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INFLUENCE OF ADDED WATER MASS ON SHIP STRUCTURE VIBRATION PARAMETERS IN VIRTUAL AND REAL CONDITIONS

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Abstract: Modelling of ship structures in a virtual environment is now standard practice. Unfortunately, many engineers forget to consideri the influence of added water on the frequency values and the amplitude of natural vibrations. The article presents the effect of water damping on the frequency values of the individual natural vibration modes. The tests were carried out in two stages. First, the mentioned values were determined using FEM and then the values obtained in this way were compared with the parameters measured during laboratory tests. For the needs of laboratory measurements, structural elements made of ship steel in one of the Polish shipyards were used. All welds of the test objects have been verified in terms of their correctness. Irregularities in the execution of welded joints could result in a measurement error that is difficult to identify. As a result of the tests, the percentage differences in the frequency of occurrence of natural vibrations of individual modes were determined according to added water mass considerations. Importantly, the research concerned a real structural element of the hull, and the results obtained confirm the need to take into account the mass of accompanying water during the hull's FEM analysis. A number of more detailed research results were obtained, the most important of which is the fact that the finite element method is a valuable method for assessing the dynamics of wetted structures: the error in determining the vibration frequency did not exceed 5% for basic modes. The method of modelling the tested structures was almost equally important: the discrepancy of the results reached 4% depending on the modelling method. When designing marine SHM systems, it is essential to consider the effect of added water mass, since the frequency variations of a damaged structure, in relation to an undamaged one, are of the same order as the effect of added water.

Keywords: Natural vibrations, ship constructions, water damping, FEM modelling.

1. INTRODUCTION

Intensively operated ships are particularly vulnerable to critical damage, such as cracks in the hulls, which local resonances may cause [Okumoto et al. 2009; Eyres and Bruce 2012; Phillips et al. 2017; Agis and Pozo 2019]. This type of damage directly affects the level of navigation safety. The issues related to changes in the dynamic parameters of the vessel hulls as a result of the impact of external factors, which consequently leads to damage, are of interest to many research centres [Mondoro, Soliman and Frangopol 2016; Zeng et al. 2016; Ince et al. 2017; Piskur et al. 2021; Szeleziński et al. 2021].

All the listed works are related indirectly or indirectly to the safety of navigation. Starting from ship motion damping [Zeng et al. 2016] (maneuverability) through work related to the optimization of propulsion systems [Piskur et al. 2021] to research work on SHM type ship safety systems [Mondoro, Soliman and Frangopol 2016; Ince et al. 2017; Szeleziński et al. 2021]. The ship's hull comprises many parts, including important components such as thin and stiffened plates. Its dynamic characteristic (vibrations) can inform us of the significant risks of damage. The authors' main goal is to create a monitoring system for the endangered areas of the ship's hull. Such a system should be characterised by high reliability in the assessment of the hull strength situation. For this reason, it is necessary to determine the dynamic characteristics of stiffened plates in a laboratory and the influence of fluids on these characteristics.

An analysis of literature enables to find current items related to the topics covered by the authors in this publication. The study of vibrations of air-backed structures in water according to naval structures was studied by [Di Trolio, Boldini and Porfiri 2021], and the issues raised by the authors focus on modelling. Additionally, the work [Aureli, Basaran and Porfiri 2012] focuses on issues related to mathematical modelling and subsequent implementation of the developed model in a virtual environment. Comparisons of commercial programs in the field of computational fluid dynamics (CFD) with the author's program were made in the work [Yu, Ong and Li 2018]. The presented results indicate that the authors improved the computational algorithms used in CFD simulations.

However, no laboratory verification of the obtained simulation results was performed. The authors of the work [Kiciński and Jurczak 2021] verify their theoretical considerations and the results obtained by modelling with measurements under real conditions. A calculation model was proposed using CFD (Computational Fluid Dynamics) and CAE (Computer Aided Engineering) techniques to determine resistance force characteristics during the design stage. However, the cited publication, despite the fact that it uses similar tools, focuses more on determining the resistance of a body moving in a fluid.

A relatively complete experiment on the vibration of a plate structure submerged in water is presented in the paper [Vu et al. 2007], which complements theoretical fluid-structure interaction and validates results in laboratory conditions. The object of research were typical steel plates without any reinforcements. Studies of a similar nature were also conducted by the authors of another publication [Valentín et al. 2014].

An analysis of the available literature performed by the authors shows that the problem discussed in this publication is still very current. There are also no publications known to the authors specifying the influence of added water mass on ship structures vibration parameters, including the comparison of the results of numerical simulations and laboratory measurements.

