

# Finite Element Model Updating of a Scale Model Ship Using Experimental Modal Analysis and Response Surface Methodology

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## Abstract

The structural dynamics of scale model ships used in towing tank experiments may affect the measured values of propeller-induced pressure fluctuations. The International Towing Tank Conference (ITTC) recommends monitoring structural responses during the tests to evaluate this effect but does not provide a specific procedure for such monitoring. Apart from considering structural behavior during towing tests, having a reliable numerical model of the scale model ship structure is essential to be aware of its vibratory behavior not only prior to lab experimentation to control experimental errors but also when replacing costly physical experiments with cost-effective computational simulations. To address this, we generate a finite element (FE) model of the scale model ship used in our towing tests. We present a methodology for characterizing this FE model based on the physical model's behavior to ensure its reliability for further numerical dynamic analyses. The properties of the FE model are updated by using response surface methodology (RSM) to align numerical and experimental resonance responses. Latin hypercube sampling (LHS) technique is employed to generate the design space, whereas the modal assurance criterion (MAC) serves as a performance metric to ensure modal consistency before and after optimization. This approach provides a reliable FE model that offers a cost-effective alternative to expensive physical tests for enhancing the precision of propeller-induced pressure measurements.

**Keywords** Scale model ship; Finite element model update; Propeller-induced pressures; Latin hypercube sampling; Response surface methodology

## 1 Introduction

### 1.1 Significance of noise and vibrations onboard ships

Noise and Vibrations present an important design and operational challenge for marine vehicles, which experi-

ence dynamic loads generated by waves, internal machinery, and propellers. With the increase in ship size and speed, this problem has become more relevant (Lin et al., 2009). Excessive noise and vibrations are detrimental to passenger comfort and crew habitability. Moreover, high levels of structure-borne vibrations may cause fatigue failure of structural components or malfunctioning machinery (ABS, 2023; Yucel and Arpaci, 2013).

In order to mitigate the hazardous noise and vibration levels and improve comfort onboard ships, international organizations such as the International Maritime Organization (IMO) and the International Organization for Standardization (ISO) published statutes and standards, e.g. IMO (2014; 2012; 2015), MLC (2006), and ISO (2018; 2017; 2016), which aim to set noise and vibration limits as well as procedures to measure any hazardous levels. In addition, classification societies have also published rules and regulations in this regard, e. g., ABS (2019; 2017), BV (2016), and DNV (2014), to guide designers in developing noise and vibration mitigation strategies.

Although ships are built in accordance with the criteria set by the above organizations and classification societies, research still documents comfort issues related to onboard noise and vibration levels. Researchers have continued developing criteria to assess onboard comfort and thus

## Article Highlights

- The elastic moduli of the FE model of a towing tank ship are updated to match the frequency and shape of experimental modal forms.
- LHS-based ANOVA and RSM optimization are implemented to improve the MAC values for the first and second normal modes of the ship model.
- The updated FE model can be used to predict the effects of the structural vibrations of ship models on hydrodynamic data measured in towing tank tests.

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understand this issue. Goujard et al. (2005) generated a questionnaire to assess comfort onboard ships, and its results showed that noise and temperature are the comfort criteria. Borelli et al. (2021) measured noise levels on Ro-Pax ships and assessed noise annoyance in cabins and its effect on sleep quality, which is not considered in current standards. Liu et al. (2023) developed a revised annoyance model based on noise and vibration measurements.

High levels of structural vibrations may cause ship structural damage. International standards (ISO, 2018; 2008) set procedures to measure vibrations on ship structures, whereas the guidelines of classification societies (ABS, 2022, 2023; LR, 2024, 2021) assess ship vulnerability to vibration-induced fatigue and structural failure. In these regulations, fatigue emerges as a main issue regarding the effect of structure-borne noise and vibration on ship structures.

The importance of the vibratory response of ship structures to dynamic loading compels designers to incorporate dynamic analysis into structural design. This approach includes experimental measurements and numerical simulation of the ship structure in early design stages to study its response to external excitations. For instance, Norwood and Dow (2013) presented different ship structural dynamic methodologies in finite element (FE) simulations, including wave-, slam-, and propeller-induced vibrations. Additional examples of such dynamic analysis and ship structural response measurements are found in the works of Zambon et al. (2021), Pais et al. (2017), and Zou et al. (2022).

Reliable numerical models of ship structures and characterized sources of noise and vibrations are essential, specifically at the early stages of design, to obtain a proper estimate of ship structural behavior due to dynamic loadings.

## 1.2 Propeller-induced hull pressures

The vibration design of marine vessels should consider multiple sources, including prime movers, shaft lines, propellers, heating, ventilation, and air conditioning (HVAC) systems, maneuvering devices, auxiliary machinery, intakes and exhausts, and the slamming phenomenon (Carlton and Vlastic, 2005). Propellers often affect onboard noise and vibration, as well as underwater radiated noise. They are a primary source of vibrations for cruise ships, fast ferries, and large container carriers (Norwood and Dow, 2013). Their rotation induces pulsing forces on the ship structure in terms of axial and torsional forces on the shaft line and bearings (Senjanović et al., 2019; Zambon and Moro, 2022) and fluctuating pressures on the stern hull above the propeller (Bosschers, 2009; Su et al., 2018).

### 1.2.1 Propeller-induced pressure measurements

Researchers have conducted studies on propeller-induced pressures to characterize propellers as noise and vibration sources. These works involved model- and full-scale measurements. These studies also led to the development of

empirical formulas for predicting hull pressures generated by propellers, such as the work done by Holden et al. (1981), and to provide ship designers with strategies to reduce these pressures, e.g., the guidelines provided in ABS (2023) and LR (2024).

The International Towing Tank Conference (ITTC) recommends guidelines and procedures for conducting hydrodynamic model-scale tests. These include the measurements of model-scale propeller-induced pressures in towing tanks, cavitation tunnels, and circulating water tunnels (Alves Pereira et al., 2024a; 2024b). Some researchers have employed these measurements to assess propeller-induced fluctuating pressures (Bosschers, 2018a; Li et al., 2015), investigate the effect of cavitation on onboard noise and vibrations (van Wijngaarden et al., 2005), or validate different numerical simulations. These simulations include multiphase delayed detached eddy simulation (DDES) (Li et al., 2015), the boundary element method (BEM) (Lafeber et al., 2009), and the unsteady Reynolds-Averaged Navier-Stokes (URANS) method (Ge et al., 2019; Zheng et al., 2019).

Model-scale propeller-induced pressure measurements are conducted by installing flush-mounted pressure sensors above propellers. During hydrodynamic tests, propeller excitations may excite the ship model's natural frequencies, and the resulting deformations affect the measured pressure values. This situation has been highlighted by ITTC cavitation committees, such as the committees of the 19th and 20th ITTCs (ITTC, 1990; 1993), who acknowledged that the pressures measured by sensors in model- and full-scale experiments are affected by structural vibratory responses. To consider this, the specialist committee on cavitation-induced pressures of the 22nd and 23rd ITTCs (ITTC, 1999; 2002) recommended measuring structural response in addition to propeller-induced pressures.

### 1.2.2 Effect of model structural vibrations on propeller-induced pressures

Alves Pereira et al. (2024a; 2024b) presented general recommendations to mitigate the impact of model-scale structural dynamics on the measured propeller-induced hull pressures. In particular, they recommended using stiff models, positioning transducers close to propellers, and using accelerometers in proximity to pressure transducers to account for model dynamics.

Other researchers also highlighted similar issues. Tani et al. (2017) performed a series of tests in a cavitation tunnel to measure the effects of cavitation on noise levels and pressure pulses. Their experiments included pressure measurements using sensors installed on a rectangular plate atop a propeller. While they acknowledged the effect of the plate's resonance properties on the measurements, these structural vibrations were not measured directly. Zhang et al. (2021) performed a numerical study to simulate the fluid-structure interaction (FSI) between the pres-

sures generated by a propeller and a rectangular plate placed above the propeller. They employed the DDES and finite volume stress models to simulate the fluid and structure, respectively. In addition to performing CFD analysis, they studied plate vibrations due to propeller excitations. Their results highlighted the importance of the plate's vibrations on the calculated pressures.

