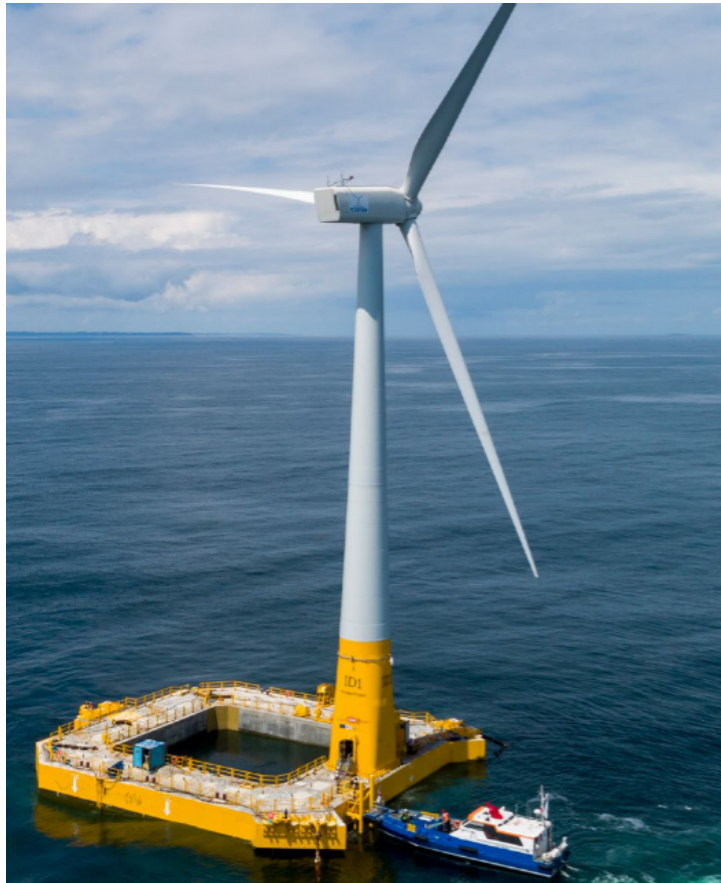


Numerical analysis in a barge-type FOWT in aNySIM

Investigation on affection of moonpool's damping lid factor in the system



1

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¹Image source: [17].

Preface

This thesis marks the completion of my Master's program in Offshore and Dredging Engineering at Delft University of Technology. It reflects both a technical exploration and a personal journey through the dynamic field of floating offshore wind energy, focusing on the complex hydrodynamic behavior of barge-type platforms with moonpools.

The research presented here would not have been possible without the guidance, support, and encouragement of several individuals and institutions. I would first like to express my sincere gratitude to my graduation committee: Dr. Oriol Colomes Gene, for his critical insights and support as the chair of this project, and Dr. Alex van Deyzen, my supervisor from Haskoning, for his continuous guidance and practical perspective throughout the daily supervision of this thesis.

I am also deeply grateful to my colleagues, and especially to Jeroen van de Sande, whose support was essential in helping me become familiar with the numerical tools used in this study, friends, and family for their encouragement throughout this journey. A sincere thank you goes to my girlfriend Diony, for her patience, understanding, and emotional support during the challenging phases of this work. Her presence made the tough days easier and the good ones even more meaningful.

Working on this thesis has deepened my passion for offshore engineering, especially in the field of floating wind turbines and their interaction with the marine environment. I hope this work contributes meaningfully to the understanding and optimization of floating wind platform design.

Delft, July 2025
Manolis

Abstract

This thesis investigates the hydrodynamic behavior of a barge-type Floating Offshore Wind Turbine (FOWT) equipped with a central moonpool, with particular focus on the influence of the damping lid method, a numerical approach used to approximate viscous damping in potential flow simulations. The damping lid factor ε was varied systematically to evaluate its impact on moonpool resonance, platform motions, and mooring system loads under both idealized and realistic sea states.

Hydrodynamic coefficients and motion response amplitude operators RAOs were first analyzed in the frequency domain using DIFFRAC, revealing strong sensitivity of the system to internal resonance modes when no damping is applied. The time domain simulations were then performed in aNySIM to assess the mooring tensions and the motion behavior under regular and irregular wave conditions. A minimum damping lid factor of $\varepsilon = 0.035$ was identified as sufficient to suppress unrealistic resonance amplification.

Finally, a case study for a North Sea scenario was carried out to compare the performance of the system between $\varepsilon = 0.035$ and $\varepsilon = 0.1$ in terms of serviceability and operational availability. Results showed that once the critical damping threshold is reached, the system response becomes relatively insensitive to further increases in ε .

The findings demonstrate that the damping lid is an effective and practical tool for modeling moonpool effects in barge-type FOWTs, and that a carefully selected damping factor for this specific barge type, can eliminate the need for more computationally intensive viscous simulations when evaluating global system behavior.

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Chapter 1

Introduction

The transition to renewable energy sources has become a global priority due to climate change and the significant demand for sustainable energy solutions. Offshore wind energy has emerged as a key technology in this transition, offering vast potential to generate clean and efficient energy. While fixed-bottom wind turbines dominate the current offshore wind market, their deployment is constrained by sea depths and coastal occupation. This led to the development of Floating Offshore Wind Turbines (FOWTs), which can deal better with wind energy in deeper waters, where the wind is stronger and more consistent.

Among the various designs for FOWTs, barge-type floaters stand out due to their structural simplicity, cost-effectiveness, and adaptability to a wide range of sea depths. However, their performance is influenced by complex hydrodynamic phenomena, particularly those associated with moonpools. Moonpools are vertical openings within floating structures, originally designed for offshore drilling operations, but now integrated into barge-type FOWTs to improve stability and reduce hydrodynamic loads. Despite their benefits, moonpools exhibit resonance effects, including sloshing and piston modes, which can adversely affect the overall performance of the floating structure.

To address these challenges, numerical tools such as DIFFRAC and aNySIM have been developed to simulate the hydrodynamic behavior of FOWT systems. These tools rely on potential flow theory, which often overpredicts resonance phenomena due to its inability to fully capture viscous effects and flow separation. To mitigate this limitation, the damping lid method is widely used to simulate viscous damping within moonpools. This method introduces an artificial damping factor, but its effects on moonpool resonance and the dynamic behavior of the coupled FOWT system remain insufficiently understood.

This thesis aims to investigate the influence of the damping lid factor on the hydrodynamic performance of a barge-type FOWT system. Specifically, it will examine how variations in the damping lid factor impact the natural frequencies of moonpool resonance modes, the loads and motions of the mooring system, and the overall system behavior under different environmental conditions. The findings of this research will provide valuable insights for improving the design and analysis of FOWT systems, enabling their efficient deployment in offshore wind energy markets.