The outer parts of the hull in the area of the wetted surface are in contact with the seawater, which affects their resonance characteristics [Liu et al. 2015; Bezerra et al. 2018; Agis and Pozo 2019; Kubit et al. 2022]. One should also consider the influence of the transported cargo and fuel, water, and oil reserves, which also change the values of the damping coefficient of the elements with which they come into contact. Therefore, it is crucial to skillfully use numerical tools such as FEM to correctly determine the resonance parameters of individual parts of the fuselage and its entirety. It is, therefore, vital to make skillful use of numerical tools such as FEM to correctly determine the resonance parameters of individual fragments of the hull and its entirety [Grządziela and Kluczyk 2021]. Choosing the confidence level and errors of calculation analysis is extremely important from the point of view of the planned SHM system [Grządziela and Kluczyk 2021; Kiciński, Szturomski and Świątek 2021; Kubit et al. 2022].

The thin plate model was analysed, and its natural vibrations frequencies without water (in the air) and coupled with the water were calculated. Thin plates are modeled with 1-dimensional (1D), 2-dimensional (2D), and 3-dimensional (3D) finite elements, as well as various finite element densities, to evaluate the accuracy of the chosen numerical modeling method.

2. NUMERICAL MODELING OF THE THIN PLATE

The structure of ships and ocean engineering facilities is usually covered with thin-walled sheets, the thickness of which, depending on the material used and the facility's application, ranges from 4 to even 100 mm. In this study, the structures were analysed using a sheet with a thickness of 8 mm. The further part of this chapter presents the simulation results and measurements of a thin-walled plate.

2.1. Vibration analysis of the thin plate without fluid contact

To verify the influence of water damping on resonance parameters, during the first stage of tests mathematical models of the thin plate 800x200x8 mm and 720x60x6 mm were built. Models were fixed at both ends in two situations. The first one has no contact with water. In the second case, the thin plate has the lower side in contact with water. Models are built using numerical modelling in the Patran-Nastran software platform. The models of the thin plate with the lower side in contact with the fluid is made using the Mfluid element in the Nastran software.

The real model of the thin plate studied in this section has the parameters shown in Table 1.

Table 1. Geometric properties of the thin plate

Dimensions	Length [mm]	Width [mm]	Thickness [mm]	Young Modulus E [GPa]	The density of the material ρ [kg/m^3]	Poisson's ratio v
Thin plate-steel	800	200	8	210	7850	0.3
	720	60	6	210	7850	0.3

Source: own study.

The natural vibration frequencies of thin plates and the influence of numerical models on the accuracy of calculation results are determined.

The authors present this calculation's results using four different numerical models in Figure 1.

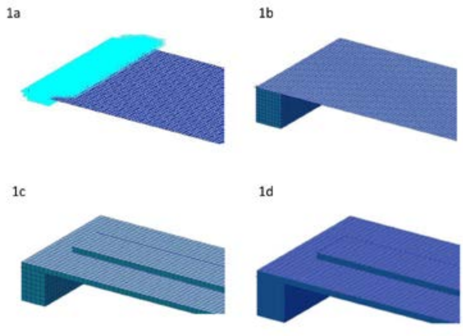

Fig. 1. Computational models of the thin plate: 2D – (1a), 2-3D – (1b), 3-D – (1c) and 3-D det – (1d) respectively

Source: own study.

All used computational models are presented in Figure 1. Model 2-D (two-dimensional): the thin plate is modeled by 2D finite element, number of elements 6400 (Quad 4), number of nodes 6601, number of degrees of freedom 36162 – Figure 1a.

Model 2-3D: the thin plate is modelled with 2D finite elements 6400 (Quad 4), and the thin plate is fixed at both ends with two plates. The two fixed plates at the ends of the thin plate are modelled with 3D elements 6400 (Hex8), the number of nodes 13981, degrees of freedom 35178 – Figure 1b.

Model 3-D (three-dimensional): both thin plates and two fixed plates are simulated by three-dimensional elements 19200 (Hex8), a number of nodes 27183, degrees of freedom 57195.

The 3D models with boundary conditions are presented in Figure 1c. Model 3-D det.: the thin plate model and two fixed plates with 3D finite elements. In this case, the number of finite elements is very large, 128000 elements (Hex8) to increase the calculation accuracy, a number of nodes 159084, degrees of freedom 303750.

Model 3-D det. presented as Figure 1d. Because the natural vibrations of the cases are relatively similar, the authors only illustrate the natural vibrations graphics of thin plates with the case of 3-D det. model. The natural vibrations of the 3-D model det. model without contact with water is presented in Figures 2a-2d.

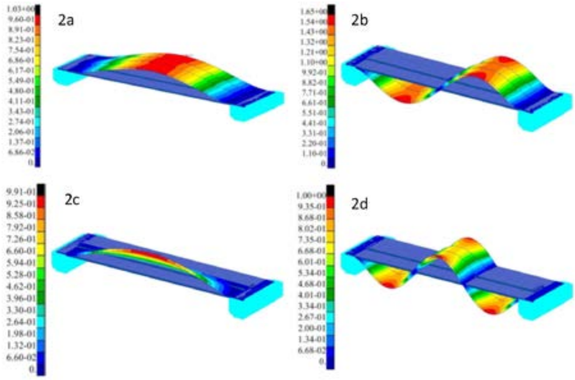

Fig. 2. Natural vibrations modes of analytical thin plates without water:
mode 1 (2a) – 135.58 Hz, mode 2 (2b) – 260.56 Hz, mode 3 (2c) – 375.68 Hz,
mode 4 (2d) – 455.77 Hz

Source: own study.

