


Article

Evaluation of the Possibility of Increasing the Energy Absorption Efficiency of Fender Devices Using the Example of Cylindrical Fenders with Additional Structural Elements Applied

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Abstract: The providers of transport services in ports must ensure there is adequate protection of the quays against the hulls of vessels. Highly elastic fenders mounted on the wharfs or on the hulls of vessels are used to absorb the energy of an impact. The structure of the fender, and the highly elastic material used to make it, are designed to ensure the highest possible absorption energy with minimized reaction force. In this work, the efficiency of energy absorption by cylindrical fenders into which additional structural elements were introduced in the form of holes of various diameters, was determined numerically using the finite element method. It was found that the features of such structural elements affect the efficiency of their energy absorption. In order to confirm the accuracy of the numerical calculations, they were verified based on experimental determination of the functional parameters of the cylindrical fenders. The reaction force and absorption energy values determined numerically and experimentally for the cylindrical fender were shown to be consistent. The verified numerical calculation methodology was used to evaluate the energy absorption efficiency and the reaction force in cylindrical fenders with additional structural elements.



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Keywords: finite element method; fender; offshore wind turbine; sustainable development; absorption energy; reaction force; polyurethane elastomer; hyperelasticity

1. Introduction

Fenders are commonly used to protect port quays against impacts from vessels during maneuvering, mooring and parking operations [1,2]. Fenders also ensure safety during the installation of wind turbines and during service works on wind farms located on reservoirs [3,4]. The purpose of using fenders is to reduce the reaction force of an impact upon both the waterfront and the watercraft via the fender absorbing as much of the impact energy as possible [2,5]. The basic performance parameters used in the selection of fenders, and for the purpose of comparing their operational effectiveness, are the reaction force and the absorption energy. The values of these parameters are determined experimentally in accordance with the PIANC guidelines [6,7].

The structural design of fenders is subject to continuous development. The areas of research into new solutions relate to the construction of the main elements of fenders that are responsible for energy absorption [8], the design of elements connecting the fender devices with the berth [5], and the development of materials with features which ensure better performance parameters [9–11]. Among these structural designs, spatial structures filling the fenders are becoming more common [12]. Predicting the durability of fenders and determining which components have the greatest impact upon their service life is important to the design process [13]. Simulation calculations are increasingly being used to more efficiently design and produce prototype bumpers [12,14–16].

Any new design or modification of the geometry of a fender, or the use of a new or modified material for its manufacture, requires experimental verification of its performance. It is especially difficult to determine the operational parameters of large fenders. The use of numerical calculations can offer an alternative to costly experimental research. Prototyping studies using numerical methods may enable both the analysis of a greater number of design solution variants, and the selection of the best performance parameters, without the need to conduct experimental tests to determine the impact of structural and material changes on the performance parameters each time. Introducing changes at the design stage may also lead to a reduction in the number of physical prototypes that do not meet the design assumptions. Limiting the number of physical prototypes produced, may, in turn, lead to a reduction in the cost of research and development, and thus reduce the costs of implementing new solutions.

The materials used for the production of fenders are polyurethane elastomers, which are a group of highly elastic materials. In order to obtain the operational parameters for fenders that are as close as possible to the parameters of real elements through numerical modeling, it is necessary to know the material constants and to properly select the hyperelastic material models. An improperly selected hyperelastic material model may significantly affect the resulting values for the operational parameters of the fenders. The assessment of the influence of different models of hyperelastic materials on the values of the reaction force and absorption energy of cylindrical fenders will be the subject of a separate publication.

In the case of cylindrical fenders, in practice, fenders with an additional hole located in their axis are mainly used.

Various design solutions for cylindrical fenders are available in publications and in patents included in the patent databases. These solutions are related to the introduction of additional structural elements into the external area of the fender, or in the material space limited by the external and internal diameter of the element. The optimal dimensions of the outer diameter and the inner bore of a cylindrical fender which allow for a sufficiently high absorption energy and low reaction force have also been determined in previous works.

This paper presents a method for the numerical determination of the reaction force and absorption energy of an initial model of a cylindrical fender using the ANSYS program. The correctness of the results obtained through numerical calculations was verified by experimental determination of the operational parameters of the cylindrical fender. After confirming the compliance of the numerical calculation results with the experimental results, additional structural elements were introduced into a virtual model of the cylindrical fender. These additional structural elements were holes with a diameter of 10 or 5 mm. The influence of both the diameter of the holes and their quantity on the absorption energy and reaction force was analyzed. Based on the determined values for the basic operational parameters, the effectiveness of the new design solutions was assessed. Further possible directions for introducing changes in the structure of cylindrical fenders are also indicated.

