on the ship hull surface above the propeller are the predominant factor for vibrations of ship structures are taken as excitation forces for forced vibration analysis. Ship structures are complex and may be analyzed after idealization of the structure. Several simplifying assumptions are made in the finite element idealization of the hull structure. In this study, a three-dimensional finite element model representing the entire ship hull, including the deckhouse and machinery propulsion system, has been developed using solid modeling software for local and global vibration analyses. Vibration analyses have been conducted under two conditions: free–free (dry) and in-water (wet). The wet analysis has been implemented using acoustic elements. The total damping associated with overall ship hull structure vibration has been considered as a combination of the several damping components. As a result of the global ship free vibration analysis, global natural frequencies and mode shapes have been determined. Moreover, the responses of local ship structures have been determined as a result of the propeller-induced forced vibration analysis.

Keywords Finite element method · Ship hull vibrations · Modal analysis

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1 Introduction

Finite element analysis is universally recognized as the most important technological breakthrough in the field of structural engineering analysis. The development of computers elevated the finite element method to one of the most popular techniques for solving engineering problems. For analyzing a complicated structure such as a ship hull, the finite element method is the only tool that yields satisfactory results.

With increases in ship sizes and speeds, shipboard vibration becomes a significant concern in the design and construction of vessels. The increase in the dimensions of merchant ships and the outputs of their propulsion systems since the end of the Second World War have caused numerous technical problems for shipbuilders; increased ship vibrations are one such problem. Increases in the size and output of the propulsion systems of ships are also the cause of complex problems, especially those involving vibrations [1]. Excessive ship vibration is to be avoided for passenger comfort and crew habitability. In addition to its undesired effects on humans, excessive ship vibration may result in the fatigue failure of local structural members or the malfunctioning of machinery and equipment [2]. It is clear that if the vibration problems, which have repeatedly been identified by experience as the most important problems, are addressed at the earliest design stage, ultimately, serious problems involving significant correction effort costs may be avoided. The focus is on planning for vibrations early in the conceptual design stage when the details have not been developed. If as much planning as possible can be performed in the conceptual design stage with the simple tools and rules of thumb available at that level of design, major vibration problems can be avoided. Major potential problems may often be present in the crude concept design definition. Just identifying and addressing those potential problems in terms of the minimal technology available at the concept design stage is considered very important to the success of ship design. The design and construction of a ship free of excessive vibration continue to be major concerns, and,as such, it is prudent to investigate, through analysis, the likelihood of vibration problems early in the design stage.

Vibration analysis is aimed at the confirmation of the many design considerations associated with stern configuration, the main propulsion machinery, the propeller/shafting system and the location and configuration of major structural assemblies. The ship hull structure includes the outer shell plating and all internal members, which collectively provide the necessary strength to satisfactorily perform the design functions in the expected sea environment. The hull structure responds as a free–free beam (both ends free) when subjected to dynamic loads. The vibration induced by the propulsion system is a common source of ship vibration. The vibration from this source manifests itself in several ways. Dynamic forces from the shafting system are transmitted to the hull through shaft bearings. The propeller induces fluctuating pressures on the surface of the hull, which induce vibrations in the hull structure. The main and auxiliary engines can directly cause vibrations through dynamic forces transmitted through their supports and foundations. The response to this forcing can cause vibrations in the hull girder, deckhouse, deck and other structures, local structures and equipment. When attempting to determine the source of vibration, it is necessary to establish the frequency of excitation and to relate the frequency of excitation to the shaft rotational frequency by determining the number of oscillations per shaft revolution.

The four elements of importance with regard to ship vibration are “excitation”, “stiffness”, “frequency ratio” and “damping”. In propeller-induced ship vibrations, the excitation may be reduced by changing the propeller’s unsteady hydrodynamics. This may involve lines or clearance changes to reduce the nonuniformity of the wake inflow or may involve geometric changes to the propeller itself. Stiffness is defined as the spring force per unit deflection. In general, stiffness increases rather than decreases when variations in the natural frequency are accomplished by variations in stiffness. Reducing system stiffness in an attempt to reduce vibration is not a recommended practice. At resonance, the excitation is opposed only by damping. Note that ω/ω_n can be varied by varying either the excitation frequency ω or the natural frequency ω_n. The spectrum of ω can be changed by changing the RPM of a relevant rotating machinery source, or in the case of propeller-induced vibration, by changing the propeller’s RPM or number of blades. The natural frequency ω_n is changed by changes in the system mass and/or stiffness; increasing the stiffness is the usual and preferred approach. The damping coefficient of structural systems in general, and of ships in particular, is small; $\zeta \ll 1$. Therefore, except very near resonance, the vibratory amplitude is approximately damping independent. Furthermore, damping is difficult to increase significantly in systems such as ships; ζ is, in general, the least effective of the four parameters available to the designer for implementing changes in ship vibratory characteristics. Four elements were previously identified as being influential in determining a ship’s vibratory response, and their relationship to vibration reduction was addressed. While quantification of all four elements is required to calculate the vibration response level, acceptable results may consistently be achieved with a reasonable amount of effort by focusing the concept design on two of the four elements. The two elements of importance are the “excitation amplitude” and the “frequency ratio”.

Two design approaches are used in ship design: “overcritical” and “undercritical” design methods. The overcritical design method refers to the condition at which the frequency of the main harmonic excitation is higher than the natural frequency. Conversely, the undercritical design method refers to the condition at which the frequency of the main harmonic excitation is lower than the natural frequency. Generally, the “undercritical” design method is preferred. In this study, global ship hull free vibration problems have been studied under two conditions, which are free–free (dry) and in-water (wet) using finite element analysis. The propeller-induced fluctuating pressures on the surface of the hull, which induces vibration in the hull structure, have been taken as the main excitation source for the forced vibration analysis.

2 Global ship hull model

For this study, an 18000 DWT chemical tanker named “PROCIDA”, which has been built in the ADIK Shipyard in Tuzla-Istanbul, has been selected for modeling (Fig. 1).

The main properties of the selected ship are listed in Table 1.

Ship hull structures are complex and may be analyzed after idealization of the structure. Several simplifying assumptions are made in the finite element idealization of

Fig. 1 Modeled ship

Table 1 Properties of the modeled ship

Property	Value
Name	Procida
Code	CT75
Type	Chemical tanker
Tonnage	18000 DWT
Length	149.108 m
Width	22.399 m
Fore draft	0.379 m
Aft draft	4.280 m
Mean draft	2.329 m
Trim	3.901 m
Center of gravity (transverse)	0.034 m
Center of gravity (vertical)	9.560 m
Center of gravity (axial)	59.469 m
Origin	Rudder axis
Service speed	14.5 knots
Propeller RPM	173
Propeller blade number	4
Main engine	MAN 8S35MC

the hull structure. The modeling requirements are that all significant structural sections are to be captured and deflection/velocity/acceleration are to be sufficiently predicted. A three-dimensional model representing the entire ship hull, including the deckhouse and the machinery propulsion system, needs to be developed for the vibration analysis. In addition to the hull structure, frames, panels, plates, beams, bulkheads and railings should also be modeled. If a global model exists from any previous tasks, such as a stress analysis, it needs to be conditioned for the vibration analysis. Mass distribution is the most important factor in vibration analysis. All heavy equipment, such as the main and auxiliary engines, are modeled using mass elements.

