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A hydroelastic vibration analysis of a container ship using finite element and boundary element methods

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ABSTRACT

This paper presents a hydroelastic investigation of a container ship using the finite element and boundary element methods. The hydroelasticity method adopted in this study has two parts: *dry* and *wet* analysis parts. In the *dry* analysis part, a finite element model of the entire ship structure was prepared using the ABAQUS finite element software, and the *in vacuo* dynamic properties (natural frequencies and associated mode shapes) of the structure were obtained. In the *wet* part of the analysis, it is assumed that the fluid is ideal, i.e., inviscid, incompressible and its motion is irrotational. The fluid-structure interaction forces are associated with the inertia effect of the surrounding fluid, and a boundary integral equation method was adopted for the calculations. The wetted surface of the ship hull was idealized using appropriate boundary elements. A higher order boundary element method was adopted in the present study. The fluid-structure interaction forces were calculated solely in terms of the generalized added mass coefficients. In order to assess the influence of fluid on the dynamic response characteristics, the *wet* frequencies and associated *wet* mode shapes were calculated. The *wet* frequencies obtained from the method of analysis were compared with those obtained from the ABAQUS finite element software. A very good comparison was obtained between the calculations.

1 INTRODUCTION

Hydroelasticity theory becomes essential for investigating the dynamic behavior of a floating, flexible structure such as a ship hull structure using a fluid-structure interaction model. The dynamic interaction between the structure and fluid medium brings together all the aspects associated with both structural dynamics and fluid dynamics. When the structure is in contact with a fluid of comparable density, such as water, the fluid loading which depends on the structural surface motions will significantly alter the dynamic state of the structure. However, the fluid-structure interaction can be considered as *feedback* coupling. In

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other words, the equations of structural and fluid motions must be solved simultaneously unless certain assumptions are made to uncouple them.

In the seventies, Bishop and Price [1] established their well-known hydroelasticity theory of ships based on the principles of applied mechanics and hydrodynamics. The theory was later extended by Bishop, *et al* [2] for allowing the analysis of three dimensional fluid-structure interaction problems. The theory has been successfully used for different kinds of structures such as SWATH (small water-plane area twin hull), jack-up rigs, etc. traveling in irregular seaway. Ergin, *et al* [3] demonstrated the applicability of the theory to structures such as submarine pressure hull and offshore pipelines by comparing the predictions with the experimental measurements. Alternatively, Ergin and Temarel [4] proposed a hydroelasticity analysis method for investigating the dynamics of elastic structures containing and/or submerged in fluid. Their method is based on a boundary integral equation method in conjunction with the method of images, in order to impose appropriate boundary conditions on the fluid's free surface. Ergin and Uğurlu [5] used the method to investigate the dynamic characteristics of partially submerged cantilever plates, and Uğurlu and Ergin [6] adopted the method for analyzing the shell structure containing flowing fluid. On the other hand, Röhr and Möller [7] described a combined finite element – boundary element procedure for elastic vibration analysis of complex wetted structures, and they calculated the *wet* eigen-frequencies and eigen-modes of a container ship.

In this paper, the dynamic characteristics (e.g., natural frequencies and mode shapes) of a container ship are investigated. The method adopted in this study is based on the boundary integral equation method and the method of images. In this investigation, it is assumed that the fluid is ideal, i.e., inviscid, incompressible and its motion is irrotational. It is also assumed that the ship hull vibrates in its *in vacuo* eigenmodes when it is in contact with fluid, and that each *in vacuo* mode gives rise to a corresponding surface pressure distribution on the wetted surface of the ship structure. The *in vacuo* dynamic analysis entails the vibration of the ship hull in the absence of any external force and structural damping, and the corresponding dynamic characteristics of the container ship are obtained using a standard finite element software.

At the fluid-structure interface, continuity considerations require that the normal velocity of the fluid is equal to that of the structure. The normal velocities on the wetted surface are expressed in terms of the modal structural displacements. By using the boundary integral equation method the fluid pressure is eliminated from the problem, and that the fluid-structure interaction forces are calculated in terms of the generalized hydrodynamic added mass terms. During this analysis, the wetted surface is idealized by using appropriate boundary elements, referred to as hydrodynamic panels. To assess the influence of the fluid on the dynamic response behavior of the container ship, the *wet* frequencies and associated mode shapes are calculated.

