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RESEARCH ARTICLE

Application of Modal Analysis for the Vibrational Comfort Investigation on Board Ships: a Case Study

Giovanni Rognoni[1], Pranas Mažeika[2], Giada Kyaw Oo D'Amore[1], Emanuele Brocco[3], Rima Mickevičienė[2], Tomas Žapnickas[2], Evgenii Iamshchikov[2], Vasilij Djačkov[2] and Marco Biot[1]

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Abstract

The design of a ship is a process facilitated by different parallel departments. Specialists from various disciplines jointly work on a project, eventually covering the entire process. Though simultaneously, these disciplines are often subject to a hierarchy, either clearly defined or dictated by necessity. Within these branches, despite a growing interest in enhancing the comfort on board ships, noise and vibration design is among the most sacrificed. Compared to hydrodynamic or structural modifications, efforts devoted to improving vibrational comfort are generally slightly impactful and costly. Consequently, these improvements are often relegated to the final stages of the design procedure. The underestimation of noise and vibrational comfort design can generate serious, unexpected issues emerging only in the advanced phases of the ship's life, and post-construction interventions are often needed. This case is exemplified in the current study, where the crew of the research vessel Mintis, a catamaran-type hull, reported discomfort in the navigation wheelhouse. A measurement campaign was set up to assess the complaints of the operating personnel regarding the high vibrational levels. Subsequent to the measurements, a numerical simulation, specifically comprising a modal analysis, was conducted to investigate the nature of the disturbance and distinguish the underlying mechanism at its origin. This paper meticulously presents and discusses the strategy undertaken to analyze and solve the vibrational problem encountered on board, with particular attention to the criteria and the modeling considerations adopted.

Keywords Modal mass fraction; Modal shape; Vibration measurement; Suspended floor; Catamaran

Article Highlights

- A vibrational problem in the wheelhouse area of a research ship is analyzed, proposing a strategy that combines on-field measurements and numerical simulations.
- Modal analysis serves as the primary tool for investigating the vibrational phenomenon on the ship wheelhouse suspended floor. It helps in understanding how different modes of vibration affect the structure.
- Two parameters, Modal Mass Fraction and Mode Shape, are selected to discretize between the significant modes among those characterizing the ship's dynamics.
- Three finite element models are developed to determine the vibrational issue's nature, which can be traced back to a tuning phenomenon between deck and wheelhouse suspended floor.

✉ Giovanni Rognoni
giovanni.rognoni@phd.units.it

[1] Department of Engineering and Architecture, University of Trieste, Via Alfonso Valerio 7/4, Trieste 34127, Italy

[2] Faculty of Marine Technology and Natural Sciences, Klaipėda University, Bijunu str. 17 Klaipėda, LT, 91225, Lithuania

[3] CS.N.I: Scarl, Corso Perrone 27/R, Genova 16152, Italy

1 Introduction

Vibrational comfort on board is increasingly becoming a crucial aspect of ship design. Low vibrational levels are indispensable for specific profiles such as ferries, passenger ships, and research vessels to guarantee acceptable operation. Vibroacoustic comfort predictive calculations on accommodation decks are uncommon practices due to their inherent complexity. The preliminary calculations are typically limited and only generally assess the coupling between main structural elements' natural frequencies and excitations. This process is often performed using practical approaches based on engineering formulas or numerical methods such as finite element analysis (FEA) or statistical energy analysis (SEA), which represent a more sophisticated approach (Carlton, 2018). These solutions need an accurate description of all the parameters involved: masses, stiffnesses, damping, and the intensity and position of principal sources, machinery, and propeller.

Noise and vibrational level predictions are challenging due to the involvement of multiple variables, primarily the distribution of masses, damping of materials, and outfitting. The insulation plan is often drawn from considerations based on practical experience. The variables affecting these evaluations include vibration sources, distance from the inhabited spaces, and the level of comfort desired in a specific area. Direct measurements are then used to validate the expected levels in the preliminary design. When excessive vibrations are recorded, the designer must resort to countermeasures, such as viscoelastic materials, floating floors, and, in some extreme cases, structural modifications to change the stiffness or the mass locally.

These countermeasures are often necessary after the construction of the vessel. Therefore, directly evaluating the vibrational levels arising on the structures when the ship is operational is essential. The measurement campaign must be meticulously planned, considering the following three basic aspects: (i) identifying the primary sources of disturbance; (ii) determining the specific areas of interest where vibrational levels need verification (such as working or accommodation spaces); and (iii) accounting for the sea state in which the ship is navigating during the survey because the motions of the ship impact the hydrodynamics of the propeller. Taking measures as closely as possible is crucial when conducting source-level measurements, preferably at the rigid connection between the source and the structure. Regarding receivers, assessments typically focus on deck levels, with acceleration recorded in all three directions. Two distinct conditions concerning the sea state may be considered: calm and rough seas. The effects of the propeller on the hull are dependent on this variable. In rough seas, pressure fluctuations on the propeller induce nonstationarity in the peaks of the measured vibrations, necessitating the use of an appropriate correction crest factor value (LR, 2006).

The specifications of the measurement techniques are standardized through a series of ISO standards (ISO, 2006; 2008; 2012; 2016). These documents define limits in terms of the overall root mean square (RMS) value calculated between 1 and 80 Hz. Additionally, they outline the procedures for processing acquired data, covering aspects such as frequency resolution, acquisition time, windowing, and averaging methods. In particular, ISO 20283-5: 2016 specifies the process for obtaining velocity or acceleration RMS for comparison with the limits. The acquired signal is transformed into a third-octave band spectrum and subsequently weighted. The overall RMS value within 1 and 80 Hz is eventually calculated. As per ISO 20283-2: 2008 and current practices, measured vibration levels are also often presented in narrowband spectrum formats. This visualization enables the immediate identification of the main disturbance sources by operators through observations of the frequency peaks. However, these peak values cannot be taken as absolute vibration values. They depend on the chosen frequency resolution, and an eventual limit should be based on these values. Limits were previously based on maximum repetitive values (ISO, 1984; DeBord et al., 1998), and RMS values were converted through an adequate crest factor. More recently, the ISO committee decided to resort to a specific RMS overall, which eliminates the ambiguity in the frequency resolution.