By analyzing the system in both frequency and time domain, and under idealized and realistic sea conditions, the research scope is to:

- Identifying the natural frequencies of piston and sloshing modes within the moon-

pool using analytical and numerical methods.

- Evaluating the effects of different damping lid factors on moonpool resonance and its interaction with the coupled FOWT system.
- Assessing the influence of damping lid factors on mooring loads and motions under a range of sea states.
- Establish the minimum effective ε value required to suppress unrealistic resonance amplification.
- Evaluate system performance under operational sea states once a stable damping lid factor is applied.

The results of this study will contribute to a deeper understanding of moonpool dynamics and inform the optimization of potential-based numerical tools for simulating FOWT systems. By addressing these critical challenges, this research will support the broader adoption of floating wind technologies, paving the way for a more sustainable energy future.

The structure of this report is as follows: Chapter 2 provides a theoretical background, including the fundamentals of moonpool resonance and the damping lid method and the problem definition. Chapter 3 presents the numerical setup, outlining the modeling tools and simulation procedures used. Chapter 4 contains the main numerical results, divided into hydrodynamic analysis and mooring response under regular and irregular waves. Chapter 5 applies the findings in a case study under real sea conditions. Finally, Chapter 6 reflects on the research objectives, presents conclusions, and outlines recommendations for future work.

Chapter 2

State of the Art

2.1 Problem definition

Offshore Wind

The European's net zero goals for many country-members have led to a rapid and significant expansion of offshore wind driving forward the transition from high carbon energy sources coming from oil and gas to low carbon and emission technologies such as wind power [8], [15]. According to the Global Wind Energy Council, offshore wind will become a crucial selection for global decarbonization. Wind energy not only provides electricity but also enables the production of green hydrogen.[15]

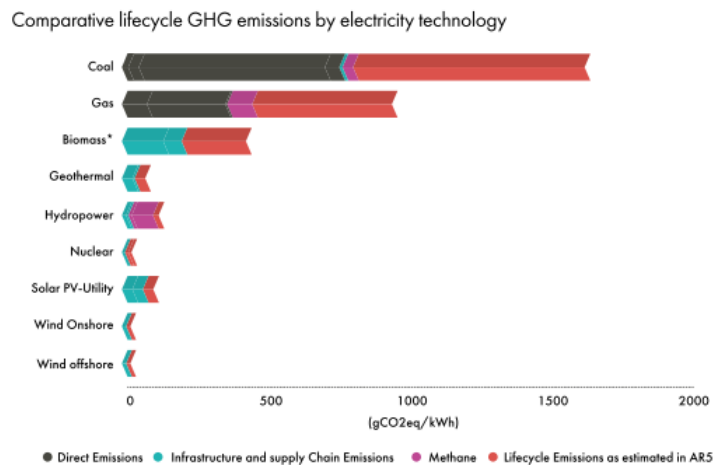


Figure 2.1: Life cycle GHG emissions by electricity technology [9]

As can be seen in the Figure 2.2, there are massive requirements for the installation of offshore wind parks. In 2020, 35 GW were installed with expectations to install 270 GW by 2030 and 2000 GW by 2050 [15] - an ambition that demands a change and diverse look in support structures.

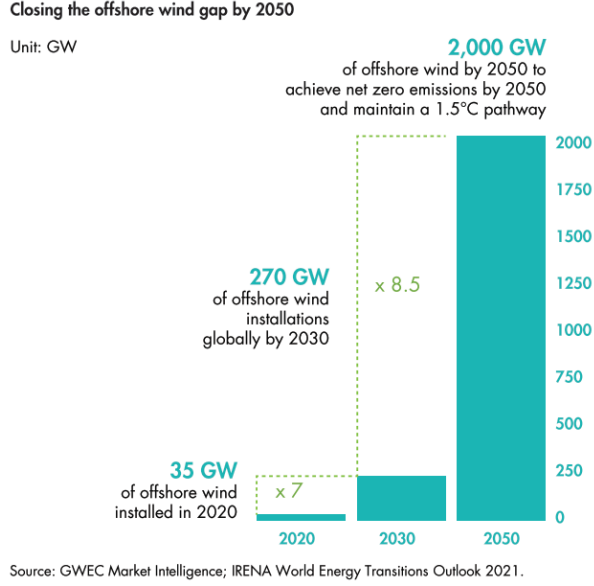


Figure 2.2: Offshore wind demands by 2050 [15]

Floating Wind

The current offshore wind market is dominated by fixed-bottom wind parks, which make up 80% of the market. Examples of existing wind parks include monopiles, jackets and concrete based support structures. However, for many countries, these types of structures are not feasible for geomorphological conditions [15],[16]. For instance, the Mediterranean Sea, where the sea depth exceeds 70m in many sites, bottom-fixed structures are impractical. Additionally, coastal areas are already occupied by these structures. According to Emma C. Edwards et al (2023) [8], wind energy is stronger and more stable in deeper waters. This is where floating offshore wind platforms, especially for deepwater applications, become essential.

Floating offshore platforms are categorized into four main types: Spar platforms, barge-type platforms, semi-submersibles, and tension-leg platforms. Their primary function is to maintain stability in water while minimizing pitch motions, ensuring efficient operation in offshore environments.

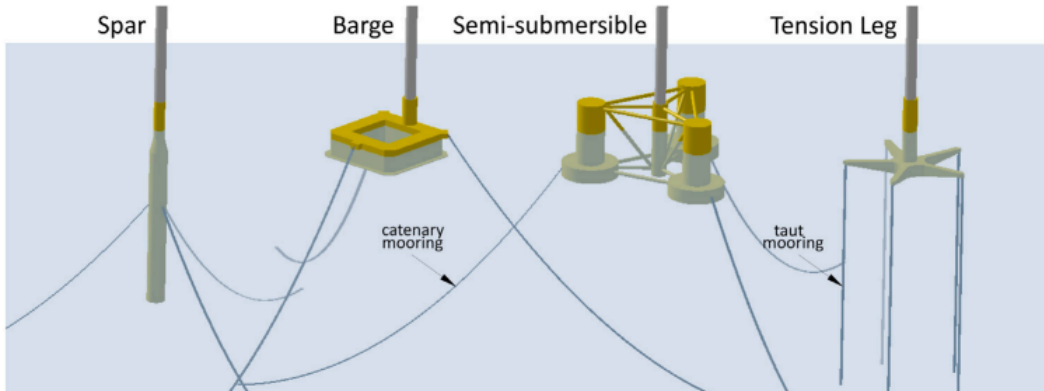


Figure 2.3: Timeline of floating wind turbine deployments [8]

Spar-type platforms are cylindrical structures characterized by a deep draft and ballast system located at the bottom, which provides stability by lowering the center of gravity [38]. The wind turbine is positioned centrally at the top of the cylindrical support structure to optimize balance and efficiency.