In a further analysis, the finite element method (ABAQUS [8]) was used for modeling both the structural and fluid domains. The fluid domain surrounding the ship hull was idealized by using acoustic fluid elements. Two different loading conditions were used in the calculations: full load and ballast conditions. The cargo loads, ballast weight, etc. applied as inertia mass elements over the cargo area, etc. The *wet* frequencies and mode shapes were calculated by solving the fluid-structure interaction problem.

2 MATHEMATICAL MODEL

2.1 Fluid Structure Interaction Problem

The fluid is assumed ideal, i.e., inviscid and incompressible, and its motion is irrotational and there exists a fluid velocity vector, **v**, which can be defined as the gradient of the velocity potential function ϕ as

$$\mathbf{v}(x, y, z, t) = \nabla \phi(x, y, z, t). \quad (1)$$

where ϕ satisfies the Laplace's equation, $\nabla^2 \phi = 0,$ throughout the fluid domain.

Before describing the responses of the flexible structure, it is necessary to assign coordinates to the deflections at various degrees of freedom. One particular set of generalized coordinates having the advantage of being unambiguous and easily commended is the principal coordinates of the *dry* structure (see, for example, Ergin *et al.* [9]). For a structure vibrating in an ideal fluid, with frequency ω, the principle coordinate, describing the response of the structure in the rth modal vibration, may be expressed by

$$p_r(t) = p_{0r}\, \mathrm{e}^{\mathrm{i}\omega \mathrm{t}} \quad (2)$$

The velocity potential function due to the distortion of the structure in the rth *in vacuo* vibrational mode may be written as follows

$$\phi_r(x, y, z, t) = \mathrm{i}\,\omega \phi_r(x, y, z)\, p_{0r}\, \mathrm{e}^{i\omega t} \qquad r = 1, 2, \ldots, M \quad (3)$$

where M represents the number of modes of interest, and p_{0r} is an unknown amplitude for the rth principal coordinate.

On the wetted surface of the vibrating structure the normal fluid velocity must equal to the normal velocity on the structure and this condition for the rth modal vibration of the elastic structure submerged in fluid can be expressed as

$$\frac{\partial \phi_r}{\partial \mathrm{n}} = \mathbf{u}_r \cdot \mathbf{n}, \quad (4)$$

where **n** is the unit normal vector on the wetted surface and points into the region of interest. The vector $\mathbf{u}_r$ denotes the displacement response of the structure in the rth principal coordinate and it may be written as

$$\mathbf{u}_r(x, y, z, t) = \mathbf{u}_r(x, y, z)\, p_{0r}\, \mathrm{e}^{i\omega t} \quad (5)$$

where $\mathbf{u}_r(x, y, z)$ is the rth modal displacement vector of the median surface of the structure, and it is obtained from the *in vacuo* analysis.

It is assumed that the elastic structure vibrates at relatively high frequencies so that the effect of surface waves can be neglected. Therefore, the free surface condition (infinite frequency limit condition) (see, for instance, [5]) for the perturbation potential can be approximated by

$$\phi_r = 0, \qquad \text{on the free surface.} \quad (6)$$

The method of images [10] may be used to satisfy this boundary condition. By adding an imaginary boundary region, the condition given by equation (6) at the horizontal surface can be omitted; thus the problem is reduced to a classical Neumann case. The details of the method of images may be found in Ergin and Temarel [4].

2.2 Numerical Evaluation of Perturbation Potential ϕ

The boundary value problem for the perturbation potential, ϕ, may be expressed in the following form:

$$c(\xi)\phi(\xi) = \iint_{S_W} (\phi^*(s,\xi)q(s) - \phi(s)q^*(s,\xi))\,dS \tag{7}$$

where ξ and s denote, respectively, the evaluation and field points on the wetted surface. ϕ^* is the fundamental solution and expressed as follows,

$$\phi^*(s,\xi) = \frac{1}{4\pi r} \tag{8}$$

$q = \partial\phi/\partial\mathbf{n}$ is the flux, and r the distance between the evaluation and field points. The free term $c(\xi)$ is defined as the fraction of $\phi(\xi)$ that lies inside the domain of interest. Moreover, $q^*(s,\xi)$ can be written as