The design of a vibrational measurement campaign on board requires a certain type of experience regarding numerous aspects: point of measure, acquisition settings, and low-frequency sea and background noise disturbance. Classification societies' guidelines, class notations (ABS, 2006; LR, 2006), and international standards can help outline a procedure. Scientific literature provides some examples of on-field campaigns (Du et al., 2020; Egusquiza et al., 2008; Mitu and Memet, 2013; Sestan et al., 2012; Umezaki et al., 1969) on some types of ships with different purposes aiming at the hull girder or local structure dynamics. The latter is oriented toward comfort assessment or machinery safety. The scientific community is frequently focused on the discussion and evolution of international standards (Biot and De Lorenzo, 2007) and discussion regarding the comfort class of classification societies (Savreux et al., 2007). Reports or technical papers are released by private companies, manufacturers, or technical offices of shipyards, mainly oriented to present new practical solutions and procedures. However, finding these documents is difficult. Finding literature focusing on developing technical innovations is more common: resilient mounts, floating floors, viscoelastic materials, or new technologies (innovative floating floors, tuned dampers, and metamaterials) (Chen et al., 2021; Liu et al., 2019; Moro et al., 2016; Tian et al., 2022).

The measurements alone are the first step to solving a vibrational issue observed on board. A strategy to address the problem is necessary once the source of discomfort and the locations affected are determined. Given the intricate nature of ship structures, analytical models often fall short of encompassing all complexities. Engineers resort to numerical models, which can simulate a vessel partially or in its entirety. Iqbal and Shifan (2018) illustrated the use of FEA in the study of ship structures from its origin, focusing on the procedure to use depending on the necessary type of analysis. In principle, modal and frequency response analyses should be performed to obtain a complete picture of the situation.

Modal analysis helps identify potential matches between excitation frequencies and vibration modes. The principal focus for authors is predicting the resonant frequencies of the ship's body (global modal analysis) or different parts of the ship, such as decks, bulkheads, or any other structural element (local modal analysis). Global mode investigations were performed on catamarans (Sun et al., 2006; Cotaquispe et al., 2023), yachts (Pais et al., 2017; Vergassola et al., 2019), tankers (Yucel and Arpaci, 2013), scientific research ships (Zou et al., 2022a; 2022b), river boats

(Moro et al., 2013), and ship models (Liang et al., 2015). In several studies, the resonant frequencies calculated numerically were compared with those measured experimentally. Boorsma and Carden (2011) used modal analysis to investigate a problem of high vibrations encountered on the navigation bridge of a tanker, identifying a mode originating from the excessive vibration in the bridge wings. Although they indicated action on the structure, they did not provide any solution.

Vibrational problems are generally addressed using frequency response analysis, which provides the vibration levels arising at the specific points examined. Pais et al. (2018) calculated the vibration levels on a superyacht, highlighting some critical areas where habitability was unfeasible due to higher levels than the limits and proposing solutions. Some authors extended the modal analysis by performing a response frequency analysis to evaluate the vibration levels (Moro et al., 2013; Yucel and Arpaci, 2013; Zou et al., 2022a), verifying the habitability of the studied vessels according to ISO 6954: 2000. Frequency response analysis requires additional data: the damping of the elements involved and the quantification of the excitation, which can be challenging to estimate accurately. Precise excitation application demands extensive data collection and, frequently, a thorough structural model. In the absence of data, a qualitative investigation (imposing a unitary excitation force) can help compare the vibrational levels before and after an intervention. Frequency response analysis allows for assessing achieved comfort levels, whereas modal analysis remains crucial for designing effective countermeasures.

The present paper mainly aims to discuss a strategy based on modal analysis aimed at examining vibrational problems encountered on board ships after their construction. Modal analysis is the first step as part of an extended procedure, whose outcome is the design of possible solutions proposed to solve the issue. The strategy presented outlines how a designer should practically approach both parts of the overall procedure: (i) the experimental part, the measurement campaign on board, and (ii) the numerical part, the FE model created to study the phenomenon. The details are carefully considered, including methods, criteria adopted, and problems encountered. The paper also outlines this process using a real-life scenario comprising identification and analysis of a vibrational issue observed on the Mintis research vessel owned by Klaipėda University, Lithuania. The paper is organized as follows: Section 2 presents the case study and the measurement campaign performed. Section 3 outlines the strategy based on the use of modal analysis for solving these types of problems, with specific reference to the criteria (3.1) chosen for modeling and the FE models developed (3.2) to understand the phenomenon. Section 4 presents the results obtained from the numerical analysis. Section 5 subsequently discusses these results and outlines the hypothesis on the nature of the vibrational problem encountered.

2 Case study and measurement campaign

The Mintis research vessel owned by Klaipėda University is a 39-m-long and 12-m-wide catamaran with multifunctional marine research capabilities built in 2014 (Figure 1). The vessel has a first-class dynamic positioning system that ensures accurate positioning of the ship at up to 18 m/s of wind and up to 3.5 m wave height. These conditions provide high maneuverability, facilitating its operation in high and harbor areas. The vessel can host up to 11 scientists and researchers for research activity in the marine environment: seabed, water column, and biological objects (http://apc.ku.lt/index.php/moksliniu-tyrimu-laivas-mintis-atverevartus-i-pasauli/). The vessel is also equipped with two Caterpillar/Stamford diesel generators (804 kW) and Schottel STP 550 Z-drives, each with a power output of 700 kW, as well as two bow thrusters (STT 100 FTP, 150 kW).