The objective of the vibration analysis is to investigate the ship vibration performance at the intended service conditions. Therefore, the loading conditions, such as the "full load condition" and the "ballast condition", in which the ship operates at the ship design speed, will be the focus of the vibration analysis. Using the two-dimensional technical drawings of the ship, the three-dimensional finite element model is prepared. The model consists of the main parts, including the hull, main deck, forecastle, stern, frames, cargo tanks, girders, superstructure and main engine room. The main hull structure consists of 48 panels called frames. The ship hull has been designed as a double-hull in which the outer sides of the frames form the outer plating and the inner sides of the frames form the inner plating of the hull. The frames are connected internally using beams called girders. The inner plating also forms the boundary of the cargo tanks. The cargo tanks are separated by high corrugated plates. In total, there are 12 cargo tanks in the ship. The main engine foundation has also been modeled in the forecastle. The thickness of the outer plating varies from 10 to 14 mm. Detailed global and local images of the three-dimensional ship hull model are shown in Figs. 2, 3, 4, 5, 6.

3 Global finite element model

In this section, the three-dimensional ship hull model constructed in the previous section has been imported to the finite element software ABAQUS to make the finite element model and assign the values for the vibration (modal) analysis. Quadrilateral elements with an average size of 500 mm have been used in the finite element modeling.

The finite element used in this model is named as S4 which is a quadrilateral general purpose finite element with finite strains. It is also defined as a 4-node doubly curved general-purpose shell element. The reason why S4 is called a doubly curved shell element is because each of nodes at the corner of S4 can have different shell normals which also is interpolated using the same shape function as displacement. This element is part of the commercial software ABAQUS and is based on a thick shell theory. They serve as general-purpose shell elements in the ABAQUS element library. The shell formulation considered is that of finite-

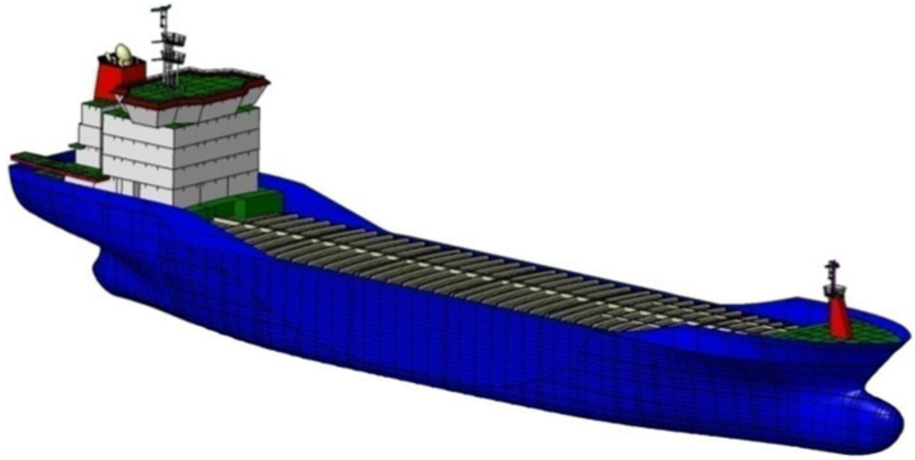


Fig. 2 Global view

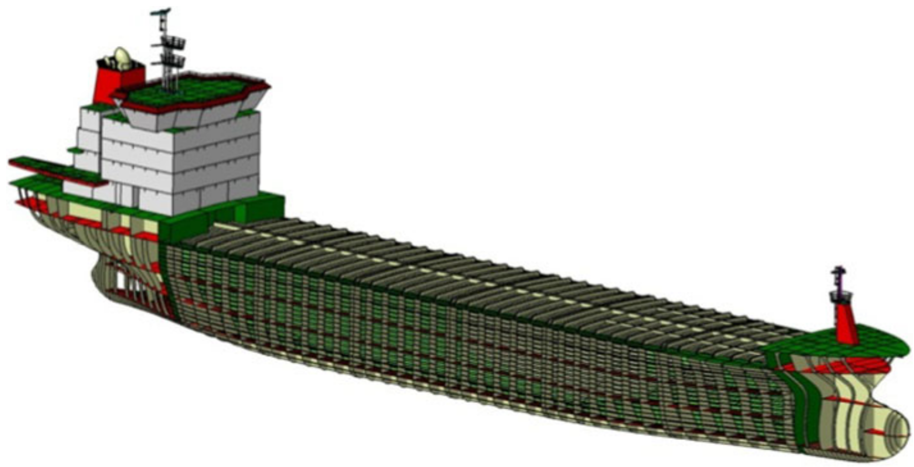


Fig. 3 Double hull

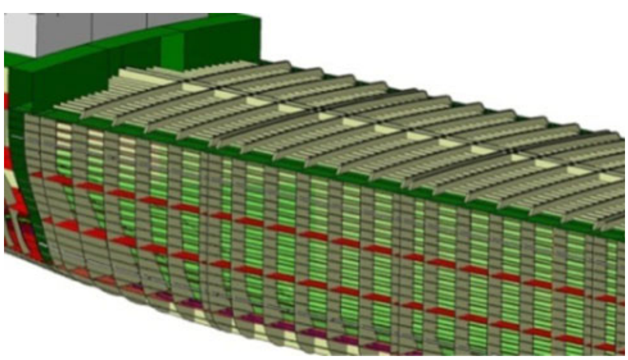


Fig. 4 Girders

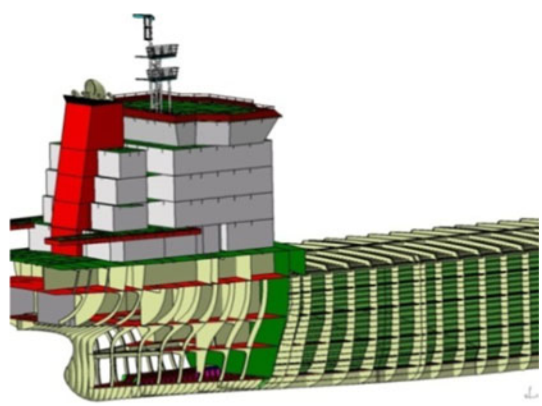


Fig. 5 Aft body

membrane strain, therefore, these elements can be used to perform large strain analyses. They are widely used for industrial applications because they are suitable for both thin and thick shells. The S4 element uses a normal integration rule with four integration points. The assumed strains approach is employed to prevent shear and membrane locking. The S4R element uses a reduced integration rule with one integration point that makes this element computationally less expensive than S4. For S4R, the assumed strains method is modified, so that a one point integration scheme plus hourglass stabilization is obtained. Hourglass modes, a form of artificial mechanisms, can arise from the use of the reduced integration rule. The hourglass stabilization is performed through an hourglass

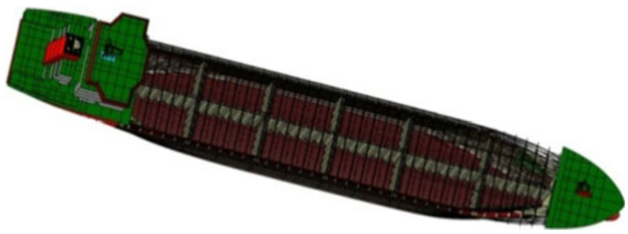


Fig. 6 Cargo tanks

control parameter. The S3 element is obtained through the degeneration of the S4 element. The ABAQUS shell library also includes the general purpose S3R element. This element is equivalent to S3, yielding identical results to those of S3.

Some sections of the global ship hull finite element model and the mesh structure have been shown in Figs. 7, 8, 9, 10, 11.