In fact, the natural vibration frequencies of the ship are usually below 500 Hz. Therefore, only the first four vibration modes were considered. The results of calculating the natural frequencies of each model are presented in Table 2.

2.2. Vibration analysis of the thin plate coupled with fluid

This section calculates the natural vibrations frequency of the thin plate combined with water. The numerical model method is based on the virtual mass method – Mfluid in Nastran software. The virtual liquid mass method is used for small movements of liquids that cannot be compressed. Fluids can be coupled with internal and external surfaces (with infinite fluid boundaries).

There is no clear liquid model; only wet structures (elements) must be identified since the liquid is represented by a combined block matrix directly attached to the structural points [Kwon 2011; Ma and Zhang 2014; Murawski and Charchalis 2014; Liu et al. 2015; Kubit et al. 2022]. A virtual liquid volume creates a block matrix representing the liquid to be coupled with a boundary consisting of structural elements and other effects, such as free surfaces, symmetrical planes, and infinite liquids. Uncompressed fluids create a defined block matrix with full coupling between acceleration and pressure on flexible structural interfaces.

The following is a brief overview of the virtual mass approach. The MSC Nastran software uses the Helmholtz method to solve Laplace's equation by distributing the set of sources on the outer boundary, each creating a simple solution to the differential equation. By connecting the assumed known boundary motions to the effective motion caused by the sources, the linear matrix equation for the magnitude of the sources is solved. Combining all these steps into a matrix equation leads to a virtual mass matrix as derived below [Kwon 2011; Byeon et al. 2018], in which:

$$\dot{u}_i = \sum_j \int_{A_j} \frac{\sigma_j e_{ij}}{|r_i - r_j|^2} dA_j. \tag{1}$$

u˙_i is the velocity of the fluid at it any point, σ_j is the value of a point source of fluid (units are volume flow rate per area) located at the location r_i, the point source is assumed to act over an area A_j, e_ij is the unit vector in the direction from point j to point i.

The other set of necessary equations are the pressures. p_i, at any point, i, in terms of the density ρ, sources and geometry, i.e.:

$$p_i = \sum_j \int_{A_j} \frac{\rho \sigma_j e_{ij}}{|r_i - r_j|^2} dA_j. \tag{2}$$

The results of integrating Eq. (1) and Eq. (2) over the finite element surfaces are collected respectively in two matrices, [χ] and [Λ], in which:

$$\dot{u}_i = [\chi]\{\sigma\}. \tag{3}$$

$$\{F\} = [\Lambda]\{\dot{\sigma}\}. \tag{4}$$

F are the forces at the grid points. The matrix [Λ] is obtained by integrating Eq. (2). An additional area integration is necessary to convert the pressures to forces. A mass matrix may now be defined using Eq. (3) and Eq. (4) as:

$$\{F\} = \left[M^f\right]\{\ddot{u}\}. \tag{5}$$

where the virtual fluid mass matrix, $[M^f]$ is:

$$\left[M^f\right] = [\Lambda][\chi]^{-1}. \tag{6}$$

The above equations are built and solved in the Nastran software to find the desired eigenvalues. Because the Mfluid method of solving fluid and structural interactions cannot be done directly in Patran pre-processing software. Therefore, the stiffened plate model is built with full geometry, material, meshing, and boundary conditions. After that, Patran software will create the file ending with *.bdf. To analyse the interaction between structure and fluid, the segment of codes will be added to the *.bdf file. The final file is analysed and processed in the Nastran software. After calculation, the authors obtained the natural vibration frequencies of the thin plate for different cases of finite elements: one-dimensional, two-dimensional, three-dimensional and different finite element densities for thin plates with one side coupled with fluid.

Figure 3 shows an example of the vibration modes of thin plates with the fluid contact surface.

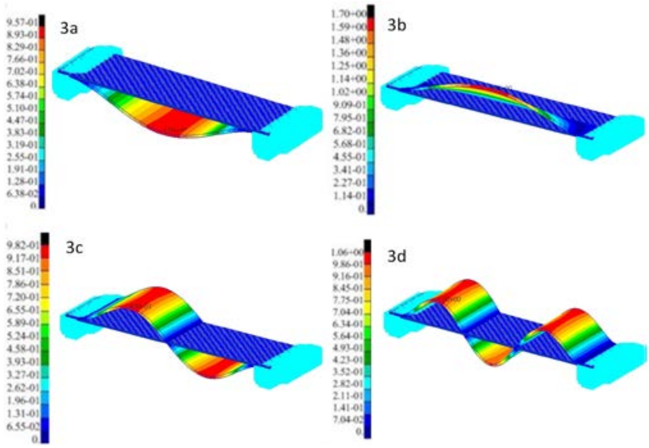

Fig. 3. Natural vibrations modes of analytical thin plates coupled with fluid: mode 1 (3a) – 130.34 Hz, mode 2 (3b) – 255.67 Hz, mode 3 (3c) – 375.46 Hz, mode 4 (3d) – 450.58 Hz

Source: own study.