Figure 1 Klaipėda University research vessel Mintis

The sea trials indicated acceptable vibration limits according to ISO 6954: 2000. However, the crew subsequently complained of high vibrational levels on the master's bridge in different operating conditions, causing discomfort to personnel involved in research activities who were unfamiliar with the ship environment. Thus, a measurement campaign was planned to investigate the problem. These tests were conducted in September 2016 in the Baltic Sea, in Lithuanian territorial waters. The measurements were performed while the ship was operating with one of the two main engines driving the two propellers. The levels obtained showed that an intervention is advisable; however, they were found to lie within the acceptable limits to maintain the ship in operation. The levels were also recorded in correspondence with the sources (engines and propulsion systems) and accommodation and working areas, for a total of 31 points. Figure 2 displays the points located on the navigation bridge. Two points, namely MT02 and MT03, exhibit the highest recorded vibration values. These points are located close to the navigation console and on the suspended floor.

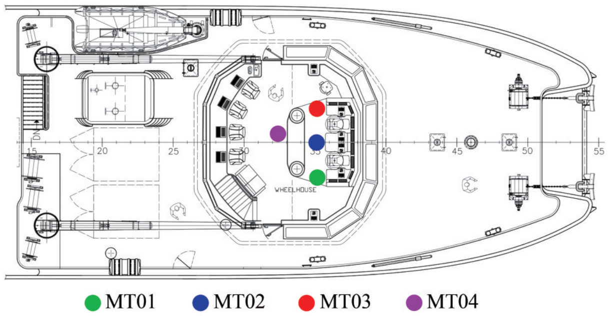


Figure 2 Measurement points where high vibrational levels were recorded

The accelerometer type used was the Wilcoxon 797. An ADASH A4400 VA4 Pro vibration signal analyzer with DDS software used for acquisition and data processing.

Six operating conditions (OCs) corresponding to six different propeller rotating speeds were checked as indicated in Table 1, which also displays the RMS velocity for points MT01, MT02, MT03, and MT04 calculated in accordance with ISO 6954: 2000. The overall RMS velocity values for these points fall below the threshold for "values above which adverse comments are probable" in C-classification areas (8 mm/s). Similarly, with reference to ISO 20283-5: 2016, these values remain below the "values of acceptable vibrations" for the navigation bridge (5 mm/s). No other points indicate values surpassing 2 mm/s.

Table 1 Velocity RMS [mm/s] calculated in accordance with ISO 20283-5: 2016

Points	Speed (r/min)					
	1st	2nd	3rd	4th	5th	6th
	130	150	200	250	280	295
MT01	0.5	0.2	0.8	1.1	0.8	1.8
MT02	1.0	0.5	0.9	1.6	4.1	4.0
MT03	0.7	0.4	0.7	1.2	4.6	3.6
MT04	0.3	0.2	0.4	0.9	1.4	1.5

Regarding the excitation frequencies aboard the vessel, the generator set is powered by a twelve-cylinder diesel engine operating at a rotational speed of 1 500 r/min (25 Hz) and a cylinder firing rate of 12.5 Hz. As shown in Table 1, the disturbance is linked to propeller speed, where elevated levels abruptly manifest for OCs 5 and 6, sharing a similar rotational speed. This finding indicates the potential excitation of a specific resonance phenomenon near a blade frequency range of 14 to 14.75 Hz. Other sources on board were ruled out as the root cause of the problem because they were unaffected by the speed of the ship.

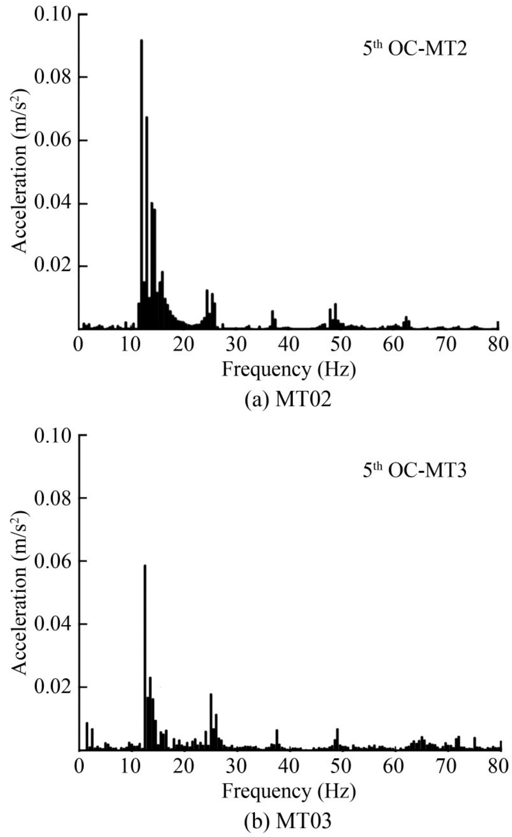


Figure 3 Acceleration spectra for points MT02 and MT03 for the 5th OC

The acceleration narrowband spectra computed for points MT02 and MT03 confirm the presence of sharp peaks in correspondence with the propeller frequency (Figure 3). These graphical representations are used to distinguish the frequency of the peaks rather than their values. Hanning windowing with a frequency resolution of 0.5 Hz was used for this data analysis. The two frequencies related to OCs 5 and 6 help determine a critical range of analysis for studying the structure and investigating the reported issue on board. Considering a margin of ±15% of the frequency range defined by the two operating frequencies, the critical range extends between 12 and 17 Hz. The considerations taken in the numerical analysis are based on this specific frequency interval.