2. Materials and Methods

The subject of this research was the cylindrical fender, which is commonly used to protect berths against damage during mooring and stopover of vessels. Examples of cylindrical fenders and the methods of their fastening to a quay are shown in Figure 1.

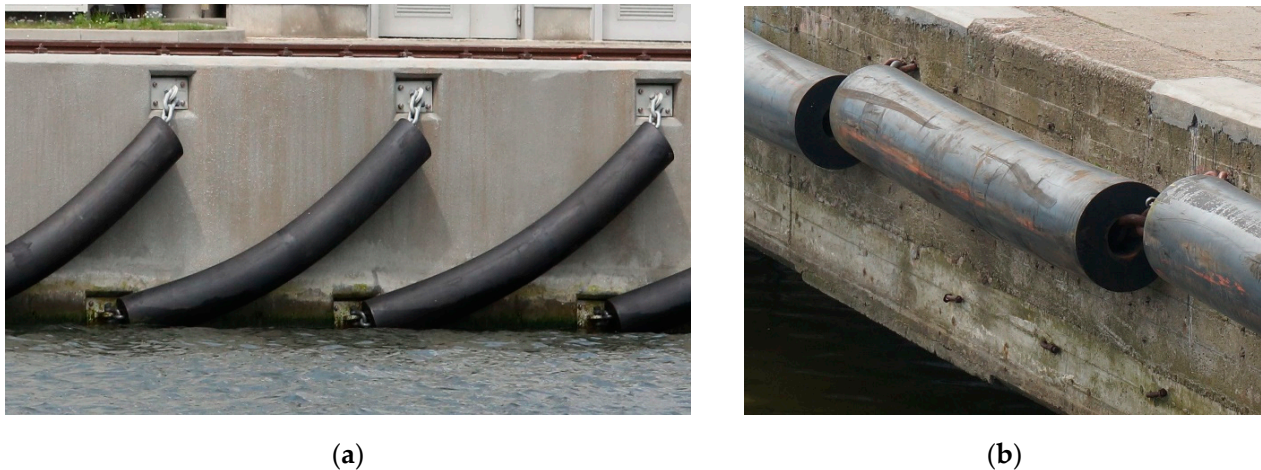


Figure 1. Cylindrical fenders and examples of their mounting on the quay. (a) Cylindrical oblique fenders; (b) Horizontally mounted cylindrical fenders.

The starting model for the numerical and experimental calculations was a cylindrical fender with an outer diameter, D , of 100 mm; an inner diameter, d , of 50 mm; and a length, L , of 500 mm (Figure 2a). The fender dimensions adopted for the calculations were based on the catalog values of the leading manufacturers of fender devices. A 2D calculation model was developed which took into account the conditions in which such elements are used (Figure 2b). The model assumed that the fender was located between two metal plates. One of the plates replicated the stationary quay, the other movable plate reflected the vessel pressing against the fender. The length of the steel plates, a , at 180 mm, and the thickness, b , at 35 mm, were the dimensions selected in order that the entire range of deformation of the fender was covered. Moreover, the chosen value for the thickness, b , was assumed to be able to prevent additional deformations in the plates during compression of the fender.