Generally, hydrodynamic tests do not consider the effect of the structural dynamics of models. However, in specific tests, the structural dynamics of models are taken into account to study their structural response to external dynamic loads, i.e., wave and slamming, in terms of springing and whipping. For instance, Houtani et al. (2018) and Maron and Kapsenberg (2014) developed hydroelastic designs for ship models to assess their response to waves. In other studies, Kim et al. (2016) and Chen et al. (2019) studied the structural responses of segmented models to oblique waves. Liang et al. (2015) performed a series of experimental tests to determine the structural dynamics of a simplified ship model's structure made of steel. They tested the model under dry and wet conditions in different loading conditions and employed the experimental data to validate numerical simulations.

Regarding hydrodynamic tests for the assessment of propeller-induced hull pressures, van Wijngaarden (2011) conducted experimental hydrodynamic studies on model-scale propeller-induced pressures in a depressurized towing tank by following the procedures presented in the ITTC committee's recommendations (ITTC, 2002; 2008). Van Wijngaarden assessed the pressures generated by model structural vibrations, namely, vibration-induced pressures (VIP), by measuring the aftship response to shaker and propeller excitations. By using data from an array of accelerometers, as mentioned in the study of Bosschers (2018b), and two in-house boundary element codes (PROCAL and EXCALIBUR), van Wijngaarden calculated VIP values in terms of acoustic pressures. His study defined a procedure for estimating propeller-induced pressures, which are discussed in ITTC guidelines. van Wijngaarden estimated the effect of the VIP on the normalized first harmonic of the blade passing frequency (BPF) to be 34% and 17% at two different shaft speeds for a single propeller mounted on a large container ship model.

To our knowledge, van Wijngaarden's work is the first to provide in-depth information on the effect of the structural vibratory behavior of model-scale ships used in towing tank measurements. Although his study was based on an extensive experimental measurement campaign, his VIP results cannot be extended past a limited number of design configurations.

### 1.3 Distinctive features of this study

Experimental studies can be conducted on a limited number of cases, and the quality of their results is affected by

measurement accuracy, environmental conditions, and experimental errors. Herein, we propose an approach that combines numerical simulations and experimental tests to consider the dynamic behavior of the structures of ship models and thus overcome the above limitations.

This paper presents a part of an extensive study on model-scale tests for estimating noncavitating propeller-induced underwater noise and onboard pressures, taking into account the model's structural vibrations. The purpose of our specific research includes the development of a procedure to assess the errors caused by a scaled model's structural vibratory response to propeller-induced pressures.

The methodology proposed in this paper provides the designer with characterized numerical models of the scale model ships used in towing tests that span over multiple design configurations of propellers and their induced pressures—excitations. This methodology, as described in Section 2, tunes the properties of a simple FE model of our complex nonhomogeneous scale model ship on the basis of the resonance characteristics of the physical model.

By elaborating upon van Wijngaarden's study (2011) on considering the effect of model structural vibrations on propeller-induced pressures, we developed an FE model of the scale ship structure as an alternative method to predict its structural dynamics. We updated the model's properties based on the results obtained through experimental modal analysis (EMA). As a more effective approach than experimental measurements, this updated FE model can provide an estimate of the impact of hydrodynamic forces acting on the hull structure and its vibratory response.

By following the proposed methodology, the updated numerical model accurately represents the physical model (the scale model ship) for linear dynamic analyses. Its computational efficiency facilitates the simulation of the dynamic behavior of such a complex structure under various dynamic loads. The proposed approach has the potential to surpass experimental testing in studying the dynamics of scale model ship structures. It is also advantageous when considering the effect of the surrounding water's added mass using vibroacoustic analysis.

## 2 Methodology

The flowchart in Figure 1 shows the methodology developed and implemented in the present research. First, we defined the ranges of factors and the number of experiments to create our design table. We defined the design table, which was composed of sets of Young's moduli, by using latin hypercube sampling (LHS) in the design space. The values of these data points were then inserted into the FE simulation of the dynamic response of the scale model ship hull. From the results, we extracted the values of natural frequencies and normal modes and compared them with

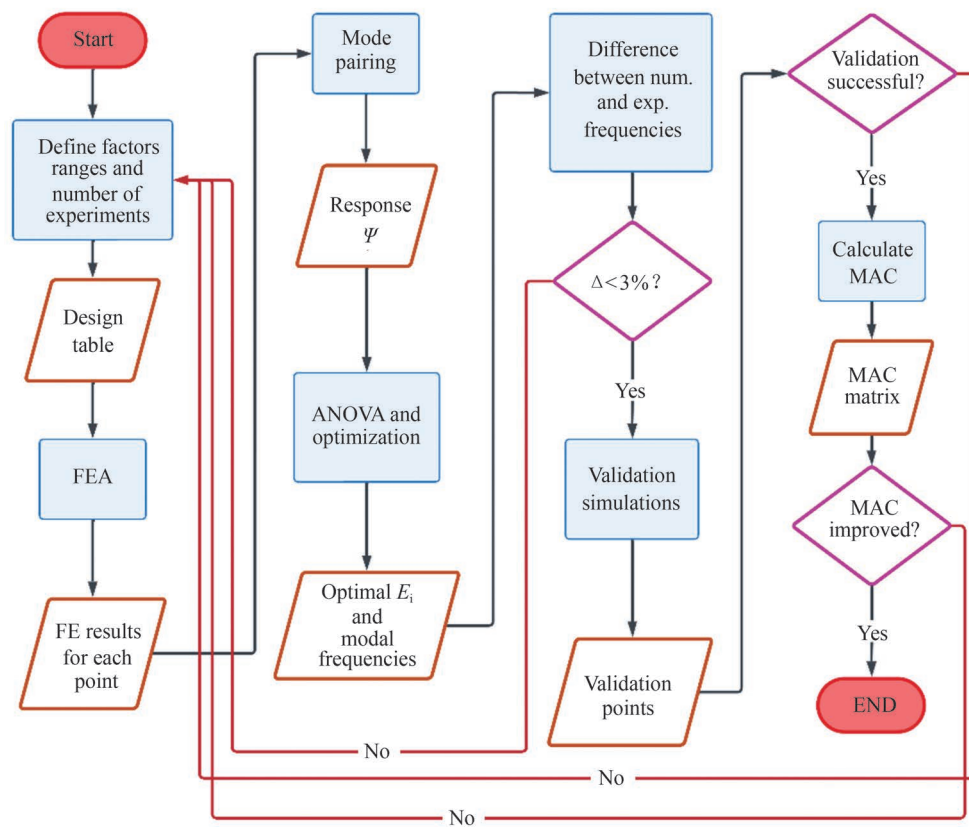


**Figure 1** Flow chart of modal compatibility and optimization to identify updated material properties

experimental modes using the procedure described in Section 2.4. For each datapoint, we estimated the value of the performance measure  $\psi$ , the response in the design table. We then used the values of the factors and responses to build the analysis of variance (ANOVA) table of this design. Once the assumptions of ANOVA were verified, we optimized the numerical Young's moduli to maximize the fit of the numerical model to the experimental data. Furthermore, we calculated the natural frequencies and normal modes of the optimal points. For this configuration, we calculated the relative difference between numerical and experimental frequencies. If this difference was less than 3%, then the optimization procedure was satisfactory. Moreover, in order to check for the accuracy of the statistical predictive model we built using ANOVA, we compared the predicted response of the model with the corresponding value obtained with the FE model in three validation points. If this second check was also successful, then we validated the response surface. Lastly, we calculated the modal assurance criterion (MAC) matrix between experimental modes and numerical modes of the optimized model, and we verified that the MAC values had improved compared with the values of our initial configuration.

The specification of the scale model ship is outlined in Section 2.1. The EMA for identifying natural frequencies and normal modes experimentally is described in Section

2.2. The numerical simulations are explained in Section 2.3. Finally, the details of model updating, including mode pairing and objective function definition, are presented in Section 2.4.