3 Strategy for modal analysis

3.1 Analysis criteria

Studying the dynamic behavior of a vibrating structure is a two-step process. The first step involves modal analysis, which allows the identification of the natural frequencies and the associated modal shapes. This identification permits verifying if a disturbance detected at a specific frequency during the experimental campaign meets immediate correspondence with the deformation associated with a mode at a resonance frequency nearby. Moreover, analyzing the modal shape allows the evaluation of which structural parts interact with the area affected by the vibration, providing some helpful information for estimating the location, efficiency, and feasibility of future interventions. The second step is the calculation of the frequency response of the structure when excited by an external force. This analysis quantitatively evaluates the vibrational levels registered in a specific location. Moreover, this step consents to check the hypothesis made through the modal analysis and the efficiency of the interventions adopted.

In the case study examined, the cause of low comfort registered on board was investigated using modal analysis. The modal analysis was performed using Hexagon NASTRAN. The selected modal extraction method was Lanczos, and the modes were mass normalized; that is, each generalized mass mode was one. The analysis results do not directly quantify the levels experienced on board. However, two parameters offer a direct measure of the impact of each mode on the dynamics of the structure: the modal mass fraction (MMF) and the modal shape (MS). MMF is computed by NASTRAN, including in the "MEFFMASS" case in the case control section (MSC 2023). If $\boldsymbol{\Phi}$ is assumed to be the eigenvector's matrix of the system, then a body vector $\overline{\boldsymbol{U}}$ with motion in a desired direction can be constructed in accordance with Eq. (1):

$$\overline{\boldsymbol{U}} = \boldsymbol{\Phi\varepsilon} \tag{1}$$

where $\boldsymbol{\varepsilon}$ is the vector of scaling factors. The corresponding body mass $\overline{\boldsymbol{M}}$ can be calculated with:

$$\overline{\boldsymbol{M}} = \boldsymbol{\varepsilon}^{\mathrm{T}}\boldsymbol{\Phi}^{\mathrm{T}}\boldsymbol{M\Phi\varepsilon} \tag{2}$$

where $\boldsymbol{M}$ is the system mass matrix. The ith MMF, which is the contribution of each ith mode to the mass, is given by:

$$\mathrm{MMF}_i = \varepsilon_i^2 m_i \tag{3}$$

where m_i is the ith component of the diagonal matrix $\boldsymbol{\Phi}^{\mathrm{T}}\boldsymbol{M\Phi}$. The MMF measures how much mass is involved in each mode's dynamics. Modal mass is correlated to the body's six translation and rotation degrees of freedom; thus, a prevalent direction for the calculation must be selected. The MMF can facilitate mode ranking, where modes with a high MMF are generally the most dominant in the system dynamic. However, MMF could be low for local modes; therefore, limiting the analysis to this parameter can lead to the misidentification of some modes that affect the restricted structures necessary for the investigation. The second parameter considered, MS, helps distinguish these local modes despite their low MMF. MS also enables visual interpretation of the deformation of the structure at a specific frequency.

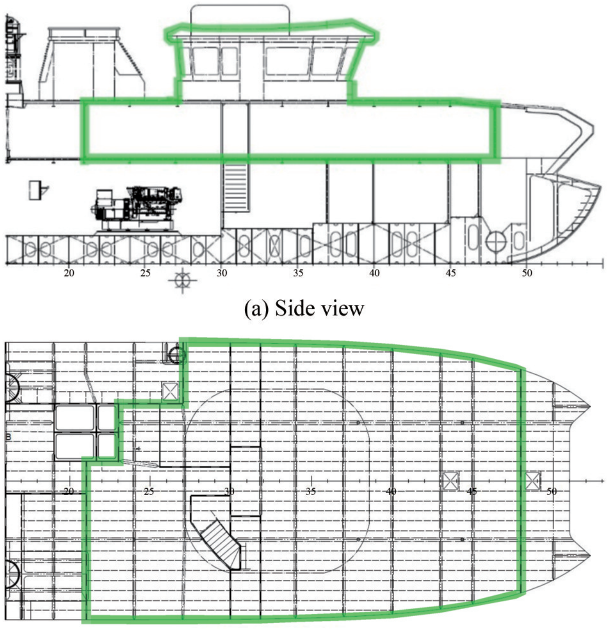
The interpretation of the modal analysis results starts with ranking the normal modes based on the MMF to individuate those more readily excited by sources on board. The selected range where this analysis must be performed is generally wider than the critical range identified by the measurement campaign to acquire additional information on the dynamics of the structure. Limiting the analysis to the critical range could lead to misinterpretations of the real situation. A mode with high MMF dominating the dynamics of the structure could be slightly out of the critical range, thereby remaining undetected. Moving on with the analysis of the results, attention is explicitly focused on the MS. The modes identified by their high MMF, as well as all the modes in the critical range, are visualized. The output of this second investigation step includes those directly affecting the zone where the disturbance was reported.

3.2 FE modeling

The first aim of the modal analysis was the identification of the structural parts and equipment that play a crucial role in developing the high level of vibrations perceived by the crew in the wheelhouse. The mechanism at the base of the case study vibrational problem was not immediately clear, despite direct identification of the sources exciting the deck vibration by the analysis of the measure campaign. The limited extent of the affected area indicated that the phenomenon was unrelated to the global dynamic behavior of the ship or the upper deck. Some considerations were made regarding the extension of the model, and some parts of the ship were not modeled to reduce the computational effort. The interaction between the unmodeled and modeled structures must be properly defined using appropriate boundary conditions.