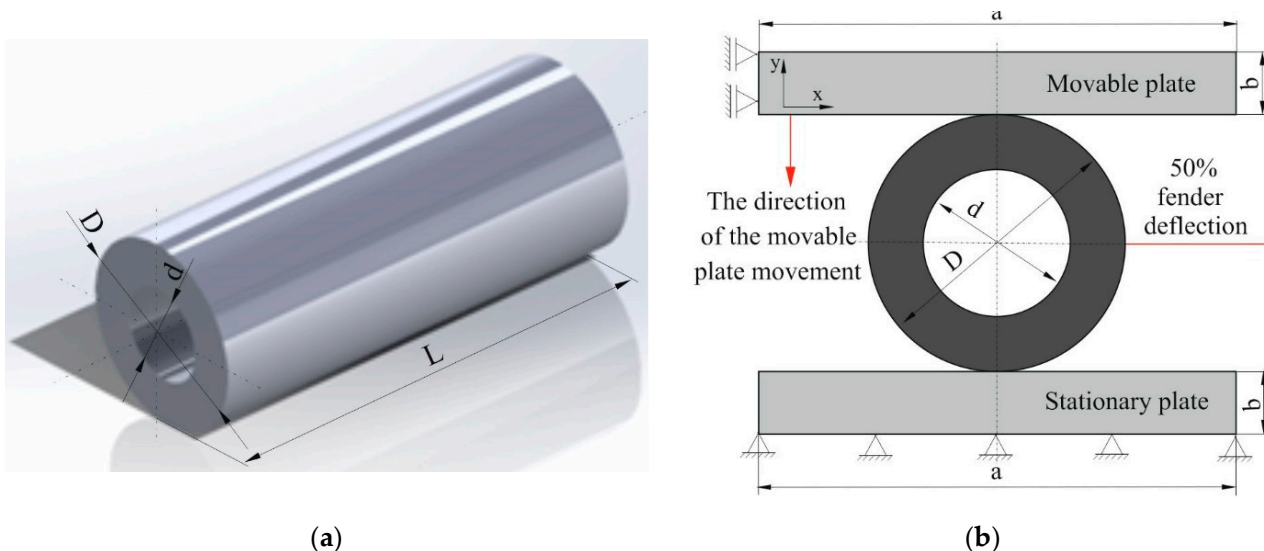


Figure 2. Model of a cylindrical fender. (a) Basic model of a cylindrical fender; (b) Basic computational model of a cylindrical fender taking into account the use conditions.

Ensuring the optimal performance parameters of fenders requires the use of materials characterized by high elasticity. For the calculation model adopted, it was assumed that the fender was made of polyurethane elastomer, while the movable and stationary plates were made of ordinary quality steel. The material constants adopted for the calculations are summarized in Table 1.

Table 1. Material constants of polyurethane elastomer and steel used for numerical calculations.

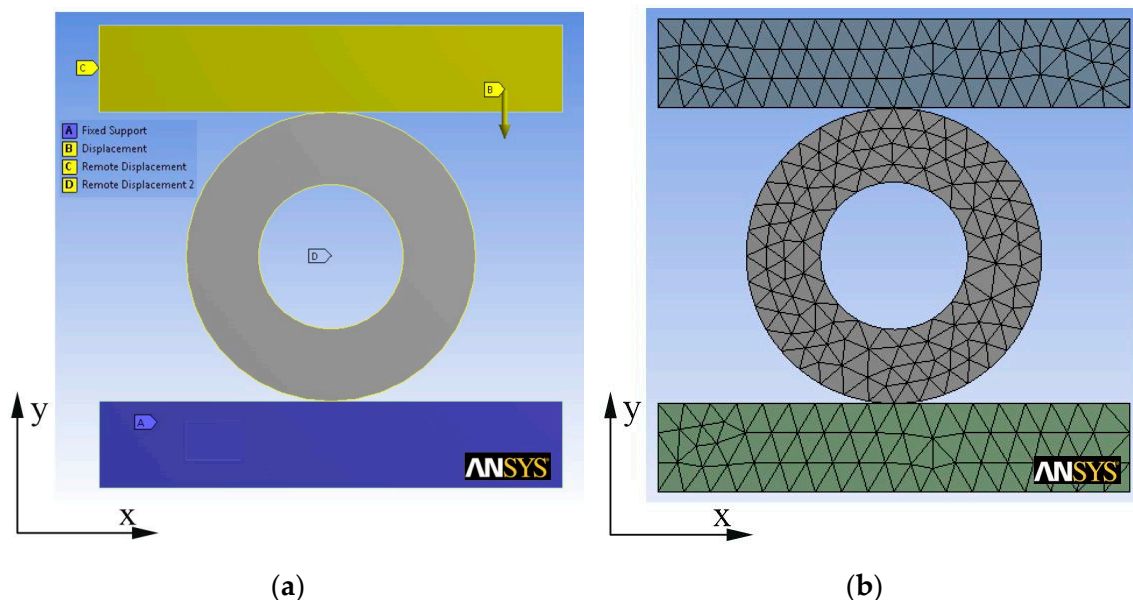
Material	Young's Module E [MPa]	Poisson's Ratio ν	Density ρ [kg·m ⁻³]
Polyurethane elastomer	13.5	0.45	1220
Steel	$2 \cdot 10^5$	0.30	7850

For the polyurethane elastomer, the material constants were determined experimentally, while in the case of the steel plates, the constants available in the ANSYS program were used [17].