$$q^*(s,\xi) = -(\partial r/\partial\mathbf{n})/4\pi r^3 \tag{9}$$

For the solution of equation (7) with boundary condition (4), the wetted surface can be idealized by using boundary elements, referred to as hydrodynamic panels, and the distribution of the potential function and its flux over each hydrodynamic panel may be described in terms of the shape functions and nodal values as

$$\phi_e = \sum_{j=1}^{n_e} N_{ej}\,\phi_{ej}, \qquad q_e = \sum_{j=1}^{n_e} N_{ej}\,q_{ej} \tag{10}$$

Here, n_e represents the number of nodal points assigned to each hydrodynamic panel, and N_{ej} the shape function adopted for the distribution of the potential function. e and j indicates the numbers of the hydrodynamic panels and nodal points, respectively. In the case of a linear distribution adopted in this study, the shape functions for a quadrilateral panel may be expressed as (see Wrobel [11])

$$\begin{aligned} N_{e1} &= ((1-\varsigma)(1-\eta))/4 \\ N_{e2} &= ((1+\varsigma)(1-\eta))/4 \\ N_{e3} &= ((1+\varsigma)(1+\eta))/4 \\ N_{e4} &= ((1-\varsigma)(1+\eta))/4 \end{aligned} \tag{11}$$

After substituting equations (10) and (11) into equation (7) and applying the boundary condition given in equation (4), the unknown potential function values can be determined from the following set of algebraic equations

$$c_k \phi_k + \sum_{i=1}^{m} \sum_{j=1}^{n_m} (\phi_{ij} \iint_{\Delta S_i} N_j \, q^* \, dS = \sum_{i=1}^{m} \sum_{j=1}^{n_m} (\mathbf{u}_{ij} \cdot \mathbf{n}_j \iint_{\Delta S_i} N_j \, \phi^* \, dS), k = 1, 2, \ldots, m \tag{12}$$

where m denotes the number of nodal points used in the discretization of the structure. ϕ_{ij} and $\mathbf{u}_{ij}$ represent, respectively, the potential and displacement vector for the jth nodal point of the ith hydrodynamic panel.

2.3 Calculation of Generalized Fluid-Structure Interaction Forces

Using the Bernoulli's equation and neglecting the second order terms, the dynamic fluid pressure on the elastic structure due to the rth modal vibration becomes

$$P_r(x, y, z, t) = -\rho \frac{\partial \phi_r}{\partial t}, \tag{13}$$

where ρ is the fluid density. Substituting equation (3) into (13), the following expression for the pressure is obtained,

$$P_r(x, y, z, t) = -\rho \, \omega^2 \, \phi_r(x, y, z) \, p_{0r} \, \mathrm{e}^{i\omega t}. \tag{14}$$

The kth component of the generalized fluid-structure interaction force due to the rth modal *in-vacuo* vibration of the elastic structure can be expressed in terms of the pressure acting on the wetted surface of the structure as

$$\begin{aligned} Z_{kr} &= \iint_{S_W} P_r(x, y, z, t) \, \mathbf{u}_k \, \mathbf{n} \, dS \\ &= p_{0r} \, \mathrm{e}^{\mathrm{i}\omega \mathrm{t}} \iint_{S_W} \rho \, \omega^2 \phi_r \, \mathbf{u}_k \, \mathbf{n} \, dS \end{aligned} \tag{15}$$

The generalized added mass, A_{kr} term can be defined as

$$A_{kr} = \rho \iint_{S_W} \phi_r \, \mathbf{u}_k \, \mathbf{n} \, dS, \tag{16}$$

Therefore, the generalized fluid-structure interaction force component, Z_{kr}, can be written as

$$Z_{kr}(t) = A_{kr} \, \omega^2 \, p_{0r} \, \mathrm{e}^{i\omega t} = A_{kr} \, \ddot{p}_r(t) \tag{17}$$

2.4 Calculation of *Wet* Frequencies and Mode Shapes

It should be noted that, in the case when a body oscillates in or near a free surface, the hydrodynamic coefficients exhibit frequency dependence in the low frequency region, but show a tendency towards a constant value in the high frequency region. In this study, it is assumed that the structure vibrates in a relatively high frequency region so that the generalized added mass values are constants and evaluated by use of equation (16). Hence the generalized equation of motion for the dynamic fluid – structure interaction system (see, e.g., Ergin *et al.* [3]), assuming free vibrations with no structural damping, is

$$\left[-\omega^2(\mathbf{a}+\mathbf{A})+\mathbf{c}\right]\mathbf{p}=0, \tag{18}$$

where **a** and **c** denote the generalized structural mass and stiffness matrices, respectively. The matrix **A** represents the infinite frequency generalized added mass coefficients.