The FE model does not need to describe an area greater than the free deck area containing the wheelhouse and is delimited by the ship sides and the two transverse bulkheads, extending from the port side to the starboard side, located immediately before and after the longitudinal limits of the wheelhouse. As shown in Figure 4, the limits assumed are indicated by green lines. All the structures in this area that connect the main deck with the deck below, specifically the trunk of the stairs and four pillars, were modeled. Such a model should allow the complete development of the local vibration phenomenon without disturbance due to the set of boundary conditions. During the vessel construction, the wheelhouse floor, which was not originally intended to be raised, was realized with a frame structure made by L-section beams welded on the deck, moving the walking surface 300 mm above. Figure 5 shows the structure.

The floor supporting structure was designed in the shipyard; thus, no drawings or technical specifications were


(a) Side view

(b) Upper deck top view

Figure 4 Green lines indicate the extension of the modeled ship structure

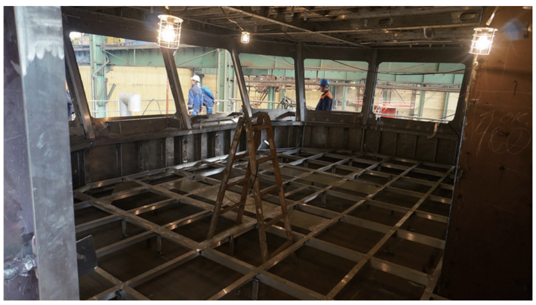
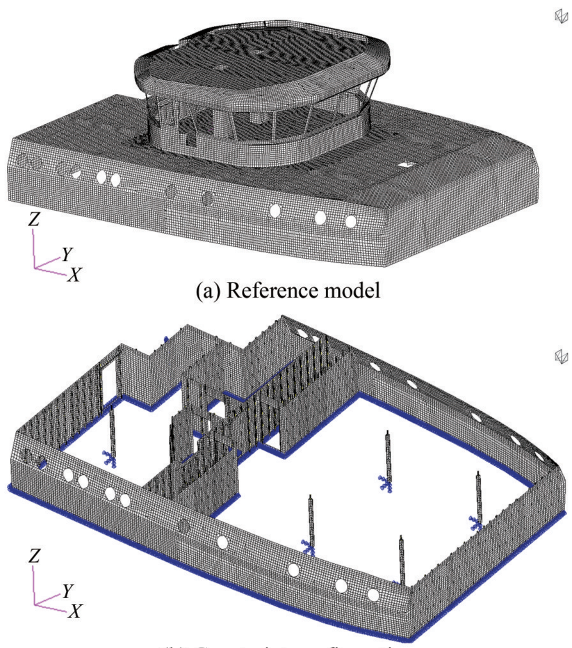


Figure 5 Picture of the frame supporting the floating floor taken during the construction

available. In the FE modeling, the dimensions and scantlings were estimated based mainly on the pictures taken on board during the construction. The documentation assumes the alignment of the longitudinal beams with the stiffeners; however, some differences were observed due to the ventilation ducts in between the suspended floor and the need to adapt the structure to the contour of the deck openings for the staircase. The framing system sustaining the floor results in a nonsymmetrical port to the starboard side.

The ship has been modeled with MSC Patran using shell QUAD4 and TRIA3 elements for plating and the webs of the main beams. Secondary beams (bulb flats), main girder flanges, the frame structure sustaining the wheelhouse floor, and the pillars were modeled using BAR2 beam elements. The chosen element size is 80 mm, which is equal to a fifth of the longitudinal stiffener spacing (400 mm). Eight to ten elements per wavelength are typically required to describe noise and vibration propagation sufficiently well. Considering an element size of 80 mm and a deck's plating thickness of 6 mm, the maximum frequency that can be safely analyzed is 91 or 142 Hz if 10 or 8 elements per wavelength are chosen, respectively (Hambric and Fahnline, 2016).

Some 0D mass elements were concentrated in the nodes placed in every geometrical center of the window to simulate the presence of the glasses in the wheelhouse structure. A ventilation room supporting instrumentation and a radar antenna are found on top of the wheelhouse. No data were available on the masses weighing on the wheelhouse top. Therefore, as indicated by the ship designer, a total value of approximately 1 500 kg was distributed as a "nonstructural mass." Regarding the boundary conditions, the nodes located at the intermediate deck level were blocked because their vibrational influence was considered negligible for the modal analysis in the wheelhouse area. This boundary condition assumes that all the nodes where the modeled structure is rigidly connected to the surrounding structure are fixed. The model described was considered the reference model and named "Model 1," as shown in Figure 6(a). Notably, the constraint configuration is shown in Figure 6(b), where fixed nodes on the intermediate deck are highlighted in blue.



(a) Reference model

(b) Constraint configuration

Figure 6 Reference model and constraint configuration

The deck, bulkheads, and ship sides are made of steel (E = 210 GPa, ν = 0.3, ρ = 7 800 kg/m^3), while the wheelhouse structure and the grillage sustaining the walking surface are made of aluminum (E = 70 GPa, ν = 0.3, ρ = 2 700 kg/m^3). Table 2 shows the principal characteristics of the structure.