3. Numerical and Experimental Determination of the Reaction Force and Absorption Energy for the Basic Model of a Cylindrical Fender

3.1. Numerical Calculations

The numerical calculations were performed for the 2D model of the cylindrical fender and steel plates (Figure 2b) using the finite element method in the Ansys program. The method of calculation for the 2D model also took into account a third dimension, $L = 500$ mm, in accordance with Figure 2a, which was assumed to be the same for both the fender and the steel plates. In the numerical calculations and in the experimental studies, assumptions were made in accordance with the PIANC recommendations. The movable steel plate was displaced by 50 mm in the direction indicated in Figure 2b, which accounted for 50% of the fender deflection. It was assumed for the purpose of the calculations that the movable steel plate moved towards the fixed plate by 50 mm ($y = -50$ mm) along the y axis, and that it was impossible for it to move along the x axis ($x = 0$); that the stationary plate could not be moved in the direction of either of the x ($x = 0$) or y ($y = 0$) axes; and that the cylindrical fender did not move in the x ($x = 0$) direction, and, owing to it being additionally blocked, was able to rotate around its axis. The computational model of the fender with the assumed boundary conditions and the method of its discretization are presented in Figure 3.

**Figure 3.** Numerical model of a cylindrical fender. (a) Boundary conditions; (b) Model discretization.

In all the models analyzed, discretization was based on TRI6 (six-node triangle) finite elements. For the model presented in Figure 3a, the division was made into 458 elements. Frictionless contact between the steel plates and the outer surface of the fender, and in the inner opening of the fender, was assumed.

The deformation of the cylindrical fender as a sequence of the stages of the movement of the movable steel plate, until it reached 50% of the fender deflection, is shown in Figure 4.

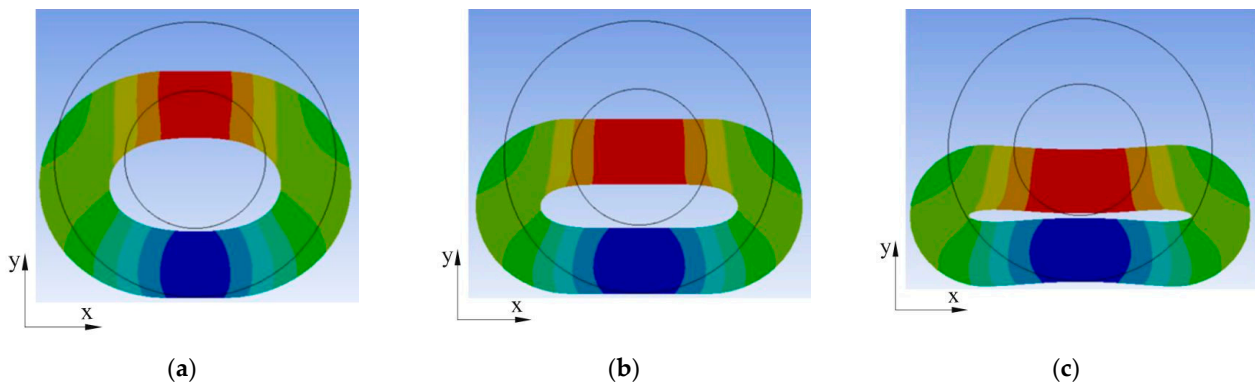


Figure 4. Stages of deformation in the numerical model of a cylindrical fender 100 mm \times 50 mm \times 500 mm until 50% deflection is achieved. (a) Stage 1, $y = -12.5$ mm; (b) Stage 2, $y = -25.0$ mm; (c) Stage 3, $y = -50.0$ mm.

For the adopted numerical model of a cylindrical fender, two basic operational parameters were determined—the reaction force, R (kN), at the contact point of the movable steel plate with the fender; and the absorption energy, E_a (J), as a function of the deflection, u (mm). The results of the calculations are presented graphically in Figure 5.