The values in Table 3 show the four natural vibrations modes of thin plates with a fluid contact surface and are discussed later in this paper.

2.3. Thin plate simulations verification

When simulation results were obtained, they were verified during measurements using a modal hammer and accelerometers. The tested object was a steel plate element with dimensions and material parameters described in Table 1. The object was fitted at both ends (welded). During the measurements with the influence of the water plate has contact with it only with the side opposite the place where the accelerometers are mounted – Figure 4. During the measurement of the real thin plate, three 4514-B accelerometers were used: Figure 4.

During the study, the plate displacement was also registered with the use of laser sensors. However, the interpretation of the results is still unclear and requires further analysis, so they cannot be more detailed here.

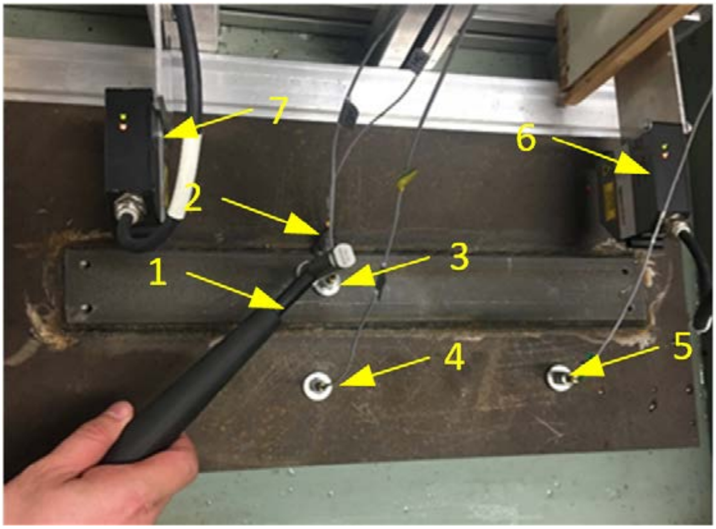

Fig. 4. Photograph of measuring model of thin plate. 1 – Impact hammer,
2 – Accelerometer (BEAMH), 3 – Accelerometer (BEAMV),
4 – Accelerometer (PLATEVCENTER), 5 – Accelerometer (PLATEVBORDER),
6–7 – Laser displacement sensors

Source: own study.

Measurements of vibration parameters were carried out in accordance with the above boundary conditions. The tests were carried out in accordance with the recommendations of normative documents, e.g., ISO 10816-1 and good engineering practice. A necessary element before starting the measurements is to perform the calibration procedure. It is recommended to calibrate the measuring apparatus before and after the measurements in order to detect possible damage to the apparatus during the tests. The obtained repeatability of the amplitudes as a domain of frequency for subsequent measurements proves high repeatability. The lack of repeatability may indicate incorrect assembly of the tested object or accelerometers and the occurrence of unforeseen local resonances. Obtained time courses were analysed, as a result of which amplitude spectra were obtained.

In Figure 5, we can see the amplitudes of vibration acceleration whose maximum values identify the frequencies of individual resonances. We mean the resonances possible to be recorded with this configuration of sensors and applied excitation using a modal hammer.

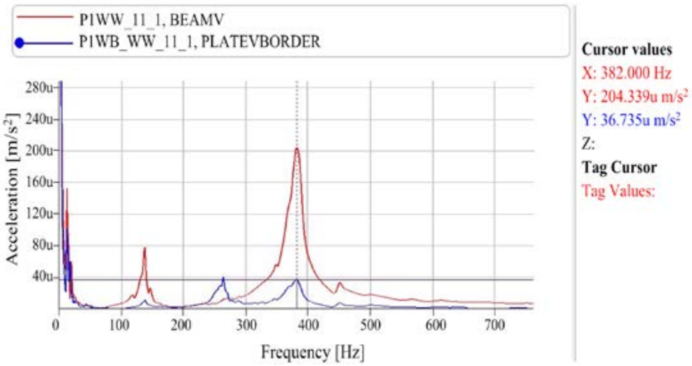

Fig. 5. A thin plate without water plate acceleration spectrum after the impact of a modal hammer

Source: own study.

Figure 6 illustrates the response of the acceleration spectrum of vibration to the impact of the modal hammer on the thin plate coupled with water.