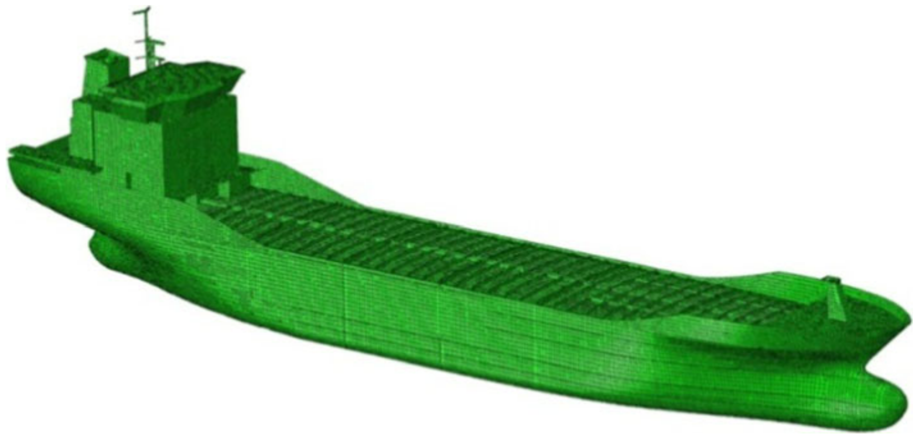


Fig. 7 Global ship hull finite element model

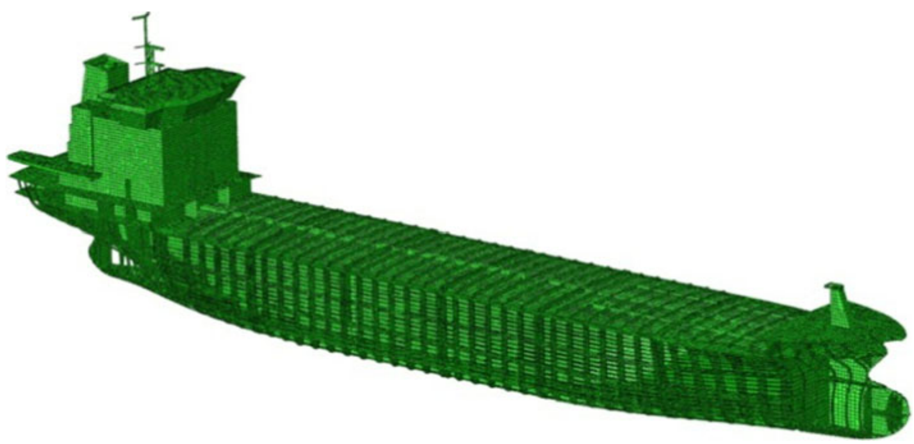


Fig. 8 Global inner mesh

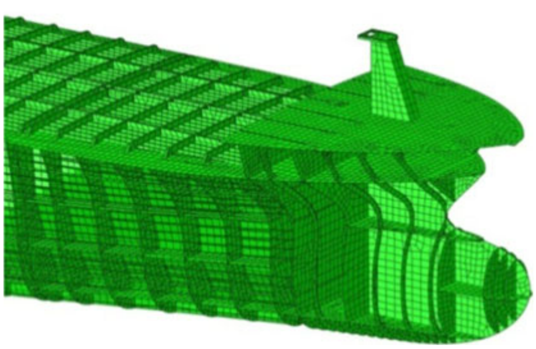


Fig. 9 Forecastle mesh

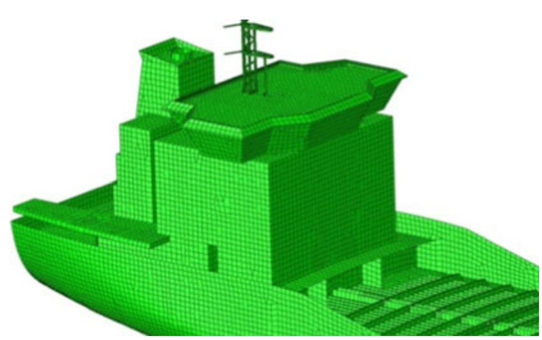


Fig. 10 Superstructure mesh

4 Global ship vibration analysis

A free vibration analysis has been conducted under two conditions, which are free–free (dry) and in-water (wet). In both of the conditions, the global natural frequencies and mode shapes of the ship structure have been determined. The Block-Lanczos algorithm has been used for solving eigenvalues. Both in free and forced vibration analyses, the procedure shown in Fig. 12 has been carried out.

Figure 12 explains the general ship vibration analysis procedure. It starts with the modelling of the ship and continues with the modelling of stability and trim conditions. Then determining the loading conditions, the whole finite element model is created. Two main analyses are conducted on this model which are free and forced vibration analyses. In the free vibration analysis section, natural frequencies and modes shapes are determined. In the other section, where forced vibration analysis is conducted, excitation forces and frequencies are determined and the response values obtained by this analysis are checked

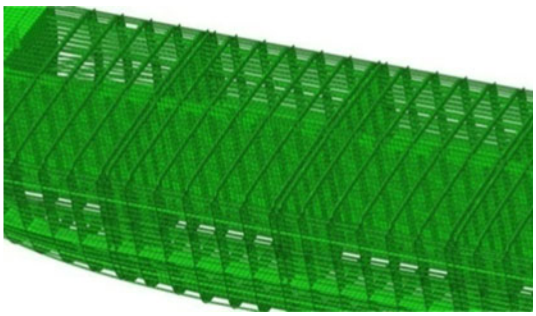


Fig. 11 Girders

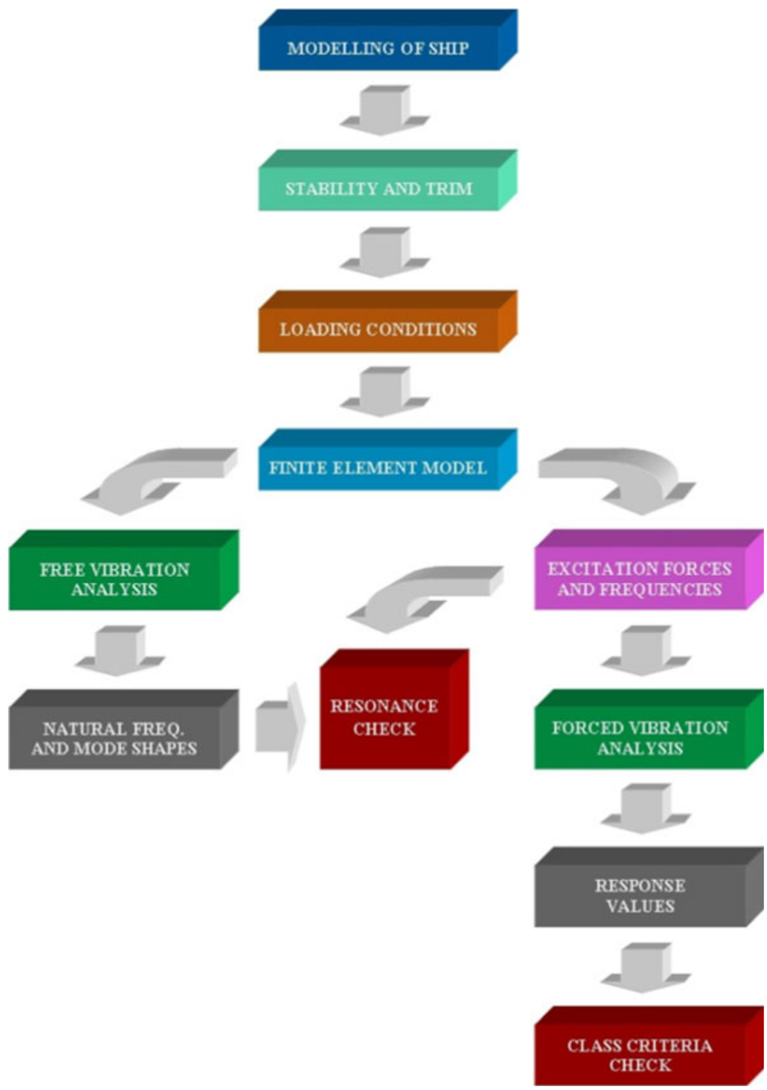


Fig. 12 Ship vibration analysis procedure

according to the values given by the Class (ship classification institution).