## 2.1 Scale model ship

### 2.1.1 Specification

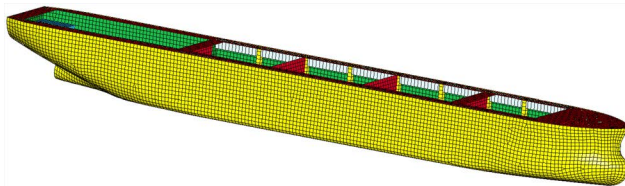
The scale model ship is a model of a post-Panamax bulk carrier constructed with different materials, such as fiber reinforced plastic (FRP), plywood plates, wooden beams, foam, synthetic polyurethane, and brass. The dimensions of the scale model and full-scale ship are outlined in Table 1. Table 2 lists the model's specifications, and Figure 2 illustrates its overall geometry.

**Table 1** Dimensions of the Scale model ship and real ship

| Dimensions         | Model | Ship   |
|--------------------|-------|--------|
| Overall length (m) | 7.142 | 226    |
| Breadth (m)        | 1.020 | 32.25  |
| Depth (m)          | 0.623 | 19.75  |
| Draft (m)          | 0.430 | 13.50  |
| Displacement (t)   | 2.685 | 85 637 |

**Table 2** Materials used in the model hull

| Type of material                | Location                                                |
|---------------------------------|---------------------------------------------------------|
| FRP                             | Outer shell and deck                                    |
| Spruce, pine, fir (SPF) plywood | Inner plating                                           |
| Brass                           | Propeller, strut, and small parts in the bottom and bow |
| RenShape 440 polyurethane board | Local plating on top of the propeller                   |
| Hi-60 foam                      | Filling between the outer shell and inner plating       |

**Figure 2** Ship model hull meshed using shell and beam elements

### 2.1.2 Geometry modeling and materials

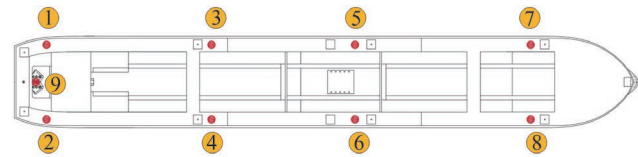
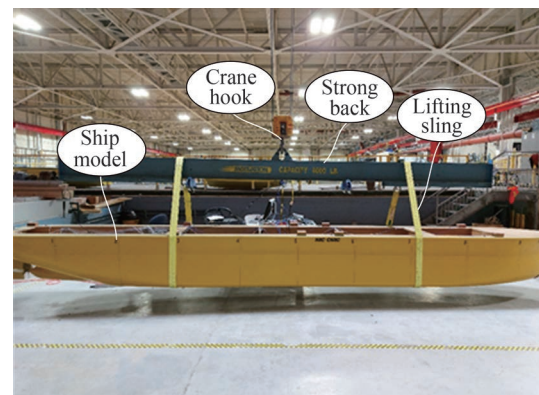
The model consisted of the outer hull and deck FRP plating, inner plywood structure, and internal wood framing. These components provide the greatest contribution to the global stiffness of the hull. Linear elastic material models were used due to the limited range of displacements and deformations of the structural elements in response to propeller excitations. Foam has a markedly lower modulus of elasticity than FRP and plywood (Abbood et al., 2021; Komaki and Kuroda, 1972; Cai and Robert, 2006); however, its role as the connector of the outer and inner layers is not negligible. Therefore, we distributed springs between the inner and outer layers and compensated for the foam's mass by distributing nodal mass along the model to balance the mass of the numerical and real models. The details of the FE simulations are described in Section 2.3.

## 2.2 Experimental procedure

### 2.2.1 Experimental modal analysis

A single-input multi-output (SIMO) EMA was conducted by using a PCB® piezoelectric hammer (model 086C03) as the exciter and nine PCB® piezoelectric accelerometers (model 352C33) on the scale model ship's deck to measure the structural response. Data were collected at a sampling frequency of 5 kHz by using three NI DAQ cards (type 9230) and processed to calculate acceleration frequency response functions (FRFs). Three impacts were applied to average FRFs to mitigate the effect of noise and improve measurement quality. Postprocessing in MATLAB® yielded the normal mode vectors and coherence functions calculated to assess the correlation between input and output signals (Section 3.1). The model was suspended from a

crane by using a strong bac® lifting beam and two lifting slings to create free-free boundary conditions for the scale model ship. Figure 3 illustrates the arrangement and numbering of accelerometers along the ship length, and Figure 4 shows the model hanging arrangement.

**Figure 3** Arrangement and numbering of accelerometers along the ship length**Figure 4** Side view of the model hanging arrangement for the EMA hammer test

### 2.2.2 Design of experiments

The optimal values of the FE model's elastic moduli were calculated by optimizing the prediction performance of the experiment. RSM, as part of design of experiments (DOE) mathematical techniques, helps predict system behavior through mathematical models within a defined design space. Eq. (1) formulates the relationship between the response and input factors in RSM, where  $\eta$  is the response,  $f$  is the true response function,  $X_i$  represents the independent variables (factors), and  $\varepsilon$  is the residual and accounts for the source of variability that is not represented by  $f$  (Myers et al., 2016).

$$\eta = f(X_1, \dots, X_k) + \varepsilon \quad (1)$$

For the current analysis,  $X_i$  represents the values of the elastic moduli in Eq. (3) used as the inputs of the FE model, and  $\eta$  is the objective function generated by the difference between experimental and numerical natural frequencies. After we estimated  $f$  through data fitting using RSM, we minimized the response function (objective function) to find the optimized values of the elastic moduli.

#### • RSM model and optimization

In this study, the quadratic model represented by Eq. (2)

was used to predict the function  $f$ . The quadratic model has a versatile structure that captures the behavior of various functions. The model's coefficients can be estimated by employing least-squares regression. This approach makes the model flexible, easy to compute, and suitable for the specific problem at hand.

$$\eta = \beta_0 + \sum_{i=1}^k \beta_i x_i + \sum_{i=1}^k \beta_{ii} x_i^2 + \sum_{1 \leq i < j \leq k} \beta_{ij} x_i x_j + \varepsilon \quad (2)$$

As a result of the simple and flexible structure of the predictive model, RSM is one of the most commonly used techniques in the optimization of computer experiments and the most reliable in the case of small sample sizes (Chen et al., 2006).

- Sampling and statistical validation

We built the design table by employing LHS. LHS, originally presented by McKay et al. (1979), is one of the most popular sampling techniques for computer experiments (Viana, 2016). Its widespread use originates from its ability to produce datasets for experiments on the basis of different statistical assumptions, its easy generalizability to design spaces of any size, and its non-collapsing nature—the design remains a latin hypercube even if some dimensions of the design space are removed (Pronzato and Müller, 2012; Viana, 2016).

By implementing LHS, we sampled through the preliminary ranges outlined in Table 3 and created 20 sets of elastic moduli, which were then utilized as inputs for the FE model to calculate numerical natural frequencies and normal modes. The values of elastic moduli were stored in the parameter vector  $\mathbf{E}$  (Eq. (3)), where  $E_p$ ,  $E_w$ , and  $E_f$  are the equivalent linear elastic (Young's) moduli for the plywood plates, wooden beams, and FRP surfaces, respectively.

$$\mathbf{E} = [E_p \ E_w \ E_f] \quad (3)$$

**Table 3** Range of the elastic moduli of the materials used in the model

| Elastic modulus (GPa) | Min | Max |
|-----------------------|-----|-----|
| Plywood ( $E_p$ )     | 3   | 6   |
| Wood ( $E_w$ )        | 6   | 10  |
| FRP ( $E_f$ )         | 10  | 20  |

## 2.3 FE simulation

### 2.3.1 FE elements

The FE mesh was created in HYPERMESH<sup>®</sup> by using quadrilateral reduced integration shell elements, which are known as Belytshko–Lin–Tsay elements, for the inner and outer plating and beam elements for the wooden framing. Table 4 outlines the thickness of the FRP and plywood sur-

faces, along with the cross-sectional dimensions of the beam elements. We considered linear elastic material properties for the model and performed mesh convergence analysis to find the optimal element size, considering the study conducted by Moro et al. (Moro et al., 2013), leading to a model composed of 22 513 shell and beam elements. The numerical model is depicted in Figure 2.