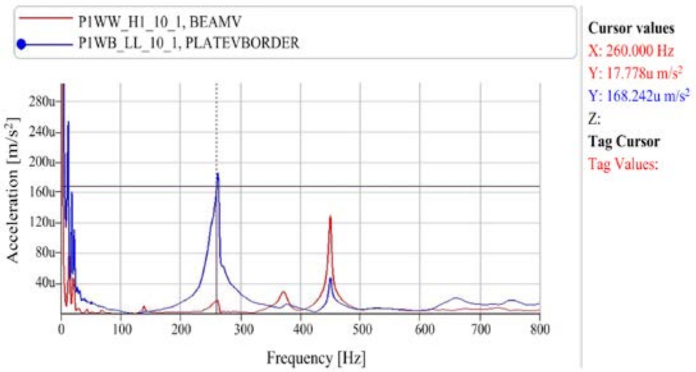

Fig. 6. A thin plate in contact with the water acceleration spectrum after the impact of a modal hammer

Source: own study.

In Figure 5 and 6, the authors presented response spectra of the thin plate without and with water, thanks to which it is possible to read the value of natural vibration modes. They are 138 Hz, 264 Hz, 382 Hz, and 451 Hz, respectively, for the thin plate without water. They are 132 Hz, 260 Hz, 371 Hz, and 448 Hz for the thin plate with water. The values of the frequencies of the other forms were not read because there is no excitation at such high frequencies in the regular operation of vessels.

A comparison of the natural vibrations frequency value of the thin plate using the numerical modelling method implemented on the Patran-Nastran software platform and the measurement method is present in Tables 2 and 3. Identification of the measured vibration modes was carried out on the basis of comparison of test results from all six sensors (see Fig. 4) and comparison with FEM calculations. Identification was not a problem because the detuning between the individual modes is sufficient (above 20%).

Table 2. Frequencies values for different models calculations for the thin plate without water

	The natural frequency of the model [Hz]				Measurement tests [Hz]
	Model 2D	Model 2-3D	Model 3-D	Model 3-D det.	
Mode No. 1	130.45	133.67	132.75	135.58	138.00
Mode No. 2	255.50	260.89	258.34	260.56	264.00
Mode No. 3	368.79	373.28	374.89	375.68	382.00
Mode No. 4	437.98	445.76	456.45	455.77	451.00

Source: own study.

Table 2 presents the results of simulations and measurements obtained for plates without contact with water and Table 3 shows those results obtained for the plate with water contact.

Table 3. Frequencies values for different models calculations for the thin plate with water

	The natural frequency of the model [Hz]				Measurement tests [Hz]
	Model 2D	Model 2-3D	Model 3-D	Model 3-D det.	
Mode No. 1	125.55	128.76	128.75	130.34	132.00
Mode No. 2	252.58	252.97	253.34	255.67	260.00
Mode No. 3	365.28	368.49	375.97	375.46	371.00
Mode No. 4	435.77	453.76	453.45	450.58	448.00

Source: own study.

Among the numerical models presented in the paper, basically all of them are characterised by a proper reflection of the dynamic characteristics of the tested objects. All models have their limitations. The simplest 2D model may not be sufficient to use for calculations of more complex structures. The influence of assembly on the obtained results is not taken into account here. First, given a 2D model, a basic plate model is often used with selected boundary conditions appropriate to the constraints. This model may be too simple to accurately reproduce the dynamic properties of thin plates. The respective error win the frequency of the first form of natural vibrations is 5.47% for the plate without the influence of the surrounding water and 4.89% with this influence taken into account. Due to the use of plate and solid elements in the 2-3D model, there are some problems in their mutual connection.

The consequence of this is the occurrence of errors. With regard to the first and third modes of vibration for a thin plate, without taking into account the influence of water, they are 3.22% and 2.14%, respectively. Taking into account the plate and the influence of the surrounding water, the values of the first three modes of vibration are burdened with errors of 2.85%, 2.70% and 2.56% respectively. The 3-D det model, is a highly complex model.The presented element using the 3-D det method has 330,000 degrees of freedom. A quick analysis of such a complex model requires significant computing power and is unavailable for desktop computers. Therefore, the 3-D det model can generate significant costs. The authors indicate a 2-3D model that is considered optimum from the point of view of technical practice. The obtained research results confirm the validity of this approach.

3. NUMERICAL MODELING OF THE STIFFED PLATE

An extremely important requirement of a new ship is that it can trade profitably. Not only should the final design take into account present economic considerations but also those likely to develop within the life of the ship. With the aid of computers, it is possible to make a study of a large number of varying design parameters and to arrive at a ship design that is not only technically fully efficient but also the most economically efficient [Falkowicz and Valvo 2023].

Ideally, the design will take into consideration the initial cost, operating cost, and future maintenance [Okumoto et al. 2009]. Due to economic benefits and excellent resistance to various types of mechanical damage, stiffened steel plates are commonly used in the shipbuilding industry. The hull plating is stiffened with structural elements such as a step, frames and stringers. At a later stage of this work, this kind of element was tested.