5 Dry (free–free) free vibration analysis

In this section, the natural frequencies and the mode shapes of the ship structure obtained in the dry mode are presented. The analysis values are listed in Table 2, and the finite element model of the girders used in the analysis are shown in Fig. 13.

These analysis values are automatically taken from the solver during the analysis. As you see, in both wet and dry analyses, the number of nodes and the number of user-defined nodes are the same. The long name of the "number of user-defined nodes" is the "number of nodes defined by the user" which means the number of nodes created by the user with meshing. Two values are equal to each other. This means that no nodes with contact pairs etc., are automatically created by the software. The same is valid for the number user-defined elements in wet analysis. "Number of Variables" is the total number of degrees of freedom plus any Lagrange multiplier variables.

The computer used for the calculation in this study has a pentium processor with 4 cores and 16 GB of memory which can be named as a standard workstation.

The first eight natural frequencies are listed in Table 3, and the first eight mode shapes following the rigid body modes are shown in Fig. 14.

Table 2 Finite element analysis values

Number of elements	185,029
Number of nodes	166,159
Number of variables	996,954
Total analysis time	69,071 s

6 Wet (in-water) free vibration analysis

The exact global ship hull mesh from the previous analysis has been used in in the wet analysis, and the sea water has been modeled using linear four-noded acoustic elements. Inadequate mesh refinement is the most common source of difficulties in acoustic and vibration analyses. For reasonable accuracy, at least six representative internodal intervals of the acoustic mesh should fit into the shortest acoustic wavelength present in the analysis. The accuracy improves substantially if eight or more internodal intervals are used at the shortest wavelength. In transient analyses, the shortest wavelength present is difficult to determine before an analysis: it is reasonable to estimate this wavelength using the highest frequency present in the loads or prescribed boundary conditions. An "internodal interval" is defined as the distance from a node to its nearest neighbor node in an element; that is, the element size for a linear element or half of the element size for a quadratic element. At a fixed internodal interval, quadratic elements are more accurate than linear elements. The acoustic wavelength decreases with increasing frequency, so there is an upper frequency limit for a given mesh. Let $L_{\max}$ represent the maximum internodal interval of an element in a mesh, let $n_{\min}$ represent the number of internodal intervals per acoustic wavelength, let $f_{\max}$ represent the frequency of excitation, and

$$c_{\mathrm{f}} = \sqrt{\frac{k_{\mathrm{f}}}{\rho_{\mathrm{f}}}} \tag{1}$$

represent the speed of sound. The requirement for maximum linear element length can then be expressed as:

$$L_{\max} \leq \frac{c_{\mathrm{f}}}{n_{\min} \cdot f_{\max}}. \tag{2}$$

The speed of sound in the fluid can be found by setting $K_{\mathrm{f}} = 2306.35$ MPa and $\rho_{\mathrm{f}} = (1.025) \times (10^{-9})$ tons/mm^3, which are commonly used values for sea water.

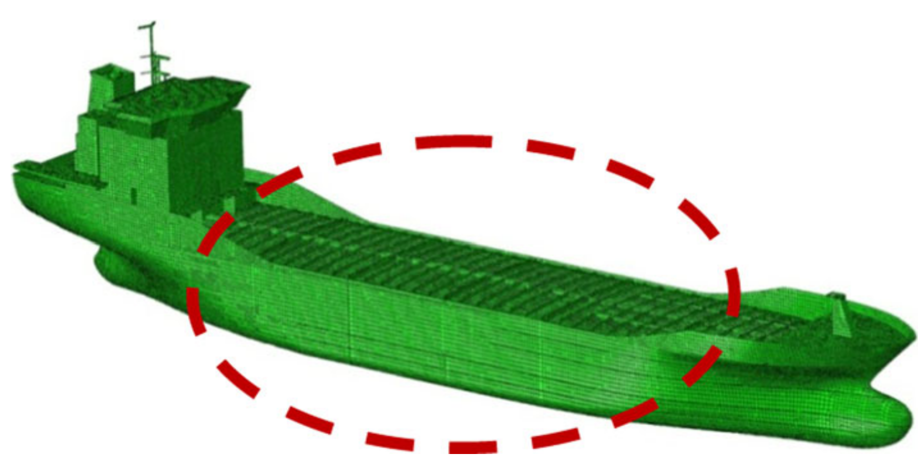


Fig. 13 Girders finite element model

Table 3 Global dry natural frequencies

Mode	Mode type (node number)	Natural frequency (Hz)
1	Vertical bending mode (2)	3.3549
2	Horizontal bending (2) + torsional mode (1)	5.4409
3	Vertical bending mode (3)	6.8837
4	Horizontal bending (3) + torsional mode (2)	11.175
5	Vertical bending mode (4)	11.314
6	Torsional mode (2)	12.056
7	Vertical bending mode (4)	14.009
8	Torsional mode (3)	16.389

$$c_f = 1500032.52 \text{ mm/s} \approx 1500 \text{ m/s}$$

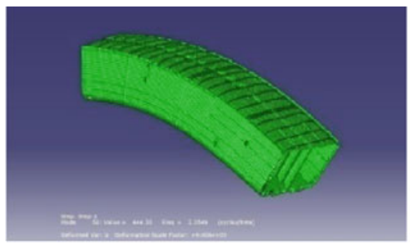
Setting $n_{\min} = 8$, the maximum linear element length that can be simulated accurately can be found:

$$L_{\max} \leq \frac{1500}{8.40}$$

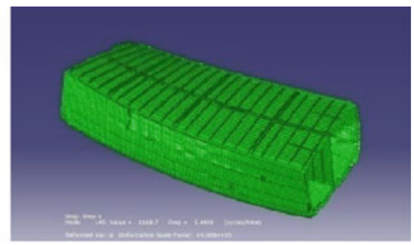
$$L_{\max} \leq 4.6875 \text{ m.}$$

In this analysis, the element length of the acoustic elements has been defined as 600 mm, which is far below the calculated $L_{\max}$. The acoustic element used in this wet analysis is called AC3D4 in ABAQUS which is a 4-node linear acoustic tetrahedral element. Analysis values are listed in Table 4.

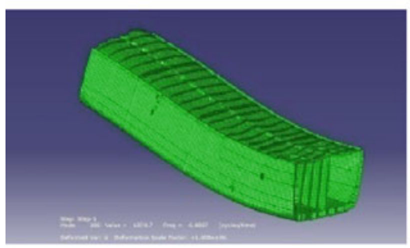
Fig. 14 Global dry mode shapes



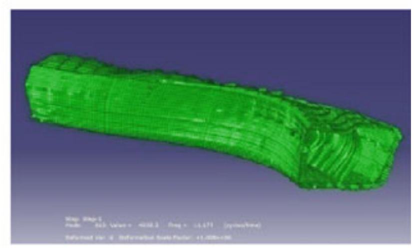
(a) Mode 1



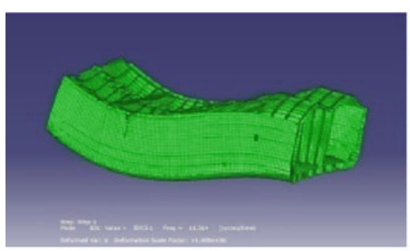
(b) Mode 2



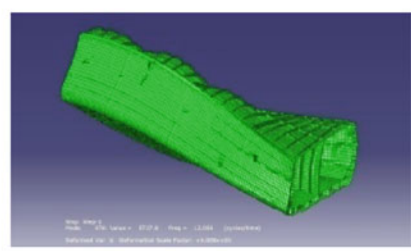
(c) Mode 3



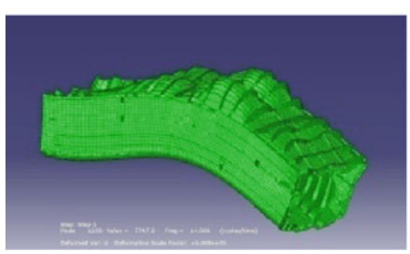
(d) Mode 4



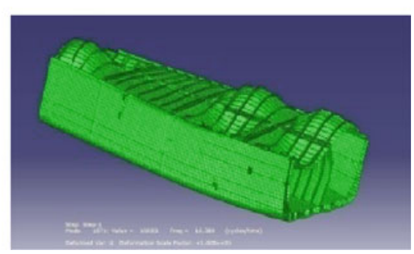
(e) Mode 5



(f) Mode 6



(g) Mode 7



(h) Mode 8

Table 4 Finite element analysis values

Number of elements	970,452
Number of user-defined elements	970,562
Number of nodes	319,497
Number of variables	1,610,306
Total analysis time	202,309 s

The global wet finite element model is shown in Fig. 15. The profile used in Fig. 15 is the general profile recommended by the regulations of ship classification institutions for global wet analyses of submerged structures which is a half cylinder with two quarter spheres at the ends.