Barge-type platforms are classified into two subtypes: those with a moonpool and those without. Typically, these platforms have a simple geometric design, either rectangular or square, and rely on their large surface area to provide stability. The inclusion of a moonpool can further enhance motion damping by allowing controlled water flow through the structure [15].

Semi-submersible platforms consist of multiple submerged hulls or pontoons connected by columns [27]. Their design enables them to achieve stability through distributed buoyancy, making them less susceptible to wave-induced motions. This category of platforms is commonly used for offshore wind applications due to their favorable hydrodynamic performance [10].

Lastly, tension-leg platforms (TLPs) are relatively flat structures equipped with a submerged buoyancy tank. Static stability is achieved through taut mooring lines that restrict vertical motion while allowing horizontal flexibility [38]. This design minimizes movement while maintaining structural integrity, making TLPs a viable solution for deep water offshore wind installations. [10]

Despite the promising outlook, not all floating platform types are equally explored. Most commercial projects have relied on semi-submersible or spar designs (Figure 2.3), with barge-type platforms receiving less attention and only more recently being deployed, such as in the FLOATGEN project in France (2018). This under-representation suggests a research opportunity to further explore and optimize barge-type designs for floating wind.

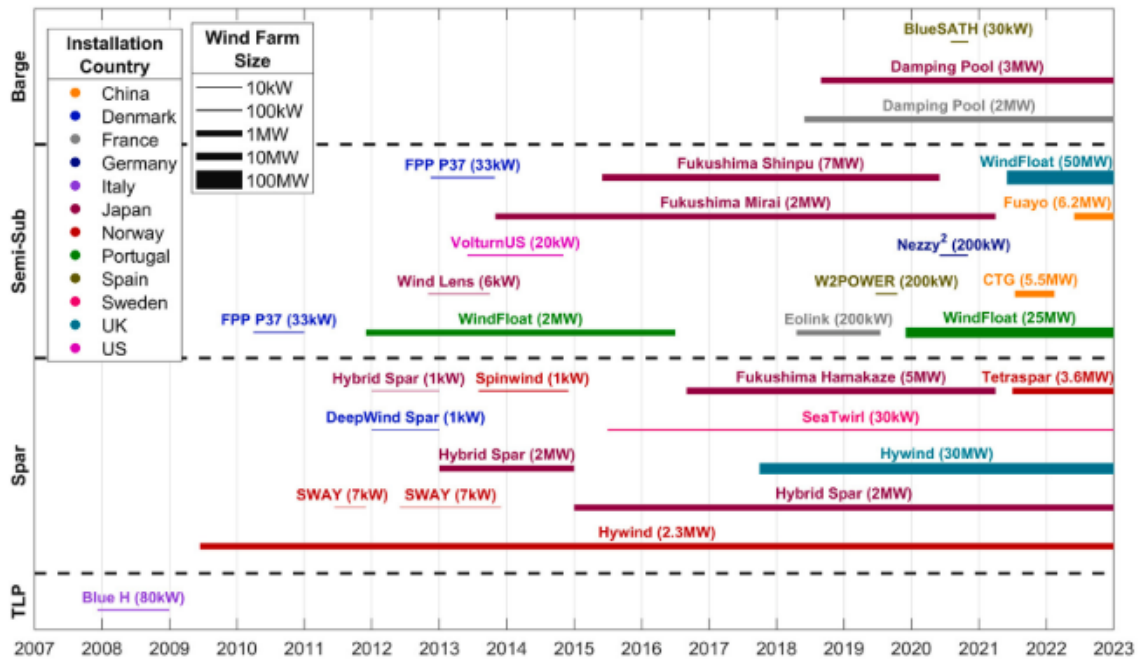


Figure 2.4: Timeline of floating wind turbine deployments [8]

Existing literature and global trends acknowledge the potential of floating wind, but there's limited focus on barge-type platforms, particularly their integration with moonpools and mooring systems under realistic sea conditions.

Barge type floater

Barge-type platforms are increasingly considered for Floating Offshore Wind Turbines (FOWTs) due to their large deck area, cost-effectiveness, and ease of construction. Compared to other floater types like semi-submersibles or spars, barge-type FOWTs offer stability in moderate wave conditions and allow for efficient turbine installation and maintenance. However, their hydrodynamic behavior, particularly in extreme environments, requires motion reduction strategies to mitigate excessive wave-induced motions [23].

One effective solution is the integration of moonpools into the barge design. A moonpool is an open cavity within the hull that modifies wave interactions by dissipating energy through fluid resonance effects. In barge-type FOWTs, moonpools help reduce vertical movements (heave) and improve the overall stability of the structure by acting as a natural damping mechanism [7]. Studies have shown that properly tuned moonpool dimensions can significantly lower platform responses, leading to enhanced turbine performance and structural longevity [41].

The hydrodynamic performance of barge-type platforms with moonpools is highly dependent on parameters such as added mass, radiation damping, and stiffness characteristics. Numerical simulations and experimental studies play a critical role in optimizing these designs to ensure wave load mitigation and station-keeping efficiency. As offshore wind energy moves toward deeper waters, the application of barge-type floaters with advanced hydrodynamic features like moonpools presents a promising avenue for cost-effective and scalable floating wind solutions [41].

Moonpool hydrodynamics

Moonpools, while originally used in drilling ships, are increasingly integrated into barge-type floating platforms for wind energy. They offer both structural and hydrodynamic advantages, such as damping incoming waves and improving motion stability. However, they also introduce complex internal resonance phenomena—namely sloshing and piston modes—that can amplify loads and motions if not well understood.

In 1984, A.B. Aalberts, a Project engineer at MARIN, developed a mathematical model to describe these relative motions inside the moonpool. Seventeen years later, Molin (2001), provided foundational insights by modeling these modes using linearized potential flow theory [33]. His work, while critical, assumed an idealized environment (e.g., motionless support, infinite depth), which limits its direct applicability to real-world offshore wind systems.

Following Molin's work, the increase in size of drillships and offshore construction vessels—which led to larger moonpools—drew further attention to moonpool resonance. Subsequent studies, including those by Faltinsen, Rognbakke & Timokha 2007 [11] and Faltinsen & Timokha 2015 [12], have provided deeper insights into the resonance phenomena of barge to barge gap and moonpools.