3.1. Vibration analysis of the stiffed plate without contact with fluid

The stiffened plates are widely used in modern technical fields such as construction, aerospace, aircraft manufacturing, and shipbuilding. Especially in the shipbuilding industry, the stiffened plates are the main components that make up the ship walls, decks, and hulls. Therefore, it is essential to study the influence of water damping on vibration characteristics. The structure's natural frequencies, when in contact with fluid, differ from the frequency of the structure in the air. Therefore, the calculation and understanding of natural frequency changes due to the presence of fluid are essential to designing structures in contact with or immersed in the fluid to meet safety conditions. In general, the effect of fluid damping on the structure is expressed in terms of mass added to the structure. It increases the mass and reduces the structure's natural frequency when measured in the air. This section calculates the vibration analysis of the stiffened plates in both air and fluid environments. Analytical calculations based on the finite element method are performed on the same software. The natural vibrations of the stiffened plates in the air is considered for different finite element densities and finite element types, the same as it was made for the thin plates.

The numerical modelling of the stiffened plate with geometric parameters is presented in Table 4, which is placed at the end of the article. This section calculates the natural vibration frequencies for the stiffened plate coupled with and without water for the variable mesh density and finite element types.

Table 4. Geometric properties of the stiffened plate

	Length [mm]	**Width [mm]**	**Height [mm]**	**Thickness [mm]**	**Young Modulus *E* [GPa]**	**The density of the material *ρ* [kg/m^3]**	**Poisson's ratio *v***
Thin plate	800	200	-	8	21	7850	0.3
Beam	720	60	60	8	21	7850	0,3

Source: own study.

The stiffened plate model in the Patran-Nastran software is illustrated in Figure 7. A vibration analysis of the stiffened plate is also considered for two cases: in the first case, the stiffened plate does not contact fluid, and the stiffened plate coupled with the liquid at the underside of the plate.

Model 2-D (two-dimensional): the stiffened plate is modelled by two-dimensional finite elements, number of elements 9856 (Quad 4), number of nodes 8341, number of degrees of freedom 46110. Model 2-3D: the stiffened plate is modelled with two-dimensional finite elements 9856 (Quad 4), and the stiffened plate is fixed at both ends with two plates. The two fixed plates at the ends of the thin plate are modelled with three-dimensional elements 6400 (Hex8), a number of nodes 15721, and degrees of freedom 45618. Model 3-D (three-dimensional): The stiffened plate and two fixed plates are simulated by three-dimensional elements 19200 (Hex8), number of nodes 27183, degrees of freedom 57195. Model 3-D det.: the stiffened plate model and two clamps with three-dimensional finite elements. In this case, the number of finite elements is very large, 192800 elements (Hex8 (to increase calculation accuracy)), number of nodes 210657, degrees of freedom 458451 – see Figure 7.

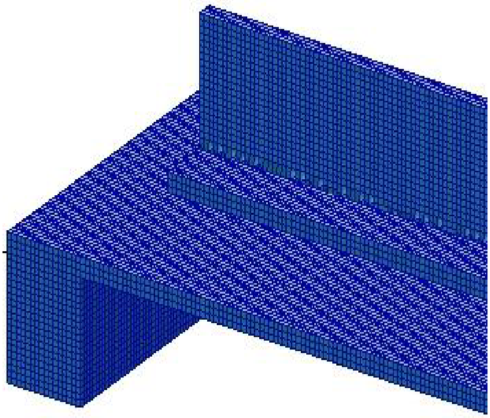

Fig. 7. An example of a computational model of stiffened plate

Source: own study.

The natural vibrations mode of the 3-D det. model is presented in Figure. 8.

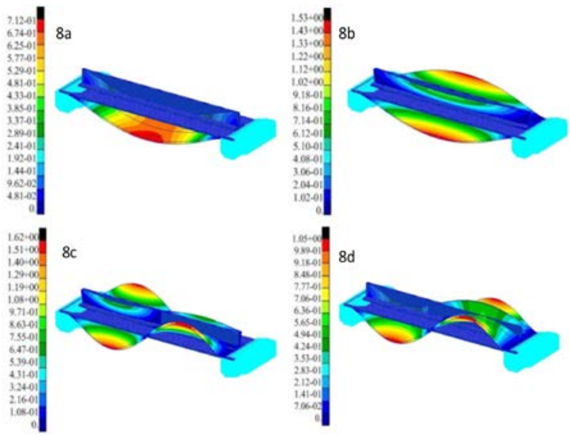

Fig. 8. Natural vibrations modes of analytical stiffened plates without fluid: mode 1 (8a) – 159.58 Hz, mode 2 (8b) – 275.47 Hz, mode 3 (8c) – 331.08 Hz, mode 4 (8d) – 400.02 Hz

Source: own study.

3.2. Vibration analysis of the stiffed plate without contact with fluid

The natural vibration frequencies of the stiffened plate combined with water were calculated using the same numerical modelling method as the thin plate in a previous chapter.