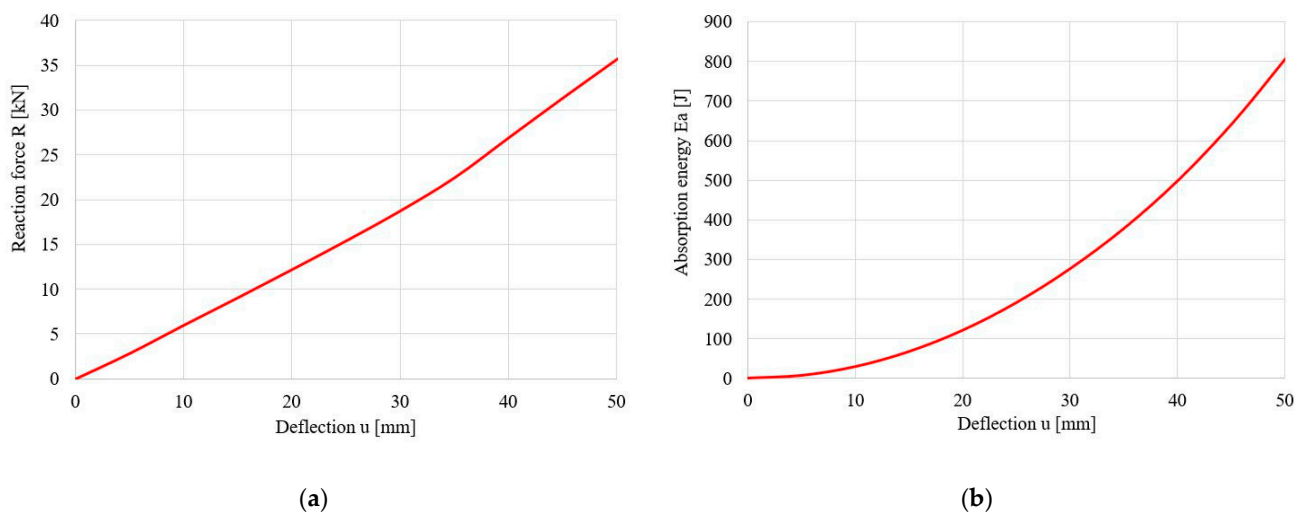


Figure 5. Numerically determined basic operational parameters of a cylindrical fender 100 mm \times 50 mm \times 500 mm. (a) Reaction force R (kN); (b) Absorption energy E_a (J).

3.2. Compression Test

In order to verify the accuracy of the numerical calculations, a load test was carried out using a cylindrical fender with dimensions in accordance with the markings shown in Figure 2a: $D = 100$ mm, $d = 50$ mm, and $L = 500$ mm. The material from which the experimental fender was made was a polyurethane elastomer, which was the same material as that used to determine the material constants for the numerical calculations (Table 1). The compression test was carried out using a testing machine in accordance with the PIANC recommendations to achieve a 50% deflection of the fender. In order to increase the squeezing surfaces situated between the fender and the jaws of the testing machine, the fender was placed between two additional steel plates. The stages of the experimental loading of the cylindrical fender placed between the two steel plates in the testing machine are shown in Figure 6.

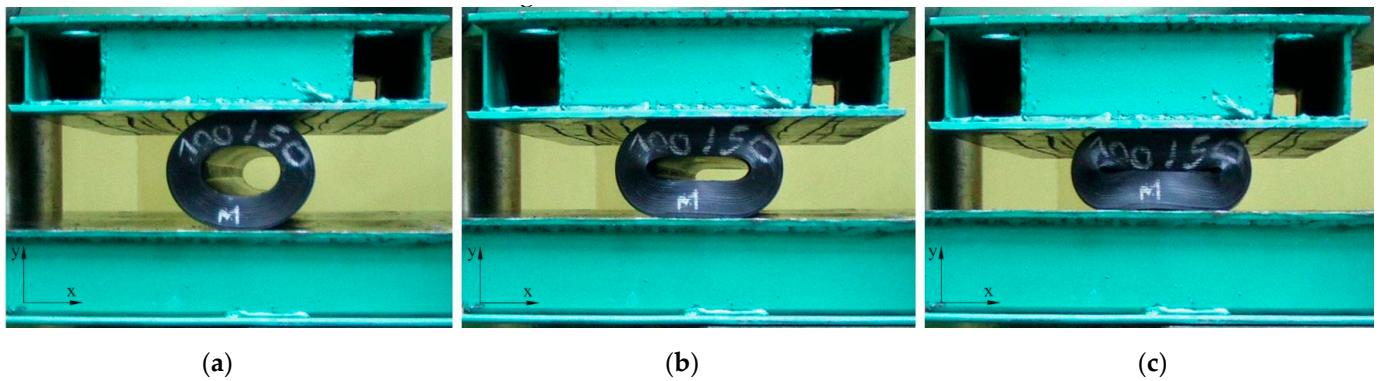


Figure 6. Stages of deformation of the experimental model of a cylindrical fender 100 mm \times 50 mm \times 500 mm until 50% deflection is achieved. (a) Stage 1, $y = -12.5$ mm; (b) Stage 2, $y = -25.0$ mm; (c) Stage 3, $y = -50.0$ mm.