Uncertainties emerge when determining the cause of the vibrational disturbance during the analysis. Therefore, two

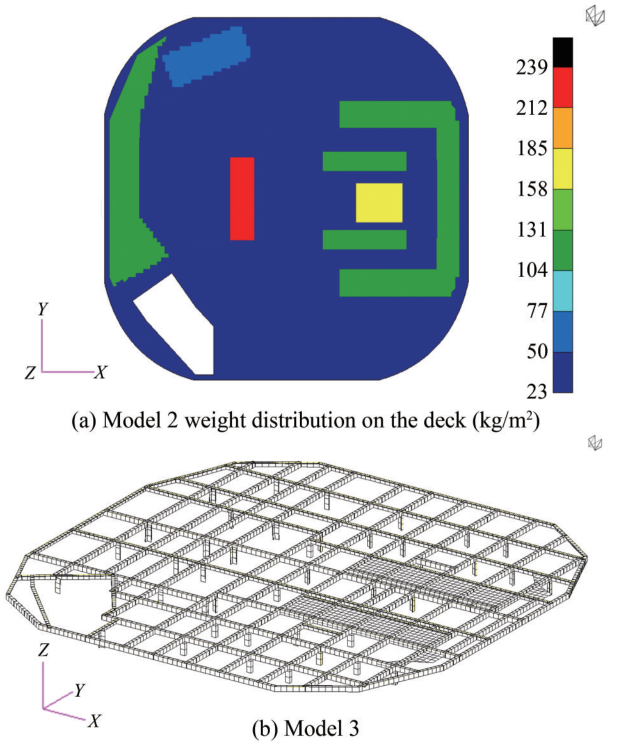
Table 2 Principal characteristics of the structure

mm

Hull scantlings (steel)		
Deck thickness		6
Frame spacing (Typical)		1 950
Stiffness spacing		400
Floor supporting structure (aluminum alloy)		
Beam L-section		75×50×6
Frame spacing	Transverse	600
	Longitudinal (average)	550

additional models were created to explore the mutual influence between the deck and the raised floor structure. Studying the same structure using FE models with different extensions and accuracies enables the validation of results occurring systematically and the isolation of the behavior of specific parts of the structure by analyzing the peculiarities between each model result.

The first modified model, named "Model 2," closely resembles Model 1, except for the exclusion of the grillage structure supporting the wheelhouse floor. In this version, the weight of the nonmodeled part was uniformly distributed among the shell elements as a nonstructural mass. The equipment and furniture weights were localized based on their real positions, ensuring the total mass remained unchanged. The boundary conditions for Model 2 are the same as for Model 1. The third model, termed "Model 3," represents the "negative" of the previous model, focusing solely on the removed grillage system. The suspended floor nodes connecting it to the wheelhouse sides and the nodes at the ends of the supporting struts were blocked in all three translations. With the two models, modal analysis can be conducted separately on the deck and floor supporting structures, providing insights into the dynamics of each component individually. Figure 7(a) illustrates the weight distribution on the deck in Model 2, while Figure 7(b) displays the structure of Model 3.



(a) Model 2 weight distribution on the deck (kg/m^2)

(b) Model 3

Figure 7 Model 2 weight distribution on the deck and the structure of Modal 3

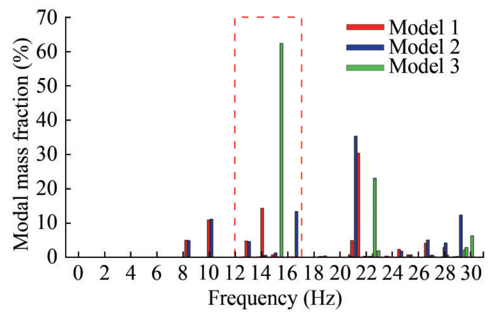


Figure 8 MMF values obtained for the analysis range 0–30 Hz

4 Numerical modal analysis results

For this specific case study, the entire frequency range analyzed is 0–30 Hz. This wide range was chosen to offer a general idea regarding structure dynamics, avoiding the risk of interpretation errors due to the restricted span of the critical frequency range (12–17 Hz). The element size chosen can appropriately discretize wavelengths in the selected analysis range. The wave at 30 Hz (the maximum frequency analyzed) is discretized by 18 nodes. Considering all the modes in the entire range, the selected MMF threshold was 1% of the total MMF computed. The MMF of the modes computed for the three models is shown in Figure 8, where the critical range is also highlighted.

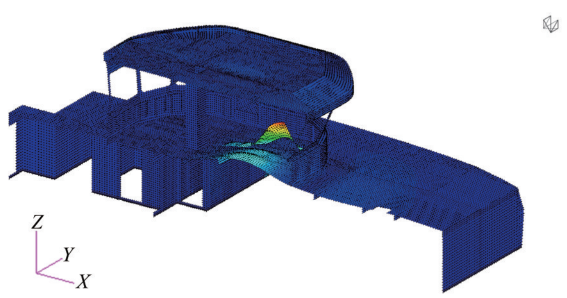
For Model 1, 84 modes were found in the 0–30 Hz range, 10 of which had an MMF of over 1%, collectively representing 91.6% of the total modal mass. Notably, the mode at 21.6 Hz accounted for 30.4% of the computed total modal mass. As for Model 2, 73 modes were evaluated in the range 0–30 Hz; 10 accounted for 94.2% of the mass computed, with the mode at 21.2 Hz contributing to 35.4% of the total modal mass. By contrast, Model 3, which is characterized by its limited size and predominantly 1D element modeling, exhibited 11 modes within the 0–30 Hz range. Six modes held 97% of the associated MMF, with the mode at 15.3 Hz having a substantial MMF of 62.4%. Table 3 shows the modes with an MMF exceeding 1% for the three models.