The first six rigid body modes are presented in Fig. 16.

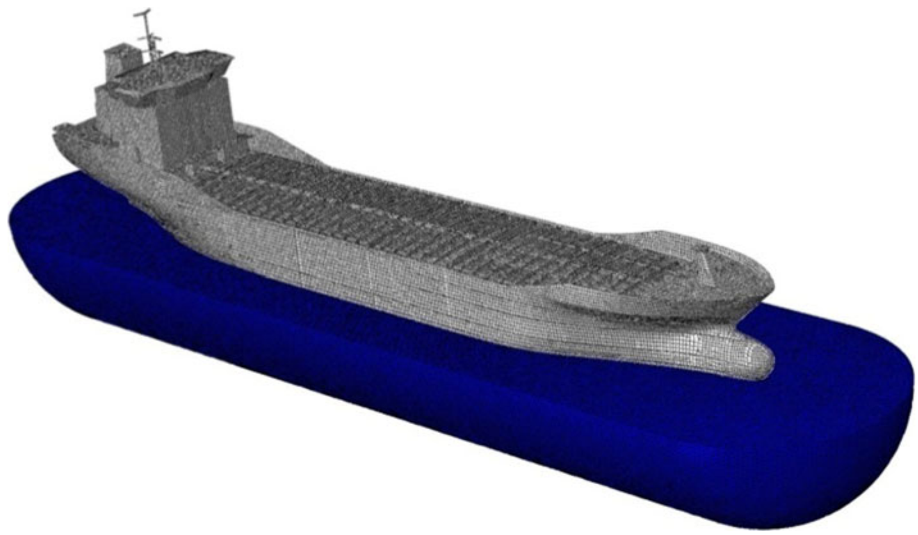


Fig. 15 Global wet finite element model

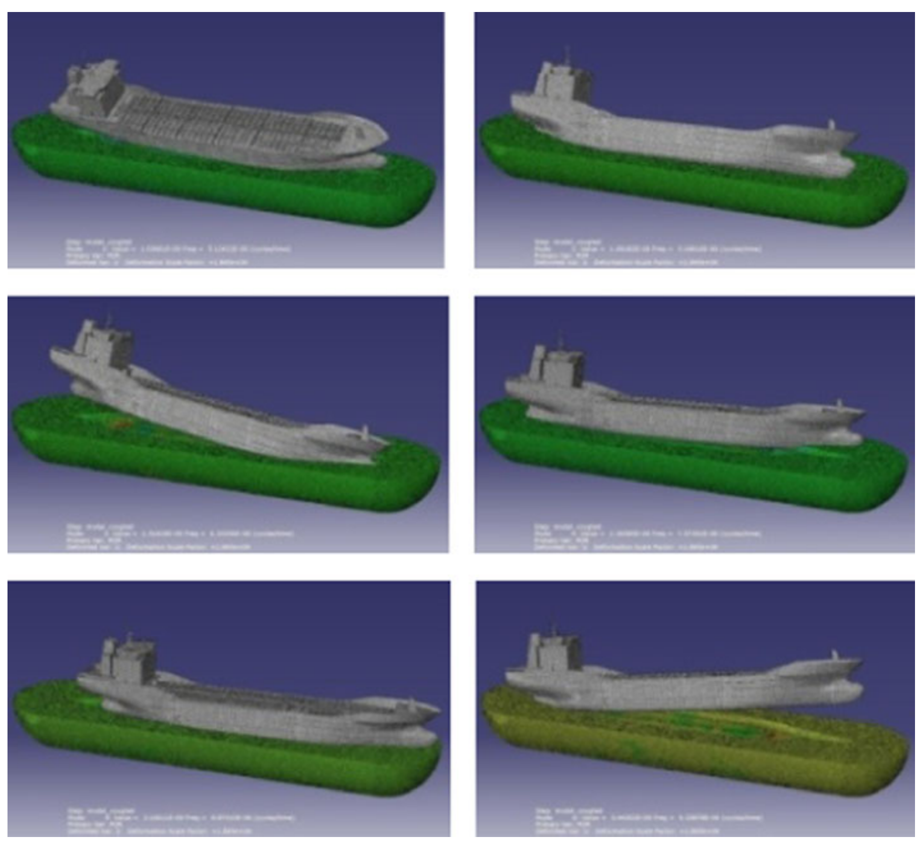


Fig. 16 Rigid body modes

Table 5 Global wet natural frequencies

Mode	Mode type (node number)	Natural frequency (Hz)
1	Vertical bending mode (2)	1.9169
2	Vertical bending mode (3)	3.7458
3	Horizontal bending (2) + torsional mode (1)	4.8353
4	Vertical bending mode (4)	5.2817
5	Horizontal bending (3) + torsional mode (1)	5.4145
6	Vertical bending mode (4)	5.6903
7	Horizontal bending (2) + torsional mode (2)	5.7154
8	Torsional mode (2)	6.7115
9	Vertical bending mode (5)	6.8863
10	Vertical bending mode (5)	7.6108
11	Horizontal bending (3) + torsional mode (2)	8.0828
12	Longitudinal mode	14.670

The first twelve natural frequencies are listed in Table 5, and the first twelve mode shapes following the rigid body modes are shown in Fig. 17.

7 Empirical free vibration analysis of a ship

The natural frequencies corresponding to the two-noded vertical bending modes of conventional ship hulls can be estimated with reasonable accuracy using Kumai's [3] formula:

$$N_2 = (3.07) \times 10^6 \sqrt{\frac{I}{\Delta_i L^3}} \tag{3}$$

$$\Delta_i = \left((1.2) + \frac{1}{3}\frac{B}{T_{\mathrm{m}}} \right) \Delta \tag{4}$$

where N_2, natural freq. of two-noded vertical bending mode (rpm); I, moment of inertia of the cross section (m^4); Δ, ship displacement (tons); Δ_{i}, virtual displacement, including the added mass of water (tons); L, length between perpendiculars (m); B breadth amidships (m); T_{m} mean draft (m).

The following formula, from Johannessen and Skaar [4], expresses the first few vertical bending natural frequencies in terms of the 2-noded value.

$$N_n \cong N_2 (n-1)^{\alpha} \tag{5}$$

where N_n, natural frequency of the n-noded vertical bending mode (rpm); n number of nodes; α, 0.845 (for general cargo ships); α, 1 (for bulk carriers); α, 1.02 (for tankers).