Figure 2.5: Piston mode (left) and Sloshing mode (right)

In 2018, Molin presented an extension on his earlier research on the natural frequencies of piston and sloshing modes, calculating these values with respect to both finite depth and beam length [34].

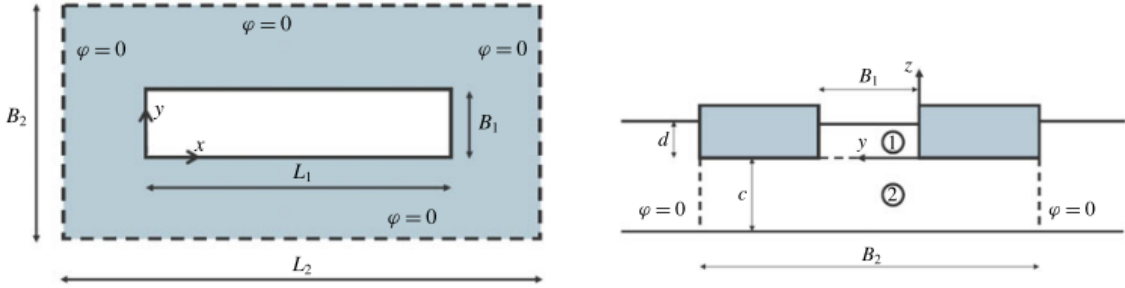


Figure 2.6: Moonpool sketch - Geometry [34]

The velocity potential in the moonpool is described as a three-dimensional flow to address scenarios with wide moonpools. In the upper sub-domain (the moonpool), the velocity potential is given as [34]:

$$\begin{aligned} \phi_1(x, y, z) = & A_{00} + B_{00} \frac{z}{d} + \sum_{m=0}^{\infty} \sum_{n=0}^{\infty} (m, n) \neq (0, 0) \\ & \times [A_{mn} \cosh(\nu_{mn} z) + B_{mn} \sinh(\nu_{mn} z)] \cos(\lambda_m x) \cos(\mu_n y), \end{aligned} \quad (2.1)$$

where:

$$\lambda_m = \frac{m\pi}{L_1}, \quad \mu_n = \frac{n\pi}{B_1}, \quad \nu_{mn}^2 = \lambda_m^2 + \mu_n^2. \quad (2.2)$$

In the lower sub-domain (beneath the moonpool), the velocity potential is similarly expressed with different wave number notations [34]:

$$\phi_2(x, y, z) = \sum_{p=1}^{\infty} \sum_{q=1}^{\infty} C_{pq} \frac{\cosh(\gamma_{pq}(z + c))}{\cosh(\gamma_{pq}c)} \sin(\alpha_p(x + a)) \sin(\beta_q(y + b)), \quad (2.3)$$

where:

$$\alpha_p = \frac{p\pi}{L_2}, \quad \beta_q = \frac{q\pi}{B_2}, \quad \gamma_{pq}^2 = \alpha_p^2 + \beta_q^2. \quad (2.4)$$

Here, L_1 and B_1 represent the dimensions of the moonpool, while L_2 and B_2 correspond to the dimensions of the lower sub-domain. Constants a and b are defined as:

$$a = \frac{L_2 - L_1}{2}, \quad b = \frac{B_2 - B_1}{2}. \quad (2.5)$$

The process for matching the velocity potentials and their vertical derivatives at the interface between the sub-domains is consistent with similar cases, ensuring continuity and smooth transition [34].

Damping Methods

It is observed that, most of the time, these resonance effects are overpredicted compared to reality. Apart from this resonance phenomena, moonpools have been noted to dampen the forces of incoming excitation waves due to eddy-making damping [13]. Several studies have emphasized the importance of including viscous effects and flow separation in the moonpool analyses, as potential flow theory does not account for these phenomena and tends to overpredict the resonance of piston and sloshing modes, particularly for sharp-edged moonpools.

Faltinsen et al. [11] proposed a method based on a domain-decomposition scheme, which led to a system of integral equations on the transmission interface and the goal of their study was to investigate the effect of vortex shedding. However, they did not account for current and forward velocity, prompting Fredriksen et al [14] to incorporate these factors into their analysis. Finally, Chen, X.B., 2005 [4], introduced an empirical approach to improve potential flow models by applying artificial damping, known as a numerical damping lid, to the free surface of the moonpool or gap.

While damping lid techniques are adopted to simulate viscous effects, there is a lack of investigation into how varying damping lid factors impact resonance frequencies, platform motions, and mooring loads in barge-type FOWTs.

Recent reaserch

As mentioned above, barge-type floaters are usually rectangular boxes with small draughts compared to other existing types of floaters. This characteristic gives them the advantage of being deployable in a wide range of sea depths. Compared to conventional barge-type floaters, the inclusion of a moonpool offers multiple benefits, such as improved sea-keeping performance, reduced construction costs and the potential to integrate WECs (wave energy converters) or fish cages. [26].

In 2007, J.M. Jonkman and M.L. Buhl Jr., conducted a coupled analysis of a barge-type floater with a catenary mooring system and a 5MW wind turbine. They found that large pitch motions of the barge resulted in significant loads on the floating turbine. To mitigate these motions, they suggested design modifications, such as incorporating moonpools or skirts[21]. In 2015, Beyer et al. presented a numerical analysis of a barge-type floater using multibody system and CFD methods, comparing the results with experimental data. They concluded that there is still a need for numerical methods capable of modeling this

type of floaters quickly as multiple environmental conditions must be evaluated [3].

In 2016, Lucie Guignier et al. presented a methodology to calculate the internal loads produced by wind and waves on a barge-type floater. In order to capture the viscous effects inside the moonpool, they included additional damping that was calibrated with a basin model [18]. In 2019, Kosasih et al. compared the NEDO 3 MW barge-type floater with two wave tank experiment models, conducting an analysis to understand the dynamic response behavior of the wind turbine in both regular and irregular waves. Their findings demonstrated good consistency with the experimental results [24].

In 2020, Ko Matias Adrian Kosasih, conducted a numerical analysis to investigate the dynamic behavior of a barge-type equipped with moonpool and skirts, moored with a catenary mooring system. In order to simulate moonpool's viscous effects used the damping lid method applying a 5% damping factor [25]. In 2021, Lei Tan et al, examined the mean wave drift forces on a barge-type floater moored by chains using the panel code WAMIT. To simulate the viscous effects that WAMIT does not inherently model, they employed the damping lid method applying arbitrarily damping values ranging from 0 to 0.08. This approach allowed them to continue their study and compute the mean drift forces of sway. [40].