**Table 4** Constant dimensions of structural elements

| Structural component | Parameter                        | Value   |
|----------------------|----------------------------------|---------|
| FRP layer hull       | Thickness (mm)                   | 6       |
| FRP layer deck       | Thickness (mm)                   | 7       |
| Plywood layer        | Thickness (mm)                   | 10      |
| Beam element I       | Cross section (mm <sup>2</sup> ) | 95 × 45 |
| Beam element II      | Cross section (mm <sup>2</sup> ) | 50 × 45 |

As discussed in Section 2.1.2, the foam filling was represented with spring elements instead of solid elements. The value of spring stiffness was evaluated by matching the shape of the first torsional and bending modes in experimental and numerical results in a preliminary analysis and was not a factor in the optimization design. The use of spring elements to simulate the role of the foam core between the inner and outer hulls provides an effective approximation of the scale model ship's global structural behavior. This simplification was adopted to simulate the foam's contribution to the global stiffness of the model and ensure computational efficiency. Although this modelization cannot accurately represent a multidimensional state of stress, this shortcoming is not relevant to our linear dynamic study justified by small deformations.

The total physical mass of the scaled model was measured to be approximately 425 kg, and the FE model was calibrated to match this value. In particular, the foam core was represented by using spring elements to capture its stiffness contribution. Although these elements are massless, the actual foam mass—120 kg—was distributed along the model. This approach ensures that the FE model accurately reflects the overall mass and inertia of the system without incurring the modeling and computational costs associated with explicitly modeling the foam by employing solid elements.

### 2.3.2 FE analyses

FE simulations were performed by using LS-DYNA<sup>®</sup> under a free–free boundary condition. The main outputs of the FE simulation are the first three global natural frequencies (eigenvalues), normal modes (eigenvectors), and relevant accelerance FRFs, which were calculated as described in Section 3.2.

## 2.4 Model updating

### 2.4.1 Review on the model updating procedure

Model updating is a procedure for identifying the optimal values of the parameters of a numerical model based on fitting experimental data. A paper published by Sehgal and Kumar (2016) provides an extensive literature review of the techniques for updating structural dynamic numerical models.

The performance function used to update the properties of the numerical model compares the behavior of the numerical model with the response measured experimentally. Generally, it is defined on the basis of the response at the natural frequency, the shape of the normal modes, or a measure of fit of the transfer functions of the numerical and experimental models. Once a performance function is selected, several optimization techniques can be deployed to identify the parameter values of the numerical model that optimize the performance function over the design space. A set of numerical simulations is conducted to explore the design space. Therefore, the main features of model updating are the design space, selected performance measures, and optimization methodology.

Adel et al. (2017) created a numerical model of a bolted joint connecting an aluminum bar to a composite bar. They used a genetic algorithm (GA) to minimize the objective function defined based on the squared differences between the first four numerical and experimental natural frequencies. They selected the Young's modulus of the aluminum bar and the directional Young's moduli of the composite bar as their optimization parameters. Ribeiro et al. (2012) used the GA to calibrate the FE model of a bowstring-arch railway bridge. The objective function was defined based on the difference between experimental and numerical natural frequencies and MAC. Mansour et al. (2016) investigated the dynamic characteristics of epoxy resins reinforced with rubber. By using tensile and hammer tests, they characterized the mechanical behavior of the composite materials and conducted dynamic tests on a cantilever configuration. By employing GA, they optimized an objective function based on experimental and analytical transfer functions. Ferrari et al. (2019) investigated the dynamic response of a historic reinforced concrete bridge. The structure was modeled by using beam elements, and Young's moduli and material densities were defined as optimization parameters. The experimental response was measured through operational modal analysis. The objective function comprised the ratios between numerical and experimental natural frequencies and the differences between numerical and experimental normal modes. Modal compatibility was assured by calculating MAC values. Cuadrado et al. (2019) updated the FE models of carbon/epoxy composite plates. They found the plates' natural frequencies and mode shapes by using a multiple-input single-output

(MISO) EMA, wherein a roving hammer provided the input force, and the response was measured with a single mono-axial accelerometer. The objective function was defined as the sum of the differences between experimental and numerical natural frequencies, and a Monte Carlo approach was implemented for the optimization. Among the factors of the experiment, the plate's elastic modulus, density, and thickness were the most relevant. Fragasso et al. analyzed the quasi-static (Fragasso et al., 2019) and harmonic (Fragasso and Moro, 2022) response of resilient mounts for marine diesel engines. The static analysis was based on the fitting of the quasi-static compression curve of the element. The factors of the experiment were the Yeoh coefficients of the rubber material. The fit was measured by using the Nash–Sutcliffe efficiency index. In the dynamic analysis, the numerical transmissibility frequency response function was fitted to the experimental data by updating the values of the frequency-dependent damping ratios. ANOVA was used to find the most relevant factors and optimize the fit of the numerical model.

Our analysis focused only on the first modes of vibration. We used simplified element models (shell, beam, and spring elements) to represent the constituents of the FE model with a limited range of motion. We applied a SIMO EMA to measure structural accelerations simultaneously, facilitating signal synchronization by using our data acquisition system. This approach overcomes the synchronization challenges when conducting a MISO EMA, such as the work done by Cuadrado et al. (2019). Similar studies in this field (Adel et al., 2017; Ribeiro et al., 2012; Cuadrado et al., 2019) leveraged the GA and Monte Carlo optimization to fit the dynamic characteristics of the numerical model. In this study, we utilized an optimization methodology based on LHS and RSM. Although this procedure requires building and validating a statistical model to predict the fitting performance of the FE model, it can provide accurate results with a limited number of experiments. Moreover, the use of LHS in our work is consistent with that of previous studies on updating models, such as the bridge model studied by Ferrari et al. (2019). The objective function that we chose is similar to that employed in other studies (Adel et al., 2017; Cuadrado et al., 2019). This function allowed us to consider different modes of vibration simultaneously and obtain a measure of fit of the natural frequencies and normal modes of the optimized numerical model. Section 2.4.2 describes the mode pairing methodology used in this paper, including the objective function established based on numerical and experimental data.

### 2.4.2 Mode pairing and objective function

The objective function  $\psi$  presented in Eq. (4) helped compare numerical and experimental natural frequencies. This function is defined as the sum of the normalized differences between experimental and numerical natural frequencies, where  $n$  is the number of modes considered for the mode

pairing and model update,  $f_i^e$  is the  $i$ th mode natural frequency obtained from signal analysis in EMA,  $f_i^a$  is the  $i$ th mode natural frequency obtained from FE simulation, and  $w_i$  is the modal weighting factor that can be used to focus the analysis on a specific frequency range. We used equal weights to balance the importance of each natural frequency in the optimization.

$$\psi = \sum_{i=1}^n w_i \left| \frac{f_i^e - f_i^a}{f_i^e} \right| \quad (4)$$

Moreover, MAC matrices (scalar correlation between experimental and numerical modal vectors) were calculated by using Eq. (5). The value  $MAC_{ij}$  is the MAC matrix element calculated for the  $i$ th and  $j$ th modes, and  $\phi_i$  and  $\phi_j$  are the modal vectors (Allemang, 2003; Pastor et al., 2012).

$$MAC_{ij} = \frac{|\phi_i^T \phi_j|^2}{(\phi_i^T \phi_i)(\phi_j^T \phi_j)} \quad (5)$$

### 3 Results and discussion

#### 3.1 Experimental results

Figure 5 shows the accelerance FRF amplitude and phase plots acquired during the impact tests. Natural frequencies were identified by selecting the peak positions of the FRF amplitude plots. The natural frequencies of the suspension system manifest as rigid body motion modes. The same phase change trend at the position of the amplitude peaks indicates potential rigid body modes, including the first translational (heave) and rotational (pitch) rigid body modes at 8.47 and 18.94 Hz, respectively. The first three natural frequencies of the scale model ship, listed in Table 5, were determined by following the same approach. For clarity, the FRF plots indicate the rigid body (R), torsional (T), and bending (B) modes.

The first torsional mode (occurring at 17.31 Hz) is close to the pitch rigid body mode at 18.94 Hz, proving the importance of finding the mode shapes in addition to only natural frequencies by conducting further signal processing. In Figures 6 and 7, the normal and rigid body mode shapes are illustrated by plotting the vectors of acceleration amplitudes. The  $x$  and  $y$  coordinates of the plots represent the longitudinal and transverse positions of the global accelerometers on the deck with zero  $y$  at the centerline, whereas the  $z$  coordinate represents the model's depth.