For the values of the modeled ship,

$I = 44.534\ \mathrm{m}^4$

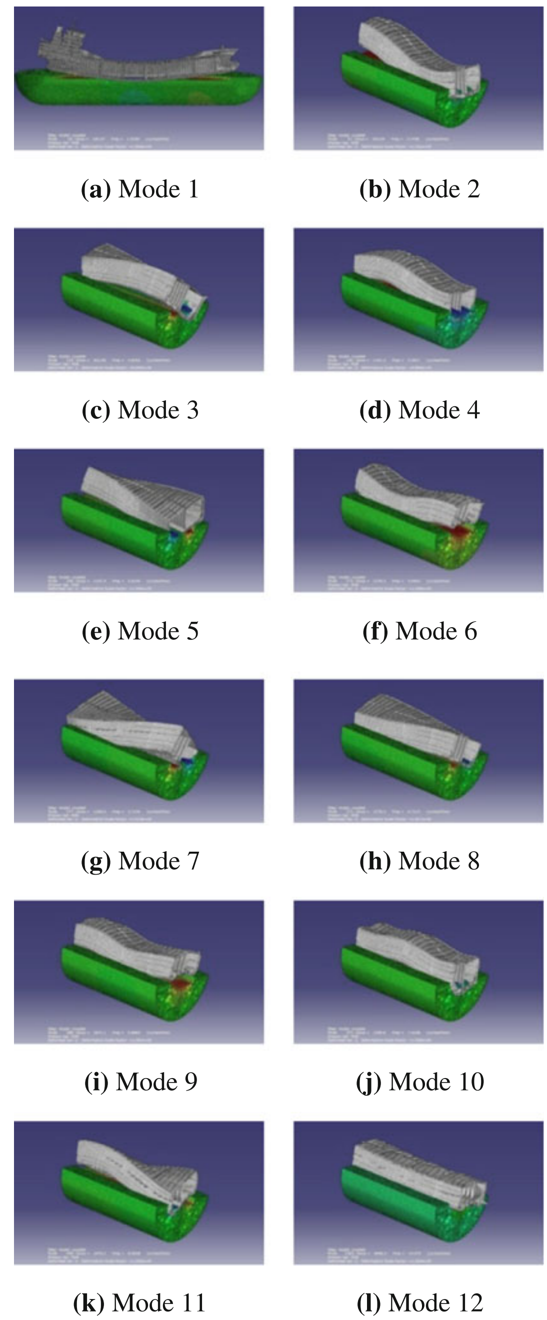


Fig. 17 Global wet mode shapes

$\Delta = 5088.87$ tons

$\Delta_{\mathrm{i}} = 22414.32$ tons

$L = 140$ m

$B = 22.4$ m

$T_{\mathrm{m}} = 2.33$ m.

The natural frequencies of the first four vertical bending modes have been calculated and are listed below

$N_2 = 1.3768$ Hz,

$N_3 = 2.7920$ Hz,

$N_4 = 4.2222$ Hz,

$N_5 = 5.6620$ Hz.

Table 6 Comparison of natural frequencies

Vertical modes	Empirical formula (Hz)	Finite element analysis (Hz)	Diff. (%)
Mode 1 (2-noded)	1.3768	1.9169	28.18
Mode 2 (3-noded)	2.7920	3.7458	25.48
Mode 3 (4-noded)	4.2222	5.2817	20.06
Mode 4 (5-noded)	5.6620	6.8863	17.78

Table 6 shows a comparison of the natural frequencies calculated empirically and obtained with the finite element analysis method.

8 Forced vibration analysis

The ship hull structure includes the outer shell plating and all internal members, which collectively provide the necessary strength to satisfactorily perform the design functions in the expected sea environment. The hull structure responds as a free–free beam (both ends free) when subjected to dynamic loads. The vibration induced by the propulsion system is a common source of ship vibration. Vibration from this source manifests itself in several ways. Dynamic forces from the shafting system are transmitted to the hull through shaft bearings. The propeller induces fluctuating pressures on the surface of the hull, which induces vibration in the hull structure. The main and auxiliary engines can directly cause vibrations through dynamic forces transmitted through their supports and foundations. The response to this forcing can cause the vibration of the hull girder, deckhouse, deck and other structures, local structures and equipment. When attempting to determine the source of vibration, it is necessary to establish the frequency of excitation and to relate the frequency of excitation to the shaft rotational frequency by determining the number of oscillations per shaft revolution. The main engine-induced unbalanced excitations encountered with slow-speed diesel-driven ships are the primary and secondary free engine forces and moments. The engine manufacturer provides the magnitude of these forces and moments. In ship vibrations, the propeller is frequently a source of vibration issues that can cause an excessive ship stern vibration problem.

The consequences of excessive vibration in the stern area can be severe. Deterioration of the structural members can be accelerated as a result of fatigue caused by long-term cyclic vibrations. Excessive vibration can damage or adversely affect the in-service performance of the ship's mechanical and electrical equipment. Prolonged exposure to vibration can also contribute to crew and passenger discomfort, increasing the opportunities for human error. Increased flexibility of the hull girder of larger and, in particular, longer ships with a fine, underwater form can significantly increase the susceptibility to vibration. Moreover, because the weight and distribution of steel within ship structures are optimized as shipbuilders attempt to control production and material costs, the propensity for vibration-related troubles, particularly in the stern section of the vessel, increases. As the demand for higher service speeds for many of these vessels also increases, attendant increases in the propulsive power are required. This translates into higher loads on propellers, which in turn leads to greater propeller excitation and an increase in the risk of vibration and vibration-induced failures. Stern vibration problems arise from the unsteady cavities that attach to the surface of the propeller blades. These create an intense, fluctuating pressure impact on the ship's hull. With modern propeller designs, a small to moderate amount of sheet cavitation is often unavoidable to maintain the required propulsion efficiency. Reconciling the challenges posed by these conflicting technical and operational demands is essential if further improvements in the speed-power-size ratio are to be realized, particularly for ultra-large containerships.

Prediction of propeller-induced hull vibration is not simple. It requires a synthetic analysis involving methodologies from many fields, such as Computational Fluid Dynamics (CFD), the Finite Element Method (FEM), and fluid cavitation dynamics. In a propeller-induced hull vibration assessment, the prediction of stern flow is central to the problem of unsteady propeller loads, cavitation, and propeller-induced hull pressure. The solution to these problems requires detailed knowledge of the turbulent stern flow (including thick and perhaps separated boundary layers), bilge vorticity, and propeller/hull interactions. Traditionally, in ship design, the technology for these predictions was mainly based on regression and empirical formulae. At best, the use of ship flow codes was restricted to potential flow calculations augmented by boundary layer predictions to approximate viscous effects. Propeller calculations were performed using empirically generated effective wakes, and the propeller's interaction with the hull was approximated with a thrust deduction coefficient. Excitation forces from the propeller are transmitted into the ship via the shaft line and also in the form of pressure pulses acting on the ship hull surface above the propeller. Whereas propeller shaft forces (bearing forces) are the most significant factor for vibrations of shaft lines, the predominant factor for vibrations of ship structures are the pressure fluctuations on the hull surface (hull surface forces). In merchant ships, for which a certain degree of propeller cavitation is generally tolerated for the sake of optimizing the propeller efficiency, approximately 10 % of propeller-induced vibration velocities are caused by bearing forces, whereas approximately 90 % are due to pressure fluctuations, or hull surface forces. That is, the excitation forces are introduced into the ship's structure by the pressure pulses acting on the ship's shell.

From experience, the pressure amplitude above the propeller alone is not adequate to characterize the excitation behavior of a propeller. Therefore, no generally valid limits can be stated for pressure fluctuation amplitudes. These amplitudes depend not only on technical constraints, such as the achievable tip clearance of the propeller and the power to be transmitted, but also on the geometry-dependent compromise between efficiency and pressure fluctuations. Nevertheless, pressure amplitudes at a blade frequency of 1–2, 2–8 and over 8 kPa at a point directly above the propeller can be categorized as low, medium, and high, respectively. The total vertical force fluctuations at the blade frequency integrated from pressure fluctuations would range from 10 kN for a small ship to 1000 kN for a high-performance container ship. Whether these considerable excitation forces result in high vibrations depends on the dynamic characteristics of the ship's structure, and can only be judged on the basis of a forced vibration analysis.