Concerning the MS, the modes with an MMF over 1% and all the modes in the critical range were carefully ana-

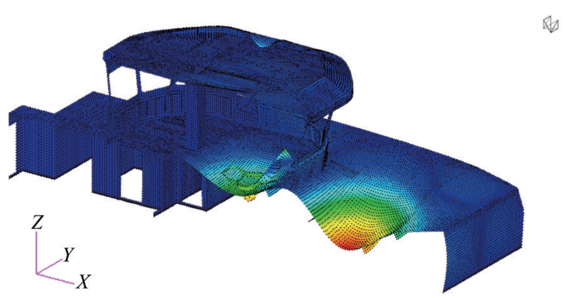
Table 3 Significant modes with relevant MMF for the three models analyzed

M1 (%)	f (Hz)	M2 (%)	f (Hz)	M3 (%)	f (Hz)
30.4	21.6	35.4	21.2	62.4	15.3
14.3	14.3	13.4	16.7	23.1	22.4
11.9	29.5	12.3	29.2	6.3	29.9
10.9	10.2	11.1	10.2	2.2	29.3
5.0	8.5	5.1	26.7	1.9	22.7
4.9	21.1	4.9	8.4	1.1	22.4
4.8	13.1	4.6	13.1		
4.1	26.8	4.2	28.1		
2.9	28.2	1.8	24.7		
2.3	24.7	1.3	15.1		

lyzed, visualizing the deformation of the structure. Any mode affecting the wheelhouse deck area was added to the list based on the MMF, even if the associated MMF was found below the fixed threshold. In Model 1, modes at 14.3 and 21.6 Hz directly affect the wheelhouse floor and the main deck, respectively. As shown in Figure 9(a) and 9(b), modes at 8.5 and 29.5 Hz affect the top of the wheelhouse; therefore, the vibration is transmitted to the area of interest. Within the modes in the critical range, the mode at 16.4 Hz, which was previously undetected, was found to be characterized by a deformation that involves the top of the wheelhouse, affecting the deck during vibration.



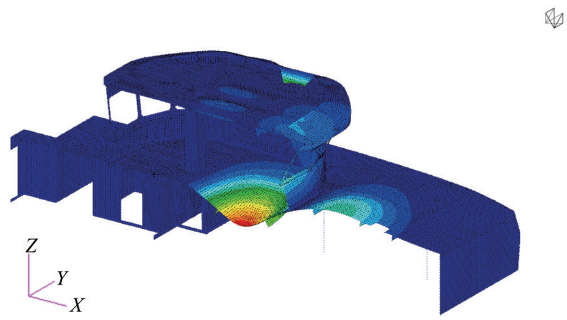
(a) Model 1 mode at 14.3 Hz



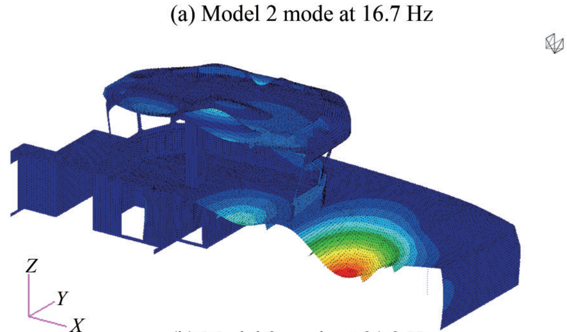
(b) Model 1 mode at 21.6 Hz

Figure 9 Model 1 mode at 14.3 Hz and 21.6 Hz

MS found for Model 2 are highly similar to those found for Model 1. The variation in the modeling of the structure sustaining the wheelhouse floor leads to slightly different MMF values associated with each corresponding mode. As for Model 1, the two modes of Model 2 with maximum MMF, namely 16.7 and 21.2 Hz, directly affect the wheelhouse floor and the main deck (Figure 10), respectively, while modes at 8.4 and 29.2 Hz affect the top of the wheelhouse. Similarly, for this model, the remaining modes with an MMF over the threshold involve parts of the structure that are not necessary for solving the issue. Regarding the critical range of vibration, the same considerations for Model 1 apply to Model 2: a mode affecting the reported zone with an MMF below 1% was detected, with an associated frequency of 16.4 Hz.



(a) Model 2 mode at 16.7 Hz



(b) Model 2 mode at 21.2 Hz

Figure 10 Model 2 mode at 16.7 Hz and 21.2 Hz

Regarding Model 3, the analysis of modes at 15.3 and 29.3 Hz MS indicates that the vibration in these modes involves the beam structure of the fore-end of the raised floor and the two metal sheets sustaining the floor, respectively. These parts are located exactly where the discomfort was detected (Figure 11). All the remaining modes involve every local part of the grillage structure and are not of interest. In the critical range of vibration, despite their MMF values, no other modes relevant to the analysis were found.

5 Upper deck vibrational performances

The analysis of the MMF and the MS allowed a critical interpretation of the role of various modes in the dynamics of the system. In the examined range, several modes with

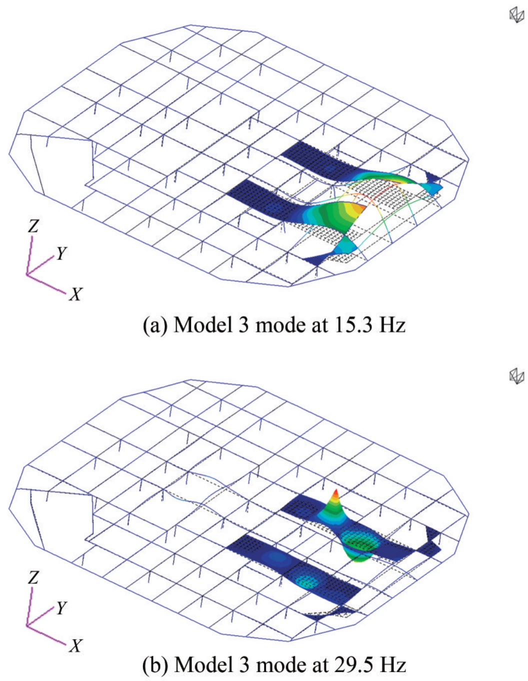


Figure 11 Model 3 mode at 15.3 Hz and 29.5 Hz

high MMF were considered important because, to varying degrees, they involve the portion of the ship where discomfort originates. Model 1 results show the presence of a mode located in the critical range (14.3 Hz) with a notable MMF and whose influence is probably dominant on the dynamics. A mode outside the range could have a substantial impact on the structure, revealing a greater MMF compared to the mode at 14.3 Hz. However, this mode is located at 21.6 Hz, which is too far from the source frequency (14–14.75 Hz) to be relevant. All the other mode frequencies are far from the excitation or are defined by an excessively modest MMF to be meaningful.