Three normal and two rigid body modes, obtained through EMA and illustrated in Figures 6 and 7, lie in the frequency range of interest. This range includes frequencies less than 100 Hz based on the propeller excitation frequencies—the multiplication of the number of blades and

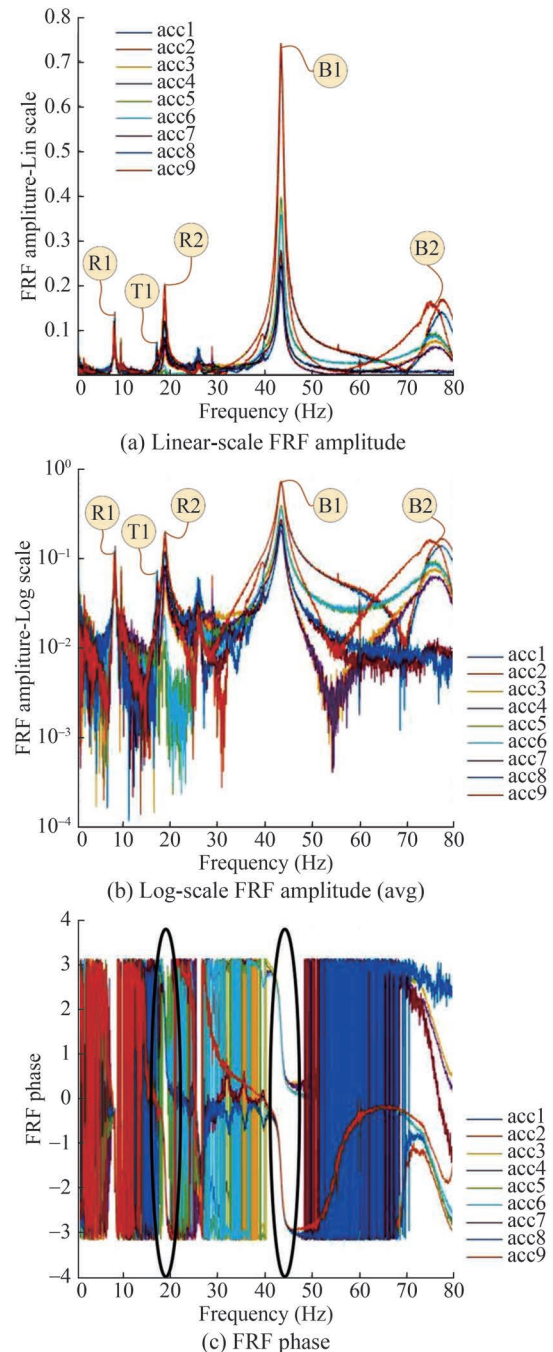


**Figure 5** FRFs (accelerance) obtained from EMA

**Table 5** Natural frequencies of the scale model ship model

| Mode number | Natural frequency (Hz) | Mode type       |
|-------------|------------------------|-----------------|
| 1           | 17.31                  | First torsional |
| 2           | 43.48                  | First bending   |
| 3           | 76.80                  | Second bending  |

shaft rotation speeds during towing tests.

Figure 8 depicts the FRF amplitude, phase, and coherence plots for eight accelerometers, with the ninth accel-

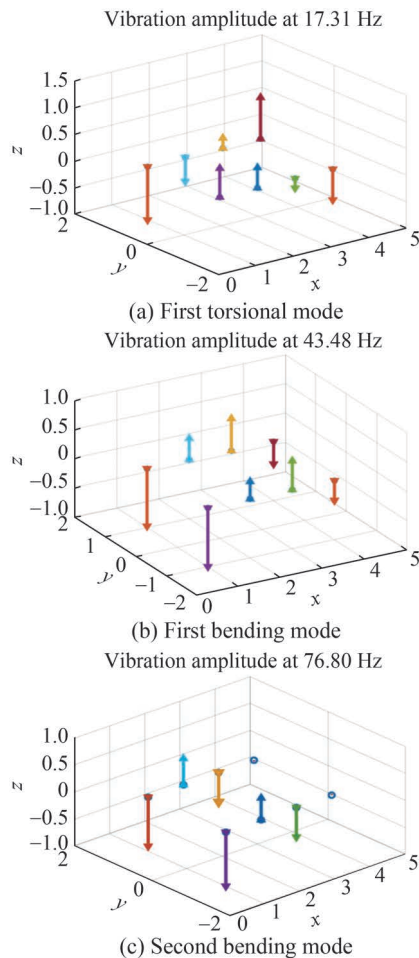


**Figure 6** Amplitude of deformation at the position of accelerometers for the first three normal modes obtained from EMA

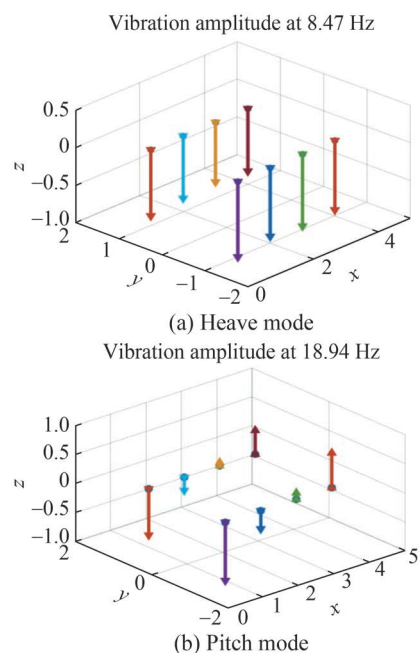


**Figure 7** Amplitude of deformation at the position of accelerometers for the rigid body motion modes from EMA

ometer positioned close to the hammer hit serving for control and reference. Coherence functions are plotted for each FRF, demonstrating acceptable results and effective noise control. Although some points in the plots have low coherence values, modal points remain acceptable, such as most of the frequency range of interest. Coherence values above 0.8 are generally considered acceptable in EMA (Ewins, 2009; Inman and Singh, 2014). In our tests, the coherence remained above this threshold across most of the frequency range, particularly at the natural frequencies. Low coherence values in the low-frequency range were not considered relevant to the modal analysis because we expected the dynamics of the model to be unaffected. Although vertical bending and torsional modes were accurately obtained through EMA, experimental limitations led to the inability to detect lateral bending modes due to monoaxial accelerometer placement. The monoaxial accelerometers installed on the deck can measure accelerations only in the vertical direction. The installation direction was selected to capture the vertical deformations of the model in the same direction as the vertical excitation from propeller pulsations.

### 3.2 Numerical results

Figure 9 displays the first three normal mode shapes of the scale model ship obtained using finite element analysis (FEA) with the preliminary material properties listed in Table 6. Figure 10 illustrates the FRF plots between the eight acceleration signals and impact hammer force. The FE outcomes based on these initial estimates are presented in Table 7, which shows the first torsional, first lateral bending, and first bending modes.

The FE-obtained natural frequencies, outlined in Table 7, with the relevant mode shapes illustrated in Figure 9, lie in the frequency range of interest mentioned in Section 3.1. Furthermore, the damping neglected in the FE model resulted in sharp FRFs, with the phase angle plots exhibiting abrupt fluctuations between  $0^\circ$  and  $-180^\circ$  in contrast to the smooth EMA plots, as depicted in Figure 10. However, it had an insignificant effect on the final calculated natural frequencies and normal modes, given that the updated model proved to be acceptable, as presented in Section 3.5.

### 3.3 Design specifications and statistical data fitting

#### 3.3.1 Table of design

Table 8 contains the design table and relevant objective function ( $\psi$ ) values as the inputs for the optimization based on 20 samples. The number of samples was obtained based on the procedure explained in Figure 1 after comparing the capability of the statistical model for designs with few samples. Figure 11 shows the LHS points in the 3D space made of different elastic moduli, proving the LHS coverage of the internal cube space in sampling.