In 2021, Tzu-Ching Chuang et al. presented a study in a barge-type floater in Taiwan under wind and wave loads at a depth of 50 m using a numerical model based on ANSYS AQWA and potential theory. This paper does not refer to how they simulated viscous damping inside the moonpool [5]. In 2023, Yiming Zhou et al, proposed a novel design for a barge-type floater optimized for shallow water conditions. This design featured four moonpools and skirts to support a 10 MW wind turbine developed by TU Denmark. To account for viscous damping effects of the moonpools, they added 8% critical damping to the system [43].

While researchers acknowledge the importance of damping for moonpool resonance, there is no clear consensus on how to choose or calibrate the damping lid factor. This thesis directly addresses this ambiguity by proposing a structured numerical investigation of how varying the damping factor influences the hydrodynamic response of the floater and its mooring system.

2.2 Thesis scope

According to the literature review, most previous researchers have used the damping lid method in their analyses to simulate the viscous damping effects inside the moonpools by incorporating damping values derived from existing experiments. The scope of this research is to conduct a comprehensive numerical analysis using DIFFRAC and aNySIM numerical tools developed by MARIN.

This analysis aims to investigate the influence of different damping lid factors on the characteristics of moonpool as well as on the loads and motions of the mooring system. Additionally, it is essential to examine the system's response under various sea conditions. The findings will aid future researchers and designers who rely on potential-based numerical tools to gain a deeper understanding of how the selection of this factor impacts their system.

The following research question has been defined to reach this scope:

“How does the damping lid factor affect the hydrodynamic behavior of a coupled barge-type FOWT system and its mooring loads and motions under various sea states?”

Here are some sub-questions that help to answer the main question:

1. What is the sensitivity of moonpool characteristics to variations in damping lid factor?
2. How do different damping lid factors affect the loads and motions on mooring systems?
3. What are the responses of the system under operational conditions in real sea states when different damping lid factors are applied?

2.3 Methodology

This section outlines the methodology and steps that will be followed to comprehensively address the main research question. It outlines the framework, tools and techniques employed to ensure a structured and systematic approach to achieve the objectives of this study. A short and clearer description of the entire procedure to be followed in this study is shown in the following figure:

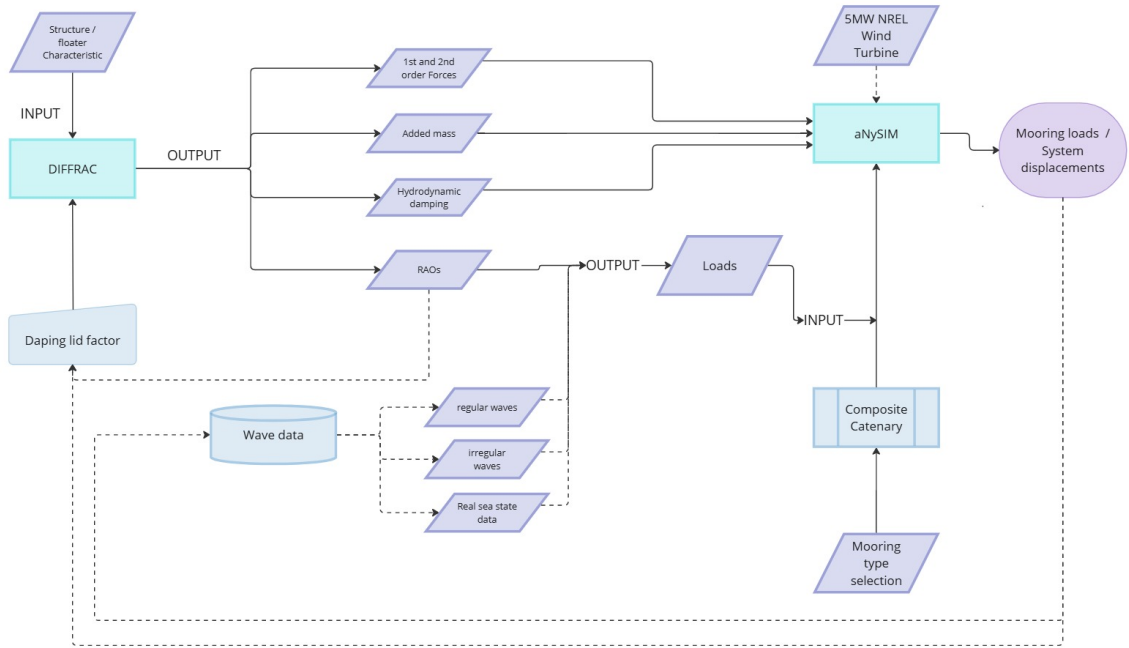


Figure 2.7: Flowchart of the procedure

The numerical framework illustrated in the flowchart represents the process of analyzing the hydrodynamic response and mooring loads of a floating offshore wind turbine (FOWT) in various damping lid factors.

This research followed a structured numerical approach to evaluate how different values of the damping lid factor (ε) affect the hydrodynamic performance of a barge-type Floating Offshore Wind Turbine (FOWT) equipped with a moonpool. The methodology was divided into three main phases: hydrodynamic analysis in the frequency domain, time-domain simulations of the full system response, and a final case study under realistic environmental conditions.

The study began with the definition of the floating system, which included a square-shaped barge-type floater with a central moonpool, a 5 MW NREL wind turbine, and a symmetric layout of catenary mooring lines. Two simulation tools developed by MARIN were used throughout the process: DIFFRAC for linear potential flow analysis in the frequency domain, and aNySIM for coupled time-domain simulations.

In the first phase, DIFFRAC was used to compute added mass, radiation damping, and wave excitation forces for multiple values of the damping lid factor ranging from 0 to 0.1. Response Amplitude Operators (RAOs) were generated for each motion degree of freedom to identify how internal resonance modes in the moonpool were affected by different ε values.

In the second phase, selected values of ε were applied in aNySIM to simulate the system's time-domain behavior under both regular and irregular wave conditions. The model included first-order and second-order wave loads and a detailed representation of the mooring system.

The influence of the damping lid was evaluated based on platform motion and mooring line tensions. Through sensitivity analysis, a threshold damping lid value of $\varepsilon = 0.035$ was identified as the minimum level required to suppress exaggerated resonance behavior. This value, along with $\varepsilon = 0.1$, was used in the final case study to compare system performance under real sea states.

In the final phase, a case study was performed for a site near North Scotland using JON-SWAP wave spectrum using weather data of the location. The two selected damping lid values were tested to assess their effect on ultimate limit states, serviceability thresholds, and annual operability. The results were used to evaluate the practical implications of damping assumptions on floating wind performance.