Solving the eigenvalue problem, expressed by (18), yields the *wet* frequencies and associated *wet* mode shapes of the structure in contact with fluid. To each *wet* frequency ω_r, there is a corresponding *wet* eigenvector $\mathbf{p}_{0r}=\{p_{r1},p_{r2},\ldots,p_{rm}\}$. The corresponding uncoupled *wet* mode shapes for the structure partially and totally in contact with fluid are obtained as

$$\bar{\mathbf{u}}_r(x,y,z)=\{\bar{u}_r,\bar{v}_r,\bar{w}_r\}=\sum_{j=1}^{M}\mathbf{u}_j(x,y,z)\,\mathbf{p}_{rj}, \tag{19}$$

where $\mathbf{u}_j(x,y,z)=\{u_j,v_j,w_j\}$ denote the *in vacuo* mode shapes of the elastic structure and M the number of mode shapes included in the analysis. It should be noted that the fluid – structure interaction forces associated with the inertial effect of the fluid do not have the same spatial distribution as those of the *in vacuo* modal forms. Consequently, this produces hydrodynamic coupling between the *in vacuo* modes of the structur. This coupling effect is introduced into equation (18) through the generalized added mass matrix **A**.

3 NUMERICAL RESULTS

A container ship is adopted for the numerical calculations, and its main particulars are given in Table 1. Two different loading conditions are considered: full load and ballast conditions. The cargo loads are distributed as inertia mass elements over the cargo area inner bottom plating. Ballast weight, heavy fuel oil and other tank weights are also applied as inertia mass elements. The light weight of the ship (LWT) is 9000 tons. The load groups are given in Table 2 for the full load and ballast conditions. The structural finite element model of the container ship is shown in Figure 1.

Table 1: The main particulars of the container ship

Length (L_{bp})	171 m
Breadth (B)	28 m
Depth (D)	16.10 m
Draught (T)	10.00 m
Speed (V)	19.50 knot

The *in vacuo* dynamic characteristics of the container ship were obtained using the ABAQUS finite element software [8]. This produces information on natural frequencies and the normal mode shapes of the *dry* container ship in vacuum. In these calculations, the ship hull was discretized by using 176 800 structural elements. The *in vacuo* dynamic characteristics of the ship structure are scaled to a generalized structural mass of 1 ton m^2.

Table 2: Load groups in full load and ballast conditions (tons).

	Full Load Condition	Ballast Condition
Cargo loads	17150	-
Ballast weight	3021	11250
Heavy Fuel Oil	1886	305
Marine Diesel Oil	165	36.8
Fresh water	206	26.7
Other weights	165	165