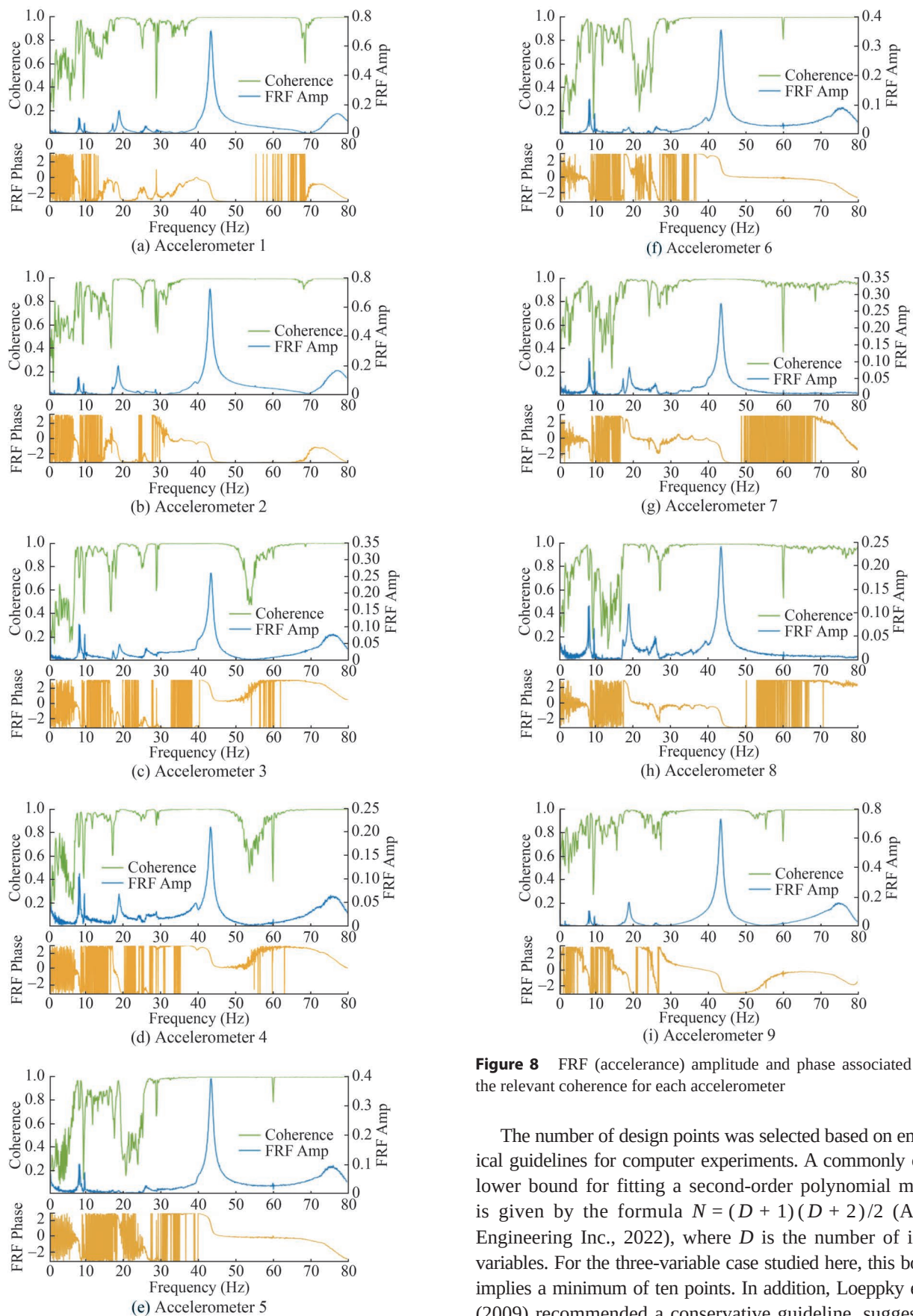


**Figure 8** FRF (accelerance) amplitude and phase associated with the relevant coherence for each accelerometer

The number of design points was selected based on empirical guidelines for computer experiments. A commonly cited lower bound for fitting a second-order polynomial model is given by the formula  $N = (D + 1)(D + 2)/2$  (Altair Engineering Inc., 2022), where  $D$  is the number of input variables. For the three-variable case studied here, this bound implies a minimum of ten points. In addition, Loeppky et al. (2009) recommended a conservative guideline, suggesting

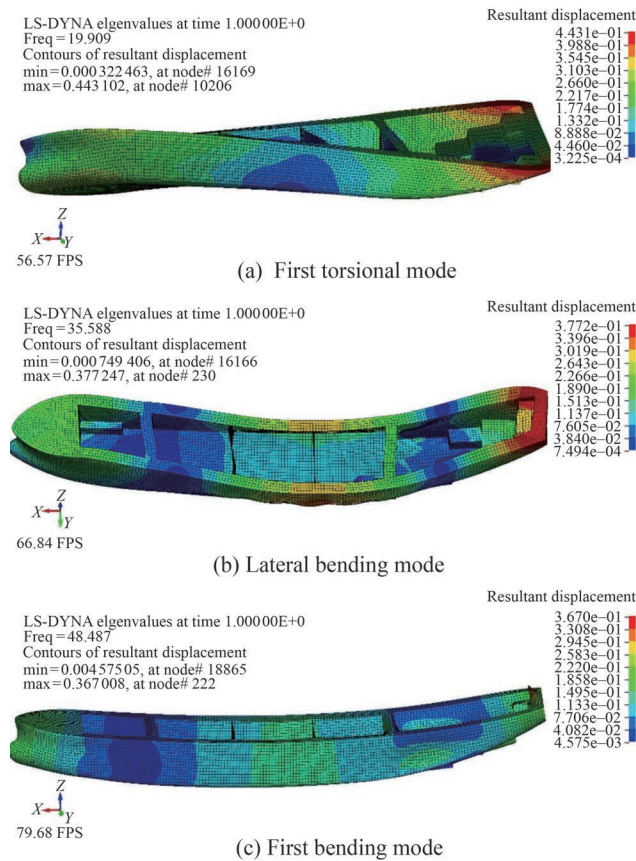


**Figure 9** First three normal modes obtained from FEM

**Table 6** Preliminary estimate of the elastic moduli of the materials used in the FEM model

| Elastic modulus (GPa) | Value |
|-----------------------|-------|
| Plywood ( $E_p$ )     | 5     |
| Wood ( $E_w$ )        | 9     |
| FRP ( $E_f$ )         | 20    |

**Table 7** Natural frequencies of the scale model ship from FEM simulation

| Mode number | Natural frequency (Hz) | Type of mode    |
|-------------|------------------------|-----------------|
| 1           | 19.91                  | First torsional |
| 2           | 35.59                  | Lateral bending |
| 3           | 48.49                  | First bending   |

that approximately  $N = 10D$  points may be necessary to explore the design space adequately, particularly in the presence of nonlinearities. Our use of 20 LHS points exceeded the polynomial fitting requirement and fell slightly below Loepky's conservative estimate. However, validation results demonstrated that the model's predictive performance was stable and accurate within the selected design space, supporting the adequacy of the chosen sampling size.