Chapter 3

Model set-up

This chapter presents a detailed analytical description of the methodology followed in this study, outlining the key steps taken to investigate the hydrodynamic behaviour of the barge-type floating offshore wind turbine (FOWT) system. It describes the selection process for the floating platform type, wind turbine type and mooring system, as well as the numerical tools used to conduct the analysis. For this thesis DIFFRAC was chosen for diffraction analysis and aNySIM as a time domain tool. The procedure includes the computation of hydrodynamic parameters, response amplitude operators (RAOs), mooring loads and system motions with a particular focus on the impact of the damping lid factor.

In addition, this chapter highlights the simplifications and assumptions made throughout the study. These considerations were necessary to balance computational efficiency and time with accuracy, ensuring that the results remain both reliable and practical. By outlining these choices, this chapter provides a structured overview of the analytical process. The subsequent sections delve into each step in detail, demonstrating how the numerical simulations and modeling techniques contribute to answering the research question.

3.1 Barge geometry selection

According to methodology, the first step is to define the barge dimensions and characteristics, which form the basis for the hydrodynamic and time domain analysis. For this thesis, a rectangular barge-type floater has been selected with a length and width of 60 meters each, a height of 15 meters, a draft of 10 meters and a mass of approximately 19,900 tonnes. These dimensions were chosen to represent a realistic floating offshore wind turbine platform.

To further investigate the influence of floater design on hydrodynamic behavior, a floater configuration is considered. The configuration features a moonpool of 40 meters by 40 meters placed exactly in the middle of the barge.

The barge configuration designed in 3ds Max 2025 and illustrated in the figure below:

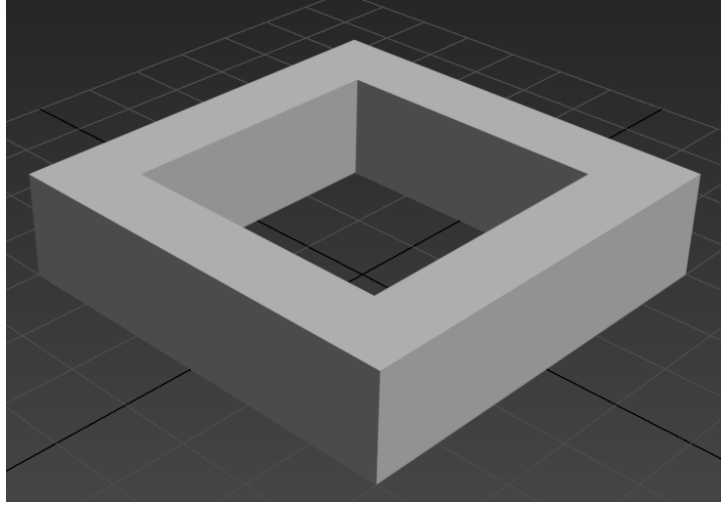


Figure 3.1: Barge with a moonpool 40mX40m

Table 3.1: Barge-type platform properties (moonpool 40m X 40m)

Parameter	Value
Size (W * L * H) (m)	60 * 60 * 15
Moon pool (W * L * H) (m)	40 * 40 * 15
Draft (m)	10
Displacement (m ³)	19,900
XCG (m)	0.00
YCG (m)	0.00
ZCG below sea level (m)	-4.86

For the calculation of the roll, pitch, and yaw radius of gyration, the corresponding moments of inertia were first determined as follows [31]:

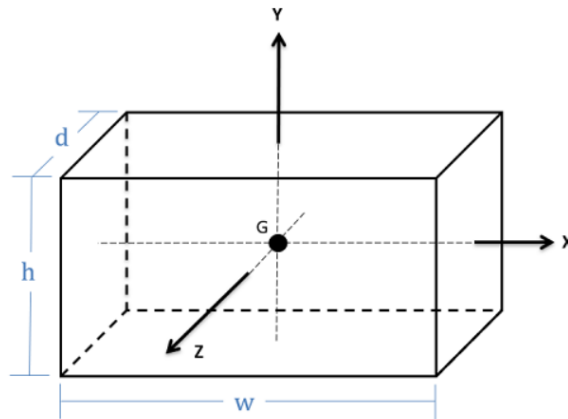


Figure 3.2: Rectangular Prism

The moments of inertia for a rectangular prism about its principal axes are given by the following equations:

$$I_{xx} = \frac{1}{12}m(h^2 + d^2) \quad (3.1)$$

$$I_{yy} = \frac{1}{12}m(d^2 + w^2) \quad (3.2)$$

$$I_{zz} = \frac{1}{12}m(h^2 + w^2) \quad (3.3)$$

where:

- m : Mass of the prism
- h : Height of the prism
- d : Depth of the prism
- w : Width of the prism

The radius of gyration k_{xx} , k_{yy} and k_{zz} can be calculated using the following equation:

$$k_{xx} = \sqrt{\frac{I_{xx}}{A}} \quad (3.4)$$

$$k_{yy} = \sqrt{\frac{I_{yy}}{A}} \quad (3.5)$$

$$k_{zz} = \sqrt{\frac{I_{zz}}{A}} \quad (3.6)$$

where:

- I_{xx} is the moment of inertia around x axis.
- I_{yy} is the moment of inertia around y axis.
- I_{zz} is the moment of inertia around z axis.
- A is the cross-sectional area.

For current barge with length and width of 60m and draft of 10m the radius of gyration calculated as followed:

$$k_{xx} = 17.32 \, m, \quad k_{yy} = 17.32 \, m, \quad k_{zz} = 24.48 \, m \quad (3.7)$$

The selected barge configurations were designed without skirts or complex geometrical features to reduce computational complexity and save calculation time. Simplifying the floater's geometry allows for a more efficient numerical analysis while still capturing the essential hydrodynamic behaviors relevant to this study.

3.2 Moonpool's natural frequencies

At this point, it is essential to determine the natural frequencies of both the piston mode and the sloshing modes within the moonpool. This step provides an initial estimate of the frequency ranges where resonance effects are likely to occur. To achieve this, the formulation proposed by Molin is applied, as it takes into account both finite water depths and the finite length of the structure, making it particularly suitable for this analysis.