There are three methods for predicting hull surface pressure: empirical methods, calculations using advanced theoretical methods and experimental measurements. With regard to the empirical methods, the most well-known and adaptable method is that of Holden et al. [5]. This method is based on the analysis of full-scale measurements for some 72 ships. The method is intended as a first estimate of the likely hull surface pressures using a conventional propeller form. Regression-based formulae for estimation of the noncavitating and cavitating pressure are proposed as follows by Holden et al. [5]. For noncavitating pressure,

$$P_o = \frac{(ND)^2}{70}\frac{1}{z^{1.5}}\left(\frac{K_o}{d/R}\right) \tag{6}$$

and for cavitating pressure,

$$P_c = \frac{(ND)^2}{160}\frac{V_s(w_{\mathrm{Tmax}} - w_e)}{\sqrt{h_a+10.4}}\left(\frac{K_c}{d/R}\right) \tag{7}$$

are given.

N: Propeller rpm

D: Propeller diameter (m)

V_s: Ship speed (m/s)

Z: Blade number

R: Propeller radius (m)

d: Distance from 0.9R to a position on the submerged hull when the blade is at the top dead center position (m)

w_e: Mean effective full-scale Taylor wake fraction

w_{Tmax}: Maximum value of the Taylor wake fraction in the propeller disc

h_a: Depth of the shaft centerline

K_0: 1.8 + 0.4 (d/R) for $d/R \le 2$

K_c: 1.7 + 0.7 (d/R) for $d/R < 1$

K_c: 1.0 for $d/R > 1$

The total pressure impulse, which combines both the cavitating 'p_c' and the noncavitating 'p_o' components, is then calculated from

$$P_z = \sqrt{P_o^2 + P_c^2}. \tag{8}$$

The values for the modeled ship are given below.

$N = 173$ rpm

$D = 4.25$ m

$V_s = 7.46$ m/s

$Z = 4$

$R = 2.125$ m

$D = 2.58$ m

$w_{\mathrm{Tmax}} = 0.298$

$w_e = 0.059$

$h_a = 3$ m

$K_0 = 2.286$

$K_c = 1.0.$ mm^2

The calculated values of P_0, P_c and P_z are as follows.

$P_0 = 1817.60$ N/

$P_c = 1355.42$ N/mm^2

$P_z = 2267.34$ N/mm$^{2.}$

The total amplitude of the hull vertical surface force has been obtained by integrating P_z over the hull surface area above the propeller [6] and found to be

$F_z = 203{,}625$ N $\approx$ 204 kN.

The location of the propeller-induced excitation caused by the hull surface force is shown in Fig. 18.

The excitation frequency with a propeller having 4 blades is given as

$$f = \frac{N \cdot Z}{60} = \frac{173.4}{60} = 11.53 \text{ Hz.}$$

The total damping associated with overall ship hull structure vibrations is generally considered to be a combination of structural damping, cargo damping, water friction, pressure wave generation and surface wave generation. For the forced vibration analysis, it is assumed that the effects due to structural damping, cargo damping, water friction and pressure wave generation can be lumped together. The effect of surface wave generation need only be considered for vibrations with very low frequencies. This effect is generally neglected. For

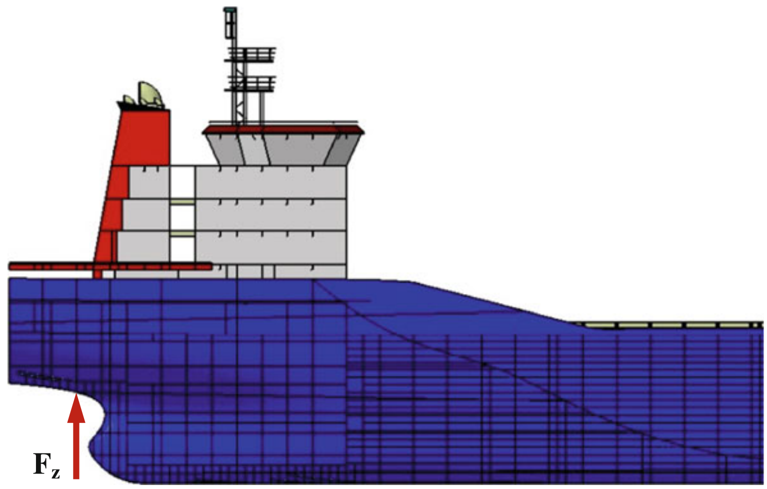


Fig. 18 Excitation caused by the propeller-induced force

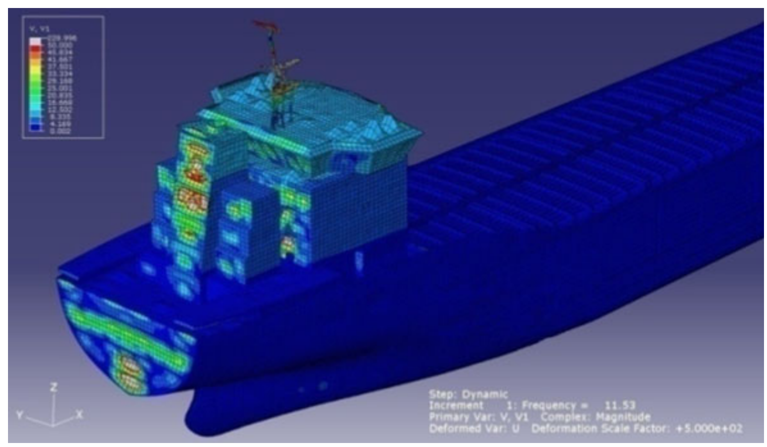


Fig. 19 Vibration velocity in the X direction

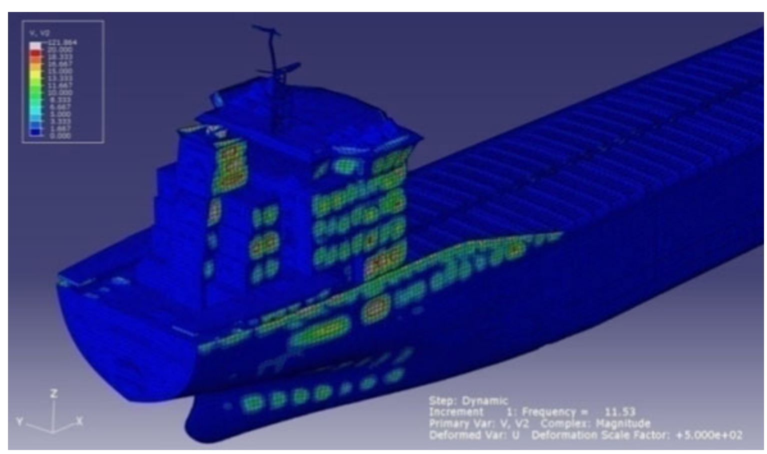


Fig. 20 Vibration velocity in the Y direction

simplification, a constant damping coefficient of 1.5 percent of the critical damping may be used for the entire range of propeller rpm and main engine orders. For the given values of excitation and damping, the forced vibration analysis has been conducted, and the vibration velocity amplitudes in the X, Y, and Z directions are shown in Figs. 19, 20, 21.

Excitation force (Hull Surface Force): $F_z = 204$ kN

Excitation frequency (Blade Rate Frequency): $f = 11.53$ Hz

Global Hull Damping Ratio: $\zeta = 0.015$.