For the adopted experimental model of a cylindrical fender, similarly to the numerical model, two basic operational parameters were determined—the reaction force, R (kN), at the contact point of the movable steel plate with the fender; and the absorption energy, E_a (J), as a function of the deflection, u (mm). The calculated results for the experimental tests are presented graphically in Figure 7. For verification purposes, Figure 7 summarizes the results of both the experimental and numerical calculations.

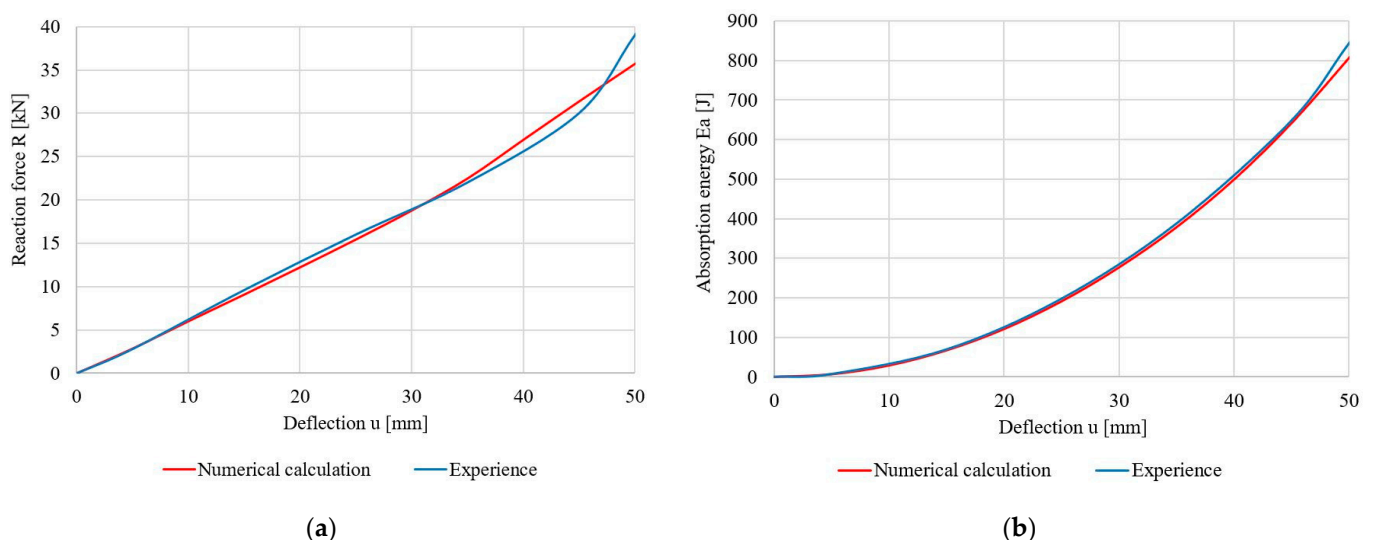


Figure 7. Experimentally determined basic operational parameters of a cylindrical fender 100 mm \times 50 mm \times 500 mm. (a) Reaction force R (kN); (b) Absorption energy E_a (J).

The numerically and experimentally determined values of the basic operational parameters of a cylindrical fender 100 mm \times 50 mm \times 500 mm showed high compliance.

4. Numerical Determination of the Reaction Force and Absorption Energy of Cylindrical Fenders with Additional Structural Elements Introduced

Structural elements constituted of additional openings of various diameters and quantities were introduced into the basic computational model of the cylindrical fender. Two different hole diameters were assumed: $\phi_1 = 10$ mm and $\phi_2 = 5$ mm. The holes were arranged symmetrically as a set of either of four or eight. The adopted virtual models of the cylindrical fender are presented in Figure 8. The remaining material assumptions and boundary conditions were the same as for the fender without additional holes considered in Section 2, for which the compliance of the values of the operational parameters determined numerically and experimentally was already confirmed.