Molin's method considers a three-dimensional flow inside the moonpool, which is especially relevant for cases where the moonpool has a considerable width. This aspect is crucial as it can significantly influence the overall hydrodynamic behavior. Based on this approach, the natural frequency of the piston mode is given by the following equation:

$$\omega_{00}^2 \simeq \frac{g}{d} \frac{1}{1 + C_{00}} \quad (3.8)$$

where,

$$C_{00} = \frac{4}{B_1 L_1 B_2 L_2 d} \sum_{p=1}^{\infty} \sum_{q=1}^{\infty} \frac{I_p^2 J_q^2}{\gamma_{pq} \tanh(\gamma_{pq} c)} \quad (3.9)$$

and

$$I_p = \int_0^{L_1} \sin \alpha_p(x + a) dx, \quad J_q = \int_0^{B_1} \sin \beta_q(y + b) dy \quad (3.10)$$

with

$$\alpha_p = \frac{p\pi}{L_2} \quad (3.11)$$

$$\beta_q = \frac{q\pi}{B_2} \quad (3.12)$$

$$\gamma_{pq}^2 = \alpha_p^2 + \beta_q^2 \quad (3.13)$$

For longitudinal sloshing modes he derived these equations for natural frequencies:

$$C_{m0} = \frac{8\lambda_m}{B_1 L_1 B_2 L_2} \sum_{p=1}^{\infty} \sum_{q=1}^{\infty} \frac{K_{pm}^2 J_q^2}{\gamma_{pq} \tanh(\gamma_{pq} c)} \quad (3.14)$$

with

$$K_{pm} = \int_0^{L_1} \sin \alpha_p(x + a) \cos \lambda_m x dx \quad (3.15)$$

$$J_q = \int_0^{B_1} \sin \beta_q(y + b) dy \quad (3.16)$$

These frequencies were calculated developing a python script to provide an initial view of where resonance is expected to occur in RAOs and below is shown the tables with these values.

Table 3.2: Moonpool's Natural Frequencies for moonpool 40m X 40m

	Modes	ω_n (rad/s)
Piston Mode	1	0.6731
Sloshing Modes	1	0.9337
	2	1.2477
	3	1.5213
	4	1.7557
	5	1.9627
	6	2.1500
	7	2.3223
	8	2.4827
	9	2.6333
	10	2.7757

3.3 DIFFRAC Modeling

The DIFFRAC tool, developed by the Maritime Research Institute Netherlands (MARIN), is an advanced wave diffraction analysis program. It is specifically designed to calculate the first and second order wave loads and response amplitude operators (RAOs) of free-floating multibody structures in regular wave conditions. The Response Amplitude Operator (RAO) expresses the ratio of structural motion to wave amplitude:

$$H(\omega) = \frac{u_a(\omega)}{\zeta_a(\omega)} \quad (3.17)$$

where $u_a(\omega)$ is the motion amplitude and $\zeta_a(\omega)$ is the wave amplitude. By employing a three-dimensional source distribution method, DIFFRAC effectively solves the linearized velocity potential problem, enabling precise computation of fluid pressures and waved drift loads, which are essential for the design of mooring and dynamic positioning systems in offshore structures. Suitable for both shallow and deep-water applications, DIFFRAC has been extensively validated against physical model tests and is widely used for various offshore platforms. Can determine the natural frequencies of a restrained system and account for viscous damping using the damping lid method which is analyzed in more detail in following subsection [28].

3.3.1 Wave-Induced Motion of Structures

Offshore structures experience six degrees of freedom (DOFs): three translations (surge, sway, heave) and three rotations (roll, pitch, yaw). Their motion response is determined by fundamental governing equations, which DIFFRAC utilizes to calculate and analyze them [28]:

$$\mathbf{X} = \frac{\mathbf{F}}{\mathbf{C} - \omega^2(\mathbf{M} + \mathbf{A}) - i\mathbf{B}} \quad (3.18)$$

where:

- **M**: Mass matrix (captures inertial effects),
- **A**: Added mass matrix (frequency-dependent, accounts for fluid-structure interaction),
- **B**: Damping matrix (accounts for energy dissipation),
- **C**: Stiffness matrix (restoring forces from hydrostatics),
- **F**: Wave and environmental forces acting on the structure,
- ω : Wave frequency.

The mass matrix M represents the rigid-body inertia of the floating barge. Since the structure is symmetric, the matrix is strictly diagonal, containing only the total mass and moments of inertia about the principal axes. There are no off-diagonal coupling terms, as mass does not directly link different motion modes. This matrix forms the foundation for dynamic analyses, defining how the barge responds to forces and accelerations.

$$M = \begin{bmatrix} m & 0 & 0 & 0 & 0 & 0 \\ 0 & m & 0 & 0 & 0 & 0 \\ 0 & 0 & m & 0 & 0 & 0 \\ 0 & 0 & 0 & I_x & 0 & 0 \\ 0 & 0 & 0 & 0 & I_y & 0 \\ 0 & 0 & 0 & 0 & 0 & I_z \end{bmatrix} \quad (3.19)$$

The stiffness matrix C represents the hydrostatic restoring forces acting on the floating barge due to buoyancy. Since the barge is free-floating and symmetric, there is no stiffness in surge, sway, or yaw, meaning the corresponding terms are zero. The diagonal terms describe the primary restoring effects, influenced by buoyancy and waterplane inertia.

$$C = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & C_{33} & C_{34} & C_{35} & 0 \\ 0 & 0 & C_{43} & C_{44} & 0 & 0 \\ 0 & 0 & C_{35} & 0 & C_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \quad (3.20)$$

When a floating body moves through water, it experiences an additional inertia known as added mass. This effect arises because the structure must displace and accelerate a portion of the surrounding fluid as it moves. As a result, the effective mass of the body increases beyond its physical weight. Added mass is influenced by several factors, including the shape of the object and the volume of water displaced. It is also dependent on frequency, meaning its impact changes based on the motion characteristics of the

floating platform. For offshore wind structures, understanding added mass is crucial for accurately predicting their dynamic behavior under wave and wind forces [23].

$$A = \begin{bmatrix} A_{11} & 0 & 0 & 0 & A_{15} & 0 \\ 0 & A_{22} & 0 & A_{24} & 0 & 0 \\ 0 & 0 & A_{33} & A_{34} & A_{35} & 0 \\ 0 & A_{24} & A_{43} & A_{44} & 0 & 0 \\ A_{15} & 0 & A_{53} & 0 & A_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & A_{66} \end{bmatrix} \quad (3.21)$$

Another significant hydrodynamic parameter is the radiation damping coefficient, which represents the energy loss due to wave radiation generated by a moving structure. When the platform oscillates, it disturbs the surrounding water, producing waves that carry energy away from the system. This outgoing wave energy reduces the motion amplitude of the floating body, acting as a natural damping force. Radiation damping plays a vital role in assessing wave-induced forces and stability, particularly in frequency domain analyses that evaluate interactions between waves and floating structures [23].

$$B = \begin{bmatrix} B_{11} & 0 & 0 & 0 & B_{15} & 0 \\ 0 & B_{22} & 0 & B_{24} & 0 & 0 \\ 0 & 0 & B_{33} & B_{34} & B_{35} & 0 \\ 0 & B_{42} & B_{34} & B_{44} & 0 & 0 \\ B_{51} & 0 & B_{53} & 0 & B_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & B_{66} \end{bmatrix} \quad (3.22)$$

Together, these coefficients added mass and radiation damping, help define the motion response and stability of offshore wind platforms.