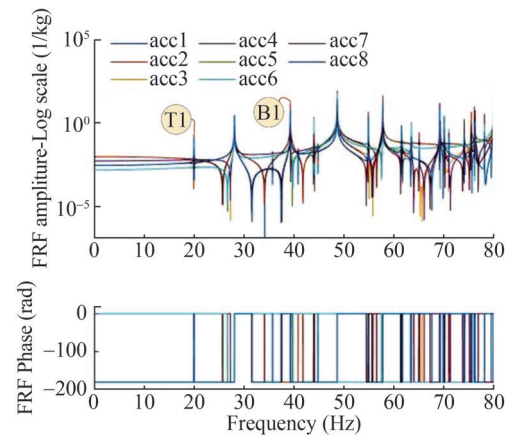


**Figure 10** Accelerance FRF obtained from FEM: log-scale FRF amplitude and FRF phase

**Table 8** Design table (from LHS) and objective function ( $\psi$ )

| Sample number | $E_p$ (A)<br>(GPa) | $E_w$ (B)<br>(GPa) | $E_f$ (C)<br>(GPa) | $\psi$ |
|---------------|--------------------|--------------------|--------------------|--------|
| 1             | 4.75               | 8.44               | 12.74              | 0.0418 |
| 2             | 5.23               | 7.91               | 13.06              | 0.0176 |
| 3             | 4.56               | 9.42               | 11.71              | 0.0823 |
| 4             | 5.96               | 7.12               | 17.92              | 0.1975 |
| 5             | 3.29               | 7.25               | 15.90              | 0.0371 |
| 6             | 5.29               | 6.72               | 18.30              | 0.1737 |
| 7             | 3.41               | 9.64               | 19.13              | 0.1460 |
| 8             | 3.08               | 8.85               | 19.59              | 0.1235 |
| 9             | 4.25               | 7.52               | 11.11              | 0.1682 |
| 10            | 4.00               | 8.76               | 10.84              | 0.1684 |
| 11            | 4.09               | 8.24               | 16.73              | 0.0800 |
| 12            | 4.89               | 9.01               | 10.46              | 0.1354 |
| 13            | 5.45               | 8.13               | 14.94              | 0.0775 |
| 14            | 3.64               | 9.27               | 15.43              | 0.0239 |
| 15            | 5.03               | 7.71               | 18.66              | 0.1943 |
| 16            | 3.90               | 9.83               | 13.75              | 0.0244 |
| 17            | 5.80               | 6.85               | 12.41              | 0.0428 |
| 18            | 5.64               | 6.37               | 16.25              | 0.1026 |
| 19            | 3.56               | 6.43               | 17.43              | 0.0463 |
| 20            | 4.47               | 6.00               | 14.47              | 0.0342 |

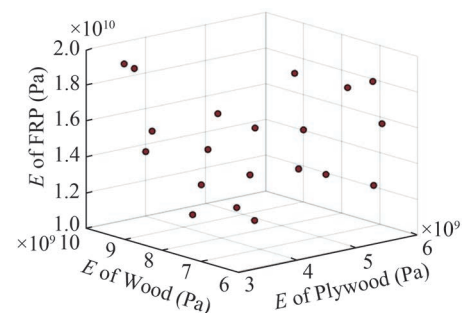


**Figure 11** Representation of the sampled data using the LHS method

### 3.3.2 RSM model assessment

The ANOVA table (Table 9) and statistical predictive model, Eq. (6), were built by using Design-Expert®. As discussed in Section 2.2, a quadratic model was selected. The table contains the  $F$ -values and  $p$ -values indicating the significance of the model factors. The normal plot of residuals, the residuals against factors, and the residuals versus the predicted values are plotted in Figures 12, 13(a)–(c), and 14, respectively.

$$\eta = 3.31747 - 0.241937 E_p - 0.064409 E_w - 0.337141 E_f + 0.016645 E_p E_f + 0.004123 E_w E_f + 0.007851 E_f^2 \quad (6)$$

We constructed an ANOVA table (Table 9) for the chosen predictive model and evaluated the ANOVA assumptions. The  $F$ -value of the model was 47.23, denoting its significance with a low chance (0.01%) of being generated by noise. The terms  $C$  ( $E_f$ ),  $AC$  ( $E_p E_f$ ),  $BC$  ( $E_w E_f$ ), and  $C^2$  ( $E_f^2$ ) were significant in our model. Although the  $p$ -values of  $A$  ( $E_p$ ) and  $B$  ( $E_w$ ) exceeded 0.1, they were used in the model due to the hierarchy of effects. In this case study, the final results indicate that the most important factor in the dynamics of the model is the Young's modulus of FRP. The final results in terms of natural frequencies and normal modes indicate high accuracy, leading to the well-minimized error.

**Table 9** ANOVA table

| Source                   | Sum of squares | DF | Mean square | $F$ -value | $p$ -value | Significance |
|--------------------------|----------------|----|-------------|------------|------------|--------------|
| Model                    | 0.0708         | 6  | 0.0118      | 47.23      | <0.0001    | Significant  |
| A: $E_p$                 | 0.0006         | 1  | 0.0006      | 2.45       | 0.1413     |              |
| B: $E_w$                 | 0.0001         | 1  | 0.0001      | 0.4344     | 0.5213     |              |
| C: $E_f$                 | 0.0056         | 1  | 0.0056      | 22.36      | 0.0004     |              |
| AC: $E_p \times E_f$     | 0.0162         | 1  | 0.0162      | 64.84      | <0.0001    |              |
| BC: $E_w \times E_f$     | 0.0017         | 1  | 0.0017      | 6.79       | 0.0218     |              |
| C <sup>2</sup> : $E_w^2$ | 0.0514         | 1  | 0.0514      | 205.54     | <0.0001    |              |
| Residual                 | 0.0033         | 13 | 0.0003      | —          | —          |              |
| Cor total                | 0.0741         | 19 | —           | —          | —          |              |

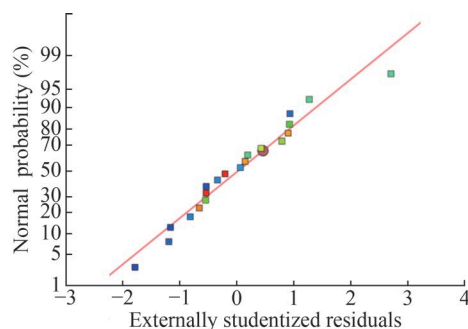


**Figure 12** Normal plot of residuals

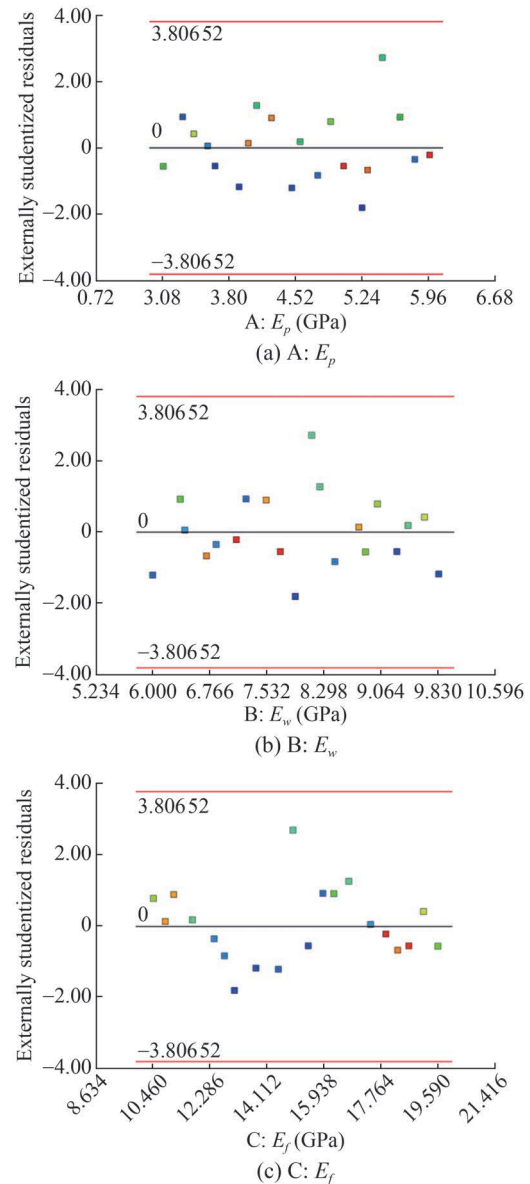


**Figure 13** Residuals against factors

Figure 12 shows the normal plot of residuals. The fitted line proved the normal distribution of these residuals. The residuals were plotted against the factors in Figure 13. Figure 14 indicates that these residuals were structureless by plotting them versus the predicted values. Furthermore, due to the nature of the FE analysis, the results of each FE run were independent of the others. Therefore, the three ANOVA assumptions (Montgomery, 2017) were satisfied.

Finally, the predictivity of the statistical model was assessed by examining the results for the validation points listed in Table 10. This validation compared the statistical function ( $\eta$ ) outcomes with the objective function ( $\psi$ ) results. We tested our predictive model on three points, obtaining satisfactory results.