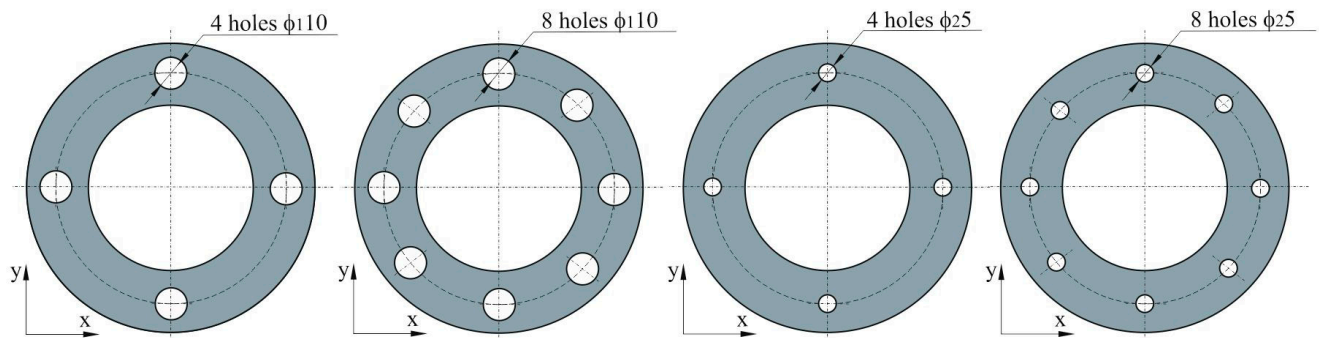


Figure 8. Cylindrical fender models (100 mm × 50 mm × 500 mm) with additional holes inserted.

The discretization of the fender models with the additional holes was based on finite elements—the same as in the model shown in Figure 3b. Due to the presence of additional holes of different geometric dimensions, the mesh had an increased density.

For the adopted models of a cylindrical fender with additional holes, two basic operational parameters were numerically determined using the finite element method—the reaction force, R (kN), at the contact point of the movable steel plate with the fender; and the absorption energy, E_a (J), as a function of the deflection, u (mm). The results of the calculations are graphically presented in Figure 9.

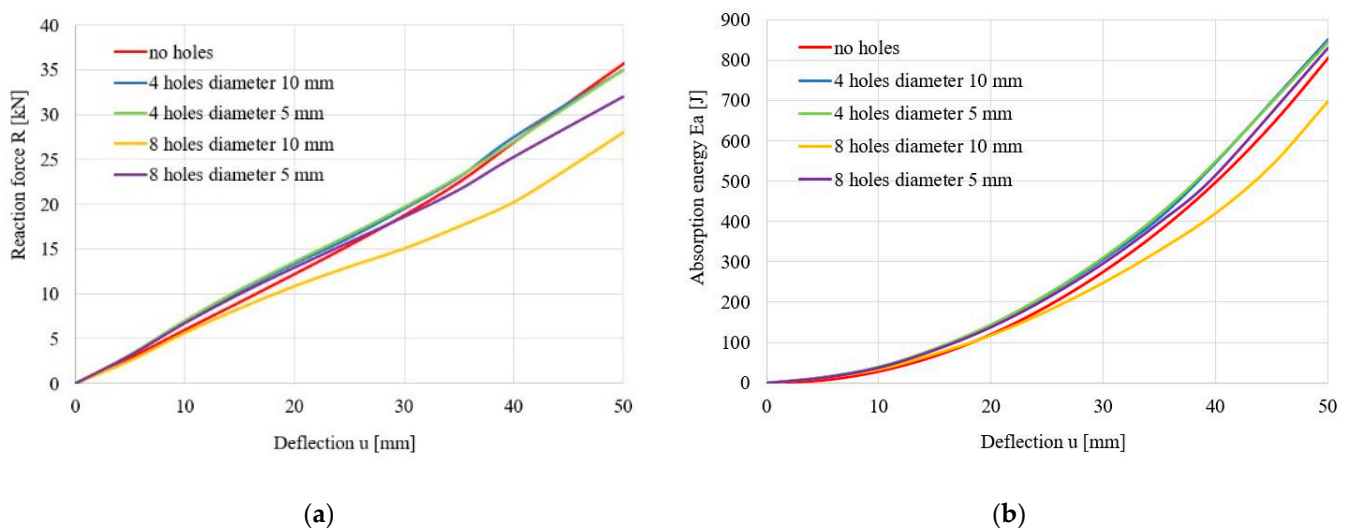


Figure 9. Numerically determined basic operational parameters of a cylindrical fender 100 mm × 50 mm × 500 mm with additional holes introduced. (a) Reaction force R (kN); (b) Absorption energy E_a (J).