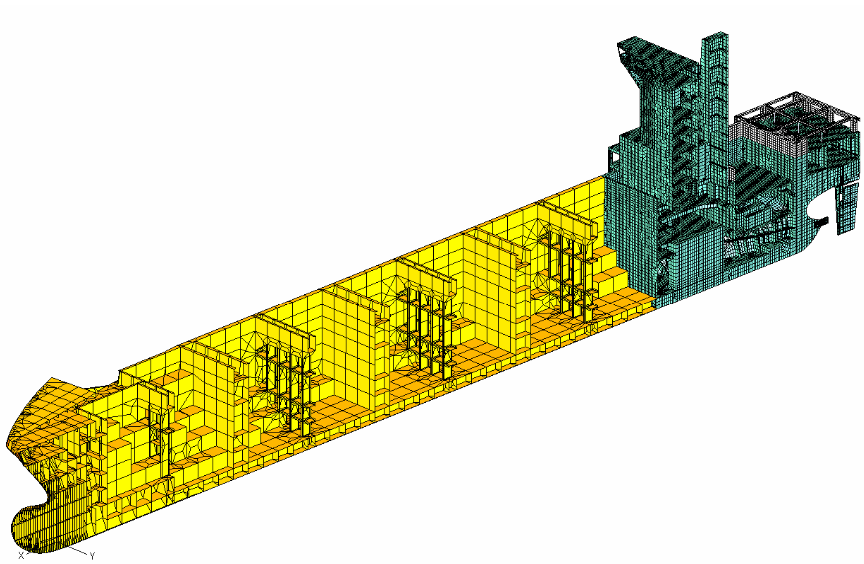

Figure 1: The structural finite element model of the container ship.

It should be noted that the finite element calculations were carried out by the Delta Marine of Turkey. The *dry* natural frequencies of the container ship are given in Tables 3 and 4 for the full load and ballast conditions.

In the *wet* part of the calculations, the fluid is assumed ideal, i.e., inviscid and incompressible, and its motion is irrotational. The wetted surface of the container ship was idealized by using boundary elements, referred to as hydrodynamic panels. 10772 and 6739 hydrodynamic panels were distributed over the wetted surface of the ship hull, respectively, for the full load and ballast conditions. A linear distribution of unknown potentials was assumed over the each hydrodynamic panel. These unknown potentials were calculated using the kinematic boundary conditions imposed at each nodal point. The discritized wetted surface is shown in Figure 2 for the full load condition. The kinematic boundary conditions were obtained from the *in vacuo* finite element calculations. By using Eq.(18), the eigenvalues and eigenvectors of the fluid-structure interaction system were calculated, and the *wet* frequencies are presented in Tables 3 and 4 for the first five global ship hull vibrations together with the *dry* results, respectively, for the full load and ballast conditions. A number of 12 *in vacuo* modes was adopted for the *wet* calculations.

In a further analysis, a standard finite element program (ABAQUS) was adopted for the *wet* calculations. The ship structure and fluid surrounding the ship hull were discretized by finite elements. For the fluid medium, 60 300 acoustic finite elements were employed for the full load condition.

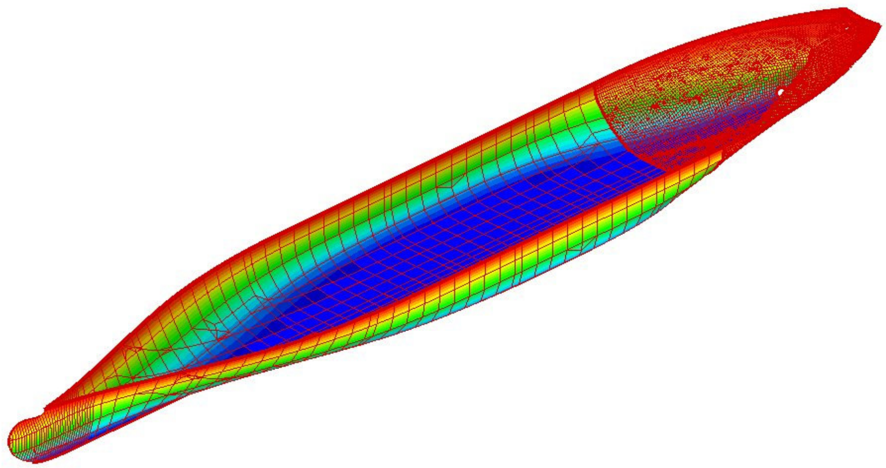

Figure 2: The discretized wetted surface of the container ship for the full load condition.

Table 3: Calculated frequencies for the full load condition.

Mode	*In vacuo* FEM	*Wet* FEM	*Wet* BEM	Difference %
1. Torsion	1.119	0.998	0.999	0.1
1. Bending	1.331	1.015	1.041	2.5
Bend.&Tors.	1.515	1.328	1.349	1.6
2. Torsion	2.547	2.325	2.344	0.8
2. Bending	2.676	2.059	2.072	0.6

Table 4: Calculated frequencies for the ballast condition.