Figure 9 shows an example of the vibration modes of the stiffened plate coupled with fluid.

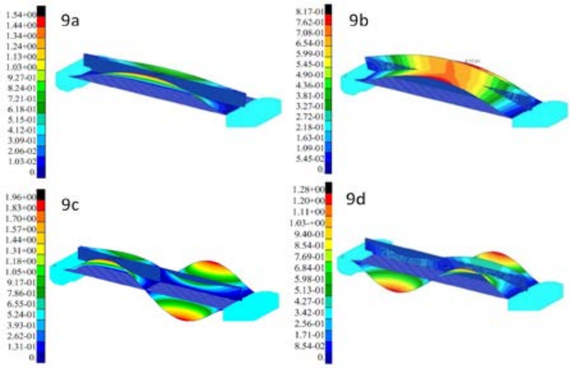

Fig. 9. Natural vibrations modes of analytical stiffened plates coupled with fluid: mode 1 (9a) – 142.52 Hz, mode 2 (9b) – 269.35 Hz, mode 3 (9c) – 300.80 Hz, mode 4 (9d) – 387.65Hz

Source: own study.

3.3. Stiffed plate simulations verification

After calculating the results of the stiffened plate natural vibrations for two cases combined with fluid and not with fluid, the laboratory verification was conducted-Figure 10.

Tested object was a steel plate element with dimensions and material parameters described in Table 4. The object was fitted at both ends (welded). During the part of measurements with the influence of the water plate has contact with it only with the side opposite to the place of accelerometers mounting – Figure 10.

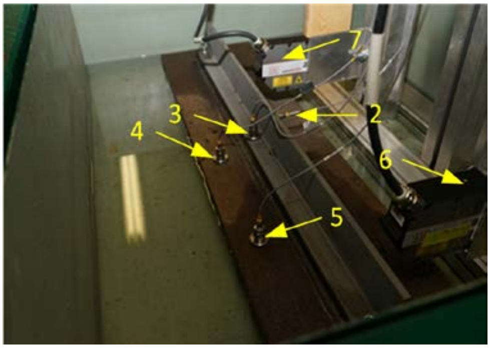

Fig. 10. Photograph of model of stiffened plate with water during measurements, 2 – accelerometer (BEAMH), 3 – accelerometer (BEAMV), 4 – accelerometer (PLATEVCENTER), 5 – accelerometer (PLATEVBORDER), 6–7 – laser displacement sensors

Source: own study.

After the measurement and calculation of FFT vibrations spectra, the response of stiffened plate without and with water using a modal hammer was shown in Figure 11. On the graph, the acceleration response of the vibration determines the natural frequency values at the graph's vertices.

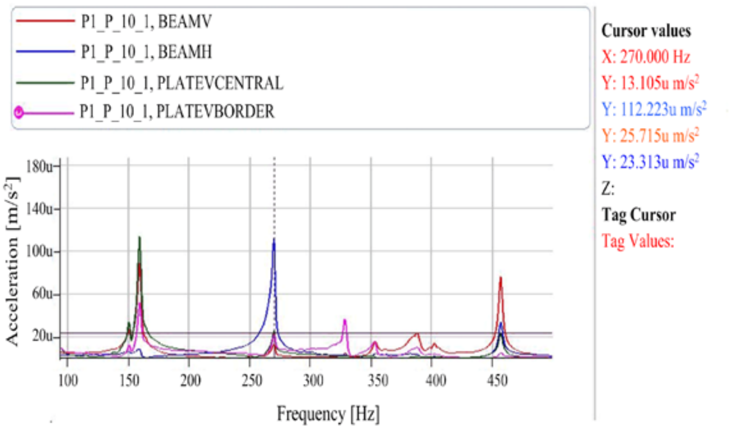

Fig. 11. Response of the acceleration spectrum after the impact of the modal hammer on the stiffened plate without water

Source: own study.

Figure 12 illustrates the response of the acceleration spectrum of vibration to the force of the modal hammer of the stiffed plate with water.

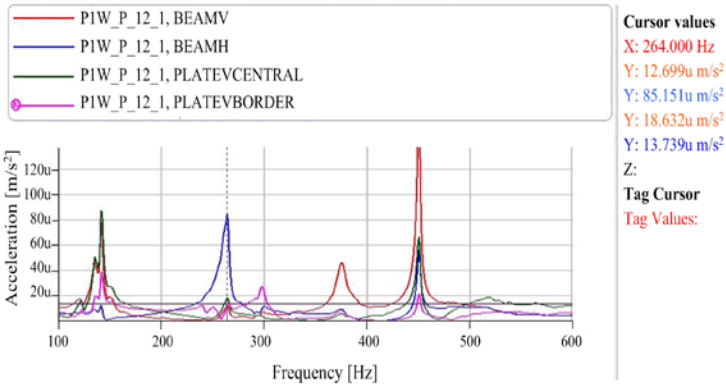

Fig. 12. Response of the acceleration spectrum after the impact of the modal hammer on the stiffened plate with water

Source: own study.