5. Results and Discussion

The numerical calculations presented in this work were carried out in two stages. In the first stage, for a cylindrical fender 100 mm × 50 mm × 500 mm, the basic operational parameters were determined numerically using the finite element method in the Ansys software—reaction force, R (kN); and absorption energy, E_a (J), as a function of deflection, u (mm). The material constants determined for the polyurethane elastomer used for the production of fender devices were also adopted for the fender calculations. In order to confirm the accuracy of the calculations, a compression test was carried out using a testing machine on a fender with the same dimensions as those used in the adopted calculation model. The experimental fender was made of the same polyurethane elastomer for which the material constants were determined for use in the numerical calculations. The basic operational parameters determined experimentally for the cylindrical fender—the reaction

force, R (kN); and the absorption energy, E_a (J), as a function of deflection, u (mm)—showed high agreement with the same parameters determined numerically.

In the second stage, numerical calculations were carried out for a cylindrical fender with the dimensions $100\text{ mm} \times 50\text{ mm} \times 500\text{ mm}$, into which were introduced additional holes with diameters of $\phi_1 = 10\text{ mm}$ or $\phi_2 = 5\text{ mm}$ in sets of either four or eight. The impact of the introduced structural changes on the values of the basic operational parameters of the fender device was assessed. It was shown that the introduction of four holes with diameters of either $\phi_1 = 10\text{ mm}$ and $\phi_2 = 5\text{ mm}$ into the design of the fender had no significant effect on the reaction force, R (kN), or the absorption energy, E_a (J), as a function of the deflection, u (mm). Increasing the number of holes to eight, arranged symmetrically on the fender cross-section, affected the values of both performance parameters. In the model of the fender with eight holes of diameter $\phi_2 = 5\text{ mm}$, the reaction force R was reduced from 36 kN to 32 kN, compared to the fender without holes, while the absorption energy E_a which was 819 J for the fender without holes was practically unchanged. In contrast, for a fender with eight holes, each with a diameter ϕ_2 of 5 mm, the E_a was 829 J. The introduction of eight holes with a diameter of $\phi_1 = 10\text{ mm}$ significantly influenced the values of the operational parameters of the analyzed fender. The reaction force, R , decreased from 36 kN to 28 kN, while the absorption energy, E_a , decreased significantly from 819 J to 698 J. The results obtained from the calculations indicate that the more favorable design, in terms of the operational parameters of a cylindrical fender, is a greater number of additional holes with a smaller diameter. A greater number of holes with a larger diameter leads to a simultaneous significant reduction of the absorption energy, E_a (J).

It was confirmed that the introduction of additional structural elements to the cylindrical fender affected its operational parameters. The computational results obtained indicate the relevance of further work using numerical optimization tools to develop a cylindrical fender design with a low reaction force, R , value and a high absorption energy, E_a , value. Another suggested direction of research to optimize the shape of cylindrical fenders is to verify the possibility of introducing closed internal structures, which have not been used in the design of cylindrical fenders so far. Changing the structural design of the fenders may force the development of new technologies for their production. As of today, the technologies used for the production of fenders prevent the use of internal closures. This study confirms that the introduction of additional structural elements characterized by a low degree of complexity can improve the performance of fender devices. Expanding the range of the performance parameters of fenders can increase their versatility, and thus allow their use not only in the relationship between vessel and wharf, but also during the implementation of side-to-side operations on vessels, securing floating platforms, or on equipment used to ensure the safety of navigation.

Another area related to the use of fenders is in the diversification of electricity generation, which can be implemented through the construction of wind farms located in ocean waters. The construction of wind farms in such areas may require the use of highly versatile fenders, both at the stage of turbine assembly, and during operation. Vessels used to service wind turbines located in seawater often have to perform maneuvering and mooring activities within the turbine area in quite unfavorable weather conditions, which can lead to damage to both the units themselves and the turbine structures. Vessels serving wind turbines vary in terms of their size, so it is important that fender devices are characterized by universal performance parameters. Fenders used in offshore wind farms will indirectly contribute to ensuring the security of electricity supply in the area of clean energy generation.

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