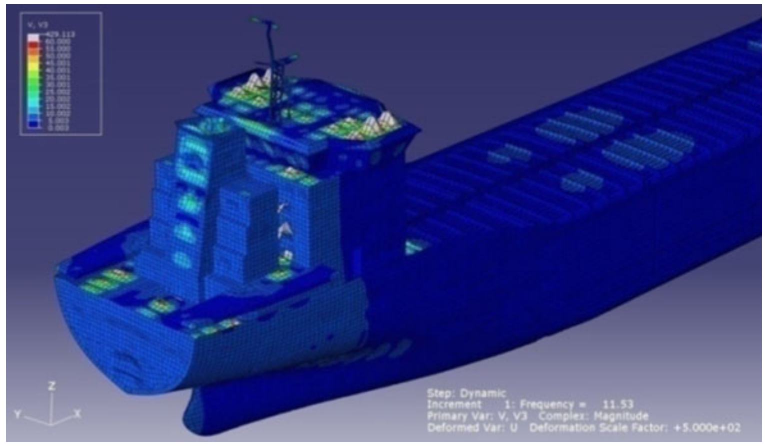


Fig. 21 Vibration velocity in the Z direction

Table 7 Overall frequency-weighted rms values

Vertical modes	Passenger accom. (mm/s)	Crew accom. (mm/s)	Work spaces (mm/s)
A	4	6	8
B	2	3	4

A, values above which adverse comments are probable; B, Values below which adverse comments are not probable

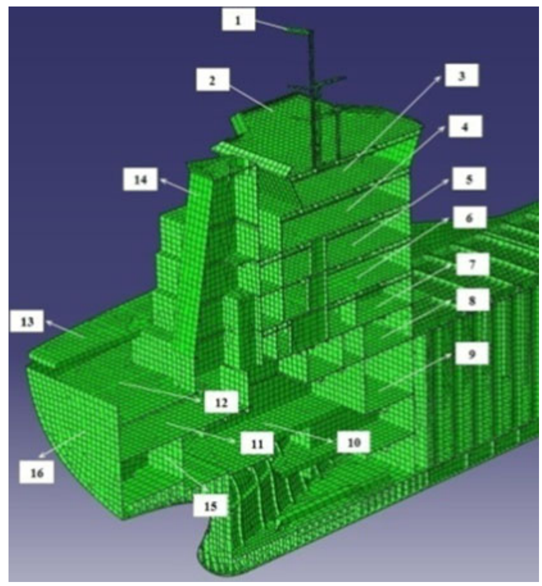


Fig. 22 Selected points for vibration velocity amplitudes

9 Vibration limits for crew, passengers and local structures

The ISO 6954 [7] has been widely used as acceptance criteria for crew habitability and passenger comfort. The criteria are designed to ensure that the vibration levels are below the level at which the crew and the passengers experience discomfort. The ISO 6954 criteria can be transformed into a statement such that for each peak response component (in either the vertical, transverse, or longitudinal direction), at 5 Hz and above, the velocity is acceptable below 4 mm/s and adverse conditions are probable above 9 mm/s. The ISO 6954 [7] has been revised to reflect recent knowledge about human sensitivity to whole-body vibrations. The frequency weighting curves are introduced to represent human sensitivity to multifrequency vibration for a broad range of frequencies, which are consistent with the combined frequency weighting in ISO 2631-2. The ISO 6954 [8] provides criteria for crew habitability and passenger comfort in terms of overall frequency-weighted rms values from 1 to 80 Hz for three different areas. The simplified presentation is shown in Table 7.

Excessive ship vibration is to be avoided to reduce the risk of structural damage on the local structures. Structural damage such as fatigue cracking due to excessive vibration may occur on local structures, including, but not limited to, engine foundation structures, engine stays, steering gear rooms, tank structures, funnels, and radar masts. It should be noted that structural damage due to excessive vibrations varies according to the local structural detail, actual stress level and local stress concentration and material property of the local structures. Therefore, the vibration limits for local structures are to be used as a reference to reduce the risk of structural damage due to excessive vibration during the normal operating conditions. Above 5 Hz, the vibration limits are specified in terms of velocity amplitude, and below 5 Hz, they are specified in terms of displacement. The local structure vibrations of primary interest are generally above 5 Hz. The vibration limits can be transformed into a statement such that for each peak response component (in either the vertical, transverse, or longitudinal direction), from 5 Hz and above, it is recommended that the velocity be kept below 30 mm/s, and damage is probable above 60 mm/s. Some critical points from crew accommodations, work spaces and local structures, shown in Fig. 22, have been selected, and their vibration velocity amplitudes are listed in Table 8.

Table 8 Vibration velocity amplitudes

Point	V_x (mm/s)	V_y (mm/s)	V_z (mm/s)
1	50.8401	4.6778	24.4326
2	10.2054	0.3286	228.9610
3	8.0873	0.1100	126.9220
4	6.6947	0.0985	122.6420
5	5.2620	0.0725	167.0880
6	3.9025	0.0481	113.6760
7	2.6865	0.0127	5.4783
8	1.9068	0.2650	4.5171
9	1.4532	0.0466	0.5298
10	1.5421	0.8152	48.9489
11	2.3346	0.8880	323.3170
12	2.7745	0.1255	89.4936
13	3.3355	0.3477	2.1349
14	77.9465	0.1909	14.4173
15	102.8100	0.2916	5.4196
16	90.2826	0.2418	5.7030

10 Conclusions

We have arrived at the following conclusions in this study in which analyses of free and forced ship vibrations have been conducted using the FEM.

- In the primitive ship model, the superstructure and girders were not present. After the superstructure, girders, bulkheads, stiffeners, foundation, railings and masts were added to the ship hull model, it was clear that the natural frequencies and mode shapes had noticeably changed.
- It has been determined that the results (natural frequencies and mode shapes) of dry analysis and wet analysis differ distinctly from each other. This has highlighted the importance of performing a wet analysis.
- It has been detected that there is an approximately 30 % difference between the natural frequency values calculated empirically and the ones obtained using finite element analysis. This shows the necessity of finite element modeling and analysis to obtain accurate results in ship vibration investigations.
- The results of propeller-induced vibration analysis have shown that the vibration velocities remain under the limits in accommodations but usually exceed the limits in local structures. This may result in the fatigue failure of local structural members or the malfunctioning of machinery and equipment.

The ship that we modelled is a double-hull ship. The empirical formula to calculate the natural frequency is the function of the moment of inertia of the cross section of the hull. This parameter is not suitable for double-hull ships. Since the finite element model is based on the real double-hull ship, finite element analysis is accepted as more accurate. Another method to verify these results is a global experimental modal analysis on the ship and this high budget work is to be conducted in our future works.

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References

1. Ozsoysal R (2004) A review of recent ship vibration papers. Shock Vib Dig 36:207–214
2. American Bureau of Shipping (2006) Guidance notes on ship vibration, New York
3. Kumai T (1968) On the estimation of natural frequencies of vertical vibration of ships. Report of Research Institute for Applied Mechanics, 16(54)
4. Johannessen H, Skaar KT (1980) Guidelines for prevention of excessive ship vibration. SNAME Trans 88:319–356
5. Holden KO, Fagerjord O, Frostad R (1980) Early design-stage approach to reducing hull surface force due to propeller cavitation. SNAME Trans 88:403–442
6. Lee SK, Liao M, Wang S (2006) Propeller-induced hull vibration—analytical methods. Proceedings of the second international ship noise and vibration conference. London, UK, June 28
7. ISO 6954 (1984) Mechanical vibration and shock—guidelines for the overall evaluation of vibration in merchant ships
8. ISO 6954 (2000) Mechanical vibration—guidelines for the measurement, reporting and evaluation of vibration with regard to habitability on passenger and merchant ships