In Figures 11 and 12, the authors presented spectra of the stiffened plate without and with water. This makes it possible to read the value of the stiffened plate's natural vibration modes without and with water. They are 159 Hz, 270 Hz, 328 Hz, 388 Hz and 457 Hz respectively for the stiffened plate without water and 142 Hz, 264 Hz, 298 Hz, 376 Hz and 451 Hz for the stiffened plate with water.

The natural frequency results obtained by numerical model analysis, using the finite element method implemented on Patran-Nastran software and experimental methods presented for two cases without and with water, are illustrated in Table 5 and Table 6.

Table 5. Parameters of calculation model for the stiffened plate without water

	The natural frequency of the model [Hz]				Measurement tests [Hz]
	Model 2D	Model 2-3D	Model 3-D	Model 3-D det.	
Mode No. 1	153.52	157.56	155.36	159.58	159.00
Mode No. 2	259.65	275.15	278.55	275.47	270.12
Mode No. 3	317.16	323.35	335.47	331.08	328.00
Mode No. 4	408.98	405.55	403.75	400.02	388.12

Source: own study.

Table 6. Parameters of calculation model for the stiffened plate with water

	The natural frequency of the model [Hz]				Measurement tests [Hz]
	Model 2D	Model 2-3D	Model 3-D	Model 3-D det.	
Mode No. 1	135.11	140.71	137.75	142.52	142.00
Mode No. 2	256.88	270.04	272.36	269.35	264.00
Mode No. 3	285.15	295.78	305.79	300.80	298.00
Mode No. 4	395.34	392.01	392.27	387.65	376.00

Source: own study.

The individual model natural frequencies calculating results and the comparison with the measurement tests are shown in Table 5 and 6, for the stiffened plate without and with water respectively. Considering the 2D model, this model has a error of calculations that occurs for the simplest 2D model and for the first natural vibrations form is 3.45% and 4.85% for the stiffened plate without and with water respectively. In the 2-3D model has relatively large errors for the first and third vibration forms for the stiffened plate without and with water, with relative errors of 3.83% and 3.07% for the stiffened plate without water and 2.70%, and 3.07% for the stiffened plate coupled with water respectively. The exact 3D det. model does not have any of the above disadvantages.

However, it is many times larger than other models. It should be noted that the model 3-D det. has more than 0.5 million degrees of freedom. A 3-D det. The model cannot analyse without using the connected supercomputer in the network. A significant increase in computing costs does not include the increased accuracy of the calculation. Therefore, the 2-3D model is considered optimum in terms of technical practice.

4. CONCLUSIONS

The authors have conducted a vibration analysis of some selected real parts that make up the hull structure and superstructure, such as beams, thin plates, and stiffened plates. The analysis and calculations are performed by numerical modelling based on the Patran-Nastran digital software platform. Calculations and analyses for the selected structures are carried out for two cases, the first in the air, and the second coupled with the fluid. The calculation and analysis results of the selected structures have been verified by measurement methods conducted at the Marine Engineering Faculty of the Maritime University of Gdynia,thereby assessing the influence of error and the dispersion on the analysis results of the selected structures when applying the numerical modelling method for calculation. Particularly in this area, the FEM method was used. The selected numerical model method - the finite element method on the Patran-Nastran software platform - gives relatively accurate calculation results. It has acceptable errors, especially when applied for calculations and structural analysis in the initial design stage. It also helps architects and designers reduce testing and prototype-making time, thereby saving costs.

Boundary conditions (constraint conditions), different types of finite elements (one-dimensional - 1D, two-dimensional - 2D, three-dimensional - 3D), and different finite element densities have a large effect on errors and dispersions in analysis results for selected structures. When the mesh density is higher, the error is smaller, and the accuracy of the calculation results are higher. As the mesh density increases, the graph becomes smoother. The model of selected structures with three-dimensional elements and large element densities results in high accuracy, and the error is usually less than 2% compared to the corresponding measurement results. Thus, as the mesh density increases, the accuracy of the resulting increases, but at the same time, it increases the computational time and requires a higher configuration computer, increasing the computational cost. Therefore, selecting the appropriate mesh density is necessary to ensure accuracy and reduce the time and price. Through measurement verification, the numerical modelling is implemented in the Patran-Nastran software platform for the 2-3D finite element resulting in relatively accurate results. The highest error in both experimental cases for the selected structures without and with water was 5.47% and 4.85%, respectively. These errors tolerable especially in the design process.

However, taking into account that the authors' main goal is to create a monitoring system for the endangered areas of the ship's hull, the effect of added water mass on ship structures' vibration parameters is of significant importance. Accepting errors of this magnitude at the design stage, assuming the possibility of small errors at further stages of building the SHM system, may lead to future misinterpretation of diagnostic data critical for the safety of the vessel.

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