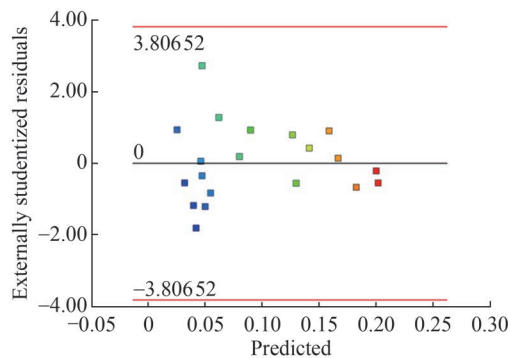


**Figure 14** Residuals versus predicted values

**Table 10** Validation of  $\eta$  based on  $\psi$

| Sample number | $E_p$ (A) (GPa) | $E_w$ (B) (GPa) | $E_f$ (C) (GPa) | $\psi$   | $\eta$   |
|---------------|-----------------|-----------------|-----------------|----------|----------|
| 1             | 3.5             | 9               | 12              | 0.141070 | 0.120235 |
| 2             | 4.5             | 8               | 14              | 0.012077 | 0.042714 |
| 3             | 5.5             | 7               | 16              | 0.099757 | 0.078089 |

### 3.4 Mode pairing results (MAC matrix)

MAC matrices were calculated between the numerical and experimental results before and after optimization, based on the first torsional and first bending modes, as illustrated in Figures 15(a) and (b). Modal compatibility was guaranteed with the close-to-one values of the MAC diagonal components and trivial off-diagonal components before and after optimization, as depicted in Figure 15. The optimization procedure improved the matching in the diagonal elements and reduced the values of the off-diagonal elements.

### 3.5 Model update results

After the use of LHS and RSM, the natural frequencies obtained through experimental and numerical analyses showed improved compatibility, as shown in Table 11. The error values, defined as the relative difference between the FE and EMA modal frequencies, decreased from 15.0% and 11.5% for the first torsional and first bending modes to 0.69% and 1.15%, respectively. The final optimized values for elastic moduli are presented in Table 12. The values obtained cleared the 3% threshold that we set.

### 3.6 Strengths and limitations of this study and future works

The methodology provided herein presents an updated FE model that simulates the dynamic behavior of the complex nonhomogeneous scale model ship. This numerical model proved highly beneficial for further studies on the dynamic response of the scale model, such as investiga-

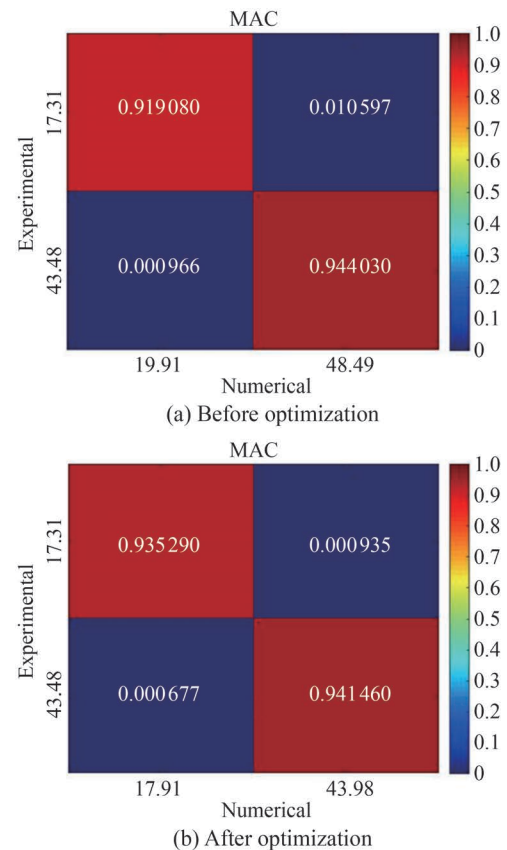


**Figure 15** MAC matrix between the EMA and FEM

**Table 11** Natural frequencies of the model before and after optimization

| Type of result                | First torsion | First bending |
|-------------------------------|---------------|---------------|
| Experimental                  | 17.31         | 43.48         |
| Numerical before optimization | 19.91         | 48.49         |
| Error (%)                     | 15.0          | 11.5          |
| Optimized                     | 17.19         | 43.98         |
| Error (%)                     | 0.69          | 1.15          |

**Table 12** Elastic moduli: preliminary estimations and optimized values (GPa)

| Elastic moduli | Preliminary estimate | Optimized |
|----------------|----------------------|-----------|
| $E_p$          | 5                    | 5.92      |
| $E_w$          | 9                    | 9.43      |
| $E_f$          | 20                   | 12.21     |

tions of the effect of added mass on the dynamic characteristics of the model presented by Jahanbakhsh et al. (2024b).

The model's natural frequencies and its resonance behavior during towing tests were used to evaluate the effect of model dynamics on the measurement of propeller-induced

hull pressures. The FE model provides the dry natural frequencies required for the analytical calculation of the vertical added mass of the model based on the method presented by Korotkin (2009). In addition, the structural dynamics that have been calculated for the unballasted ship model can be used to understand how different ballast conditions affect model dynamics during towing tests (Jahanbakhsh et al., 2024a).

The updated FE model can be used for further vibroacoustic simulations, such as frequency response analyses, to account for the effect of various propeller-induced pressures in exciting the resonance frequencies of the model structure. These analyses can be used in support of the traditional towing tank tests and in combination with CFD analyses, as highlighted by Cilkaya et al. (2024), to meet the requirements mentioned in ITTC procedures (Alves Pereira et al., 2024a, 2024b) regarding monitoring the effect of model structural vibrations on propeller-induced pressures.

Optimization was performed within the bounds derived from the documented material property ranges for FRP, wood, and foam, ensuring that the estimated Young's moduli remain physically reasonable. While the FE model does not aim to replicate the full anisotropic or layered behavior of the physical model, it successfully captures the global dynamic response. The proposed approach decouples stiffness from thickness and density variations, enabling efficient and stable model updating. While this simplification may limit the exploration of stiffness–density–thickness interactions, it serves the intended purpose of computationally efficient dynamic modeling. Future studies may extend the model to incorporate geometric or density-based optimization for enhanced sensitivity and physical interpretability.

Regarding the limitations of this study, we focused only on the vertical modes of the scale model ship and neglected its lateral modes. This decision was made according to the vertical nature of propeller excitations in addition to the limitations in the type of accelerometers at hand. If taking into account the lateral modes in the model updating is relevant, then this need can be addressed by using triaxial accelerometers or different acceleration arrangements during the EMA.

The EMA was also conducted in water to account for the added mass effect of the surrounding water; however, considering facility restrictions, we obtained noisy results, as proven by low coherence values. Future studies may explore the added mass effect during the EMA by providing sufficient facilities to mitigate the noise in wet hammer tests.

Determining the appropriate modal damping for such composite structures is another topic for further investigation.

Finally, our study does not aim to replicate the detailed ply-level stress–strain behavior of composite structures but rather focuses on capturing the effective global response through simplified elements. The FE model approximates the overall laminate behavior by using homogenized mate-

rial properties rather than explicit laminate stiffness matrix derivations. While classical lamination theory (CLT) provides a rigorous tool for such analyses, it was not applied herein due to the lack of detailed ply information and the focus on computationally efficient modeling for global dynamic assessment. Future work may include CLT-based validation and stiffness matrix comparisons when detailed composite specifications are available.

## 4 Conclusions

The main purpose of our study is to provide FE models of the scale model ships used in towing tank experiments to consider the impact of the scale models' structural vibrations on propeller-induced pressures and address the recommendations from the specialist committee on cavitation and noise of the 30th ITTC (Alves Pereira et al., 2024a, 2024b). Herein, we propose a procedure to update the FE model of our base scale model ship by fitting the dynamic response of the numerically simulated model to the data acquired experimentally through EMA. We used RSM to fit the modal response of the simulation to experimentally measured modes. The high precision of the results is proven by the improvements in MAC matrix values and the trivial difference between the frequencies of the first two mode shapes. The proposed model updating procedure provides researchers with a practical approach to create simplified and reliable FE models of the scale model ships that can accurately simulate the dynamics of ship models.

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**Competing interest** Lorenzo Moro is an editorial board member for the Journal of Marine Science and Application and was not involved in the editorial review, or the decision to publish this article. All authors declare that there are no other competing interests.

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