Therefore, this paper focused on the mode at 14.3 Hz, which is relatively close to the frequency of the critical velocity peak measured on board. The exploration of possible countermeasures to reduce the vibration level affecting the wheelhouse floor will necessarily rely on this specific mode of vibration. Determining the mechanism at the base of the interaction between the deck and the raised floor structure is crucial. Thus, the analyses performed on Models 2 and 3 were explicitly designed to accomplish this task because Model 1 alone does not provide the necessary information to clarify if the disturbance is attributed to the deck or the floor-sustaining structure.

The result from the reduced FE models shows that both models present at least a mode in the critical range with a notable MMF associated: mode at 16.7 Hz in Model 2 (Figure 10(a)) and 15.3 Hz in Model 3 (Figure 11(a)). Therefore, when taken separately, both systems develop a resonance phenomenon in the proximity of the critical frequency range of investigation. The vibration mechanism generating the disturbance on board is attributed to the interaction between the deck and the supporting floor structure. Considering the proximity of the frequencies of the two modes, this mechanism can be traced back to the behavior of a harmonically tuned resonator. According to this hypothesis, the supporting structure, behaving similarly to a tuned damper, would absorb the kinetic energy of the deck in correspondence with its resonance. This behavior is generally the desired outcome of a tuned damper. However, in this case, the walking surface of the wheelhouse experiences an increase in vibrational levels, generating discomfort for the crew.

This deduction can be verified by identifying the two peaks generated by the tuned damp mechanism. The placement of a tuned resonator on the system for dampening eliminates the designed peak, causing two additional peaks. In other words, two new modes develop, one with a lower frequency and one with a higher frequency than the original one. The lower mode is detectable in Model 1 (the mode at 14.3 Hz), while the identification of the higher mode is indirect. The current analysis failed to detect the mode because it does not meet any of the criteria selected: a mode that has an MMF below 1% and is outside the critical range. Thus, the research was extended to find this mode. The modes with a frequency over 16.7 Hz were visualized to find a mode shape involving the deck-supporting floor structure, which did not correspond with any of the Model 2 modes. A mode compliant with these characteristics was found at 19.1 Hz and is depicted in Figure 12.

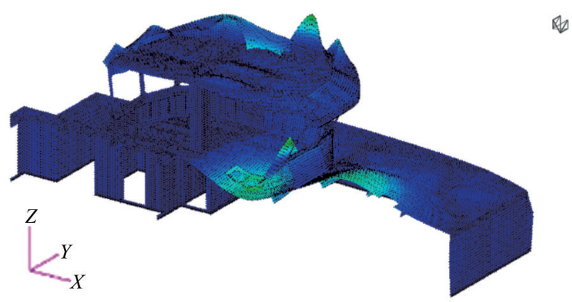


Figure 12 Model 1 mode at 19.1 Hz

The procedure outlined helped to clarify the dynamics of the structures involved in the vibration phenomenon registered on board by the measurement campaign. The mode at 14.3 Hz, generated by the interaction between the deck and the raised floor structure, is the leading cause of the vibration. Therefore, the procedure possibly suffers from generic problems afflicting dynamic calculation accuracy: the unprecise estimation of the real masses, the change in mass distribution on the structure during the time, and the secondary hull structures and outfitting modifications that cannot be checked by direct survey after ship completion. These data affect the model creation, which is based on nominal values. Therefore, a margin of uncertainty must always be considered. Emphasizing that modal analysis is the first step of a two-step procedure in the study of ship dynamics is essential. The second step, namely frequency response analysis, estimates the vibration levels arising on the ship, compensating for the principal limitation of modal analysis.

6 Conclusions

This paper outlines the entire process of how a real case can be addressed when a vibrational discomfort is experienced onboard, with specific emphasis on the understanding of the mechanism at its base. The adopted rational strategy clarified the interaction between the deck and the suspended floor that created the disturbance. The approach, which encompasses a measurement campaign, FE model creation, critical range definition, classification of relevant modes based on specific parameters (modal mass fraction and mode shape) and criteria adopted, submodel definition, and analysis, can be generalized and extended to other scenarios where local structures exhibit high vibration levels due to the interaction with the onboard sources. The proposed strategy emphasizes the effectiveness of modal analysis through the process described.

The outcome of the outlined procedure enables the design of potential solutions to address the problems encountered. The optimal solution can be selected once a set of possibilities is presented, maximizing efficacy while minimizing invasiveness and cost. A comparative modal analysis of the solutions must be performed to verify the decoupling between the structures and the excitations, highlighting the influence of structural modifications on modes. Frequency response analysis can help quantify the effectiveness of the devised technical solutions. A typical solution to modify the dynamic of the structure would involve moving the resonant frequencies, adding mass, or strengthening a part. Adding viscoelastic materials is unhelpful at low frequencies, and installing a floating floor is impractical. A suggestion for future research is to investigate the possibility of rigidly connecting the two parts, namely the deck and suspended floor, with a set of beams or additional pillars.

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