Mode	*In vacuo* FEM	*Wet* FEM	*Wet* BEM	Difference %
1. Torsion	1.340	1.221	1.217	0.3
1. Bending	1.608	1.290	1.210	6.2
Bend.&Tors.	1.981	1.787	1.776	0.6
2. Torsion	3.337	2.504	2.386	4.7
2. Bending	3.455	-	3.129	-

It can be seen from Tables 3 and 4 that there are very good agreement between the results obtained from the finite element analysis (ABAQUS) and boundary element analysis proposed in this study. The largest differences obtained are 2.5% and 6.2%, respectively, for the full load and ballast conditions. On the other hand, it can be observed from the tables that the frequencies decrease with increasing area of contact with the fluid. The ship hull has the largest area of contact for the full load condition, and therefore, the lowest frequencies were obtained for this case.

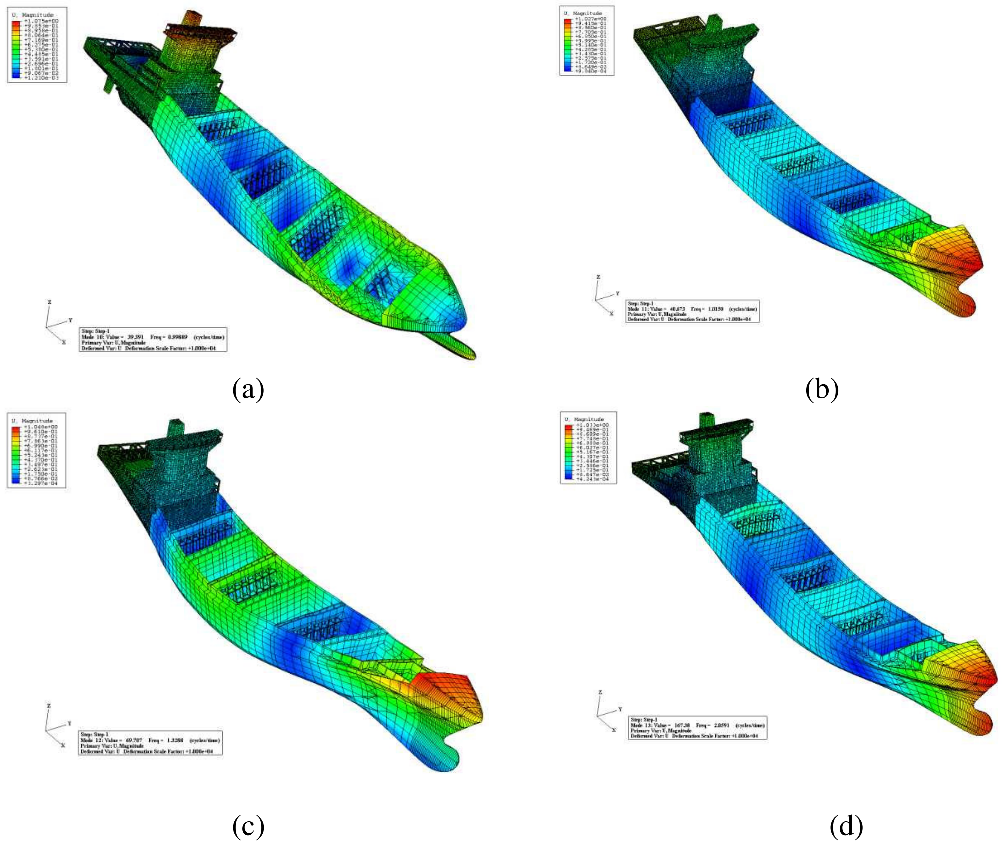

Figure 3: The first four global modes of the container ship: (a) 1st torsion, (b) 1st bending, (c) horizontal bending & torsion, (d) 2nd bending.

The mode shapes indicated in Tables 3 and 4 are the global vibration modes of the ship hull, and the first four global mode shapes are shown in Figure 3.

4 CONCLUSIONS

The dynamic characteristics (*wet* frequencies and associated mode shapes) of a container ship for two different load conditions were calculated by an approach based on the boundary integral equation method, and the frequencies obtained were compared with the results of the finite element calculations. A very good comparison was obtained between the predictions of the boundary element and finite element methods. It can be concluded from the results that the boundary integral equation method proposed in this study is suitable for the dynamic analysis of ship hull structures and also for other elastic structures fully or partially in contact with fluid.

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