Environmental forces are derived using wave-induced velocity potentials:

$$\phi(x, t) = \phi_0 + \phi_T + \sum_{m=1}^M \sum_{j=1}^6 \phi_{mj}(x) \quad (3.23)$$

where:

- ϕ_0 : Incident wave potential,
- ϕ_T : Diffracted wave potential,
- ϕ_{mj} : Radiation potentials caused by the body's oscillatory motions.

Wave forces are calculated as:

$$F_k = -\rho \int_S p \mathbf{n} dS \quad (3.24)$$

where p is the pressure, $\mathbf{n}$ is the surface normal, and S is the wetted surface.

The radiation and diffraction problems describe the interaction of waves with structures, satisfying boundary conditions such as [28]:

- No-flux condition on the body surface,
- Continuity at the free surface,
- Decay at infinity for outgoing wave disturbances.

The diffraction potential satisfies:

$$\phi_T(x) = -\phi_0(x) \quad (3.25)$$

These problems are solved using Green's functions applied to boundary integrals.

3.3.2 Wave Elevations and Motion Response

Wave elevations are determined as:

$$\zeta(\omega, x) = \zeta_0(\omega, x) + \sum_{i=1}^6 (\zeta_i(\omega, x) + \zeta_T(\omega, x)) \quad (3.26)$$

Relative motions between the structure and wave surface are defined as:

$$r(\omega, x) = \zeta(\omega, x) - z(\omega, x) \quad (3.27)$$

3.3.3 Mesh validation

Mesh validation is a critical step in ensuring that DIFFRAC can accurately and efficiently calculate the forces acting on the floating barge, thereby providing reliable results. According to DNV standards [6], the size of mesh panels should be approximately 1/6 of the maximum wave length that impacts the structure. In this study, the maximum selected frequency is 2.2 rad/s. Using this frequency and considering seabed of 50, 100 and 150 meters, the corresponding wave length are determined. the calculation of the wave lengths is performed using the dispersion equation:

$$\omega^2 = gk \tanh(kh) \quad (3.28)$$

where:

- ω : Angular frequency (rad/s)
- g : Acceleration due to gravity (9.81 m/s²)
- k : Wave number ($k = \frac{2\pi}{\lambda}$)
- h : Water depth (meters)
- λ : wave length (meters)

To investigate the relationship between angular frequency (ω), water depth (h), and wave length (λ), the dispersion equation $\omega^2 = gk \tanh(kh)$, where $k = \frac{2\pi}{\lambda}$, was utilized. Due to the implicit nature of this equation, numerical methods were employed to solve for λ across a range of ω values (0.1 to 2.2 rad/s) and water depths of 50, 100, 150, 200, 250, and 300 meters. Using Python, the `scipy.optimize.fsolve` function was implemented to iteratively determine the wave length for each combination of frequency and depth, starting with an initial guess of 10 meters. The results are shown in the following figure:

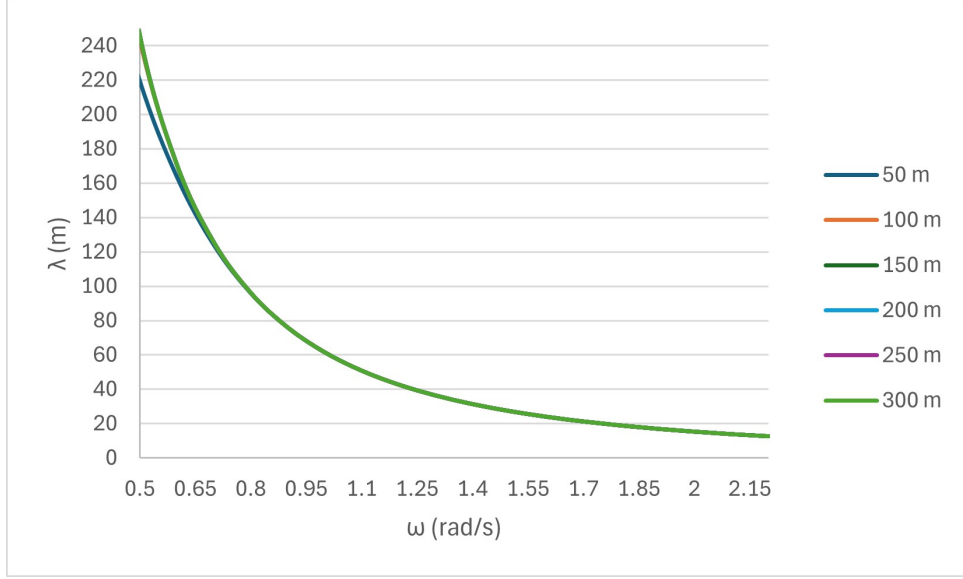


Figure 3.3: Wave Length in relation with wave frequencies and water depth

According to DNV standards [6], ensuring proper mesh refinement is essential for accurate simulations. For panel methods where the potential remains constant over each panel, several principles should apply:

- The length of the panel mesh should not exceed one-sixth $1/6$ of the smallest wave length being analyzed.
- Near the waterline, smaller panels are necessary to accurately compute wave-induced forces.
- The discretized model's water surface area and volume should closely match those of the actual structure.

As illustrated in the figure above, for wave frequencies greater than 0.7 rad/s , the wave length becomes independent of the water depth. This occurs because, at these frequencies, the influence of the seabed is negligible, indicating that the waves propagate under conditions of deep water. At a frequency of 2.2 rad/s , the smallest wave length is 12.73 m . Based on this value, the appropriate mesh size should be calculated as one-sixth of the wave length, following DNV standards. Thus, the recommended mesh panel length is:

$$\text{Maximum Panel mesh size} = \frac{12.73}{6} = 2.12 \text{ m} \quad (3.29)$$

For the scope of this thesis, the open-source design tool SALOME was utilized to create the geometry of the barge, as described in the previous section, and to generate its mesh. A panel size of 2 m was selected for the models. The illustrations of this model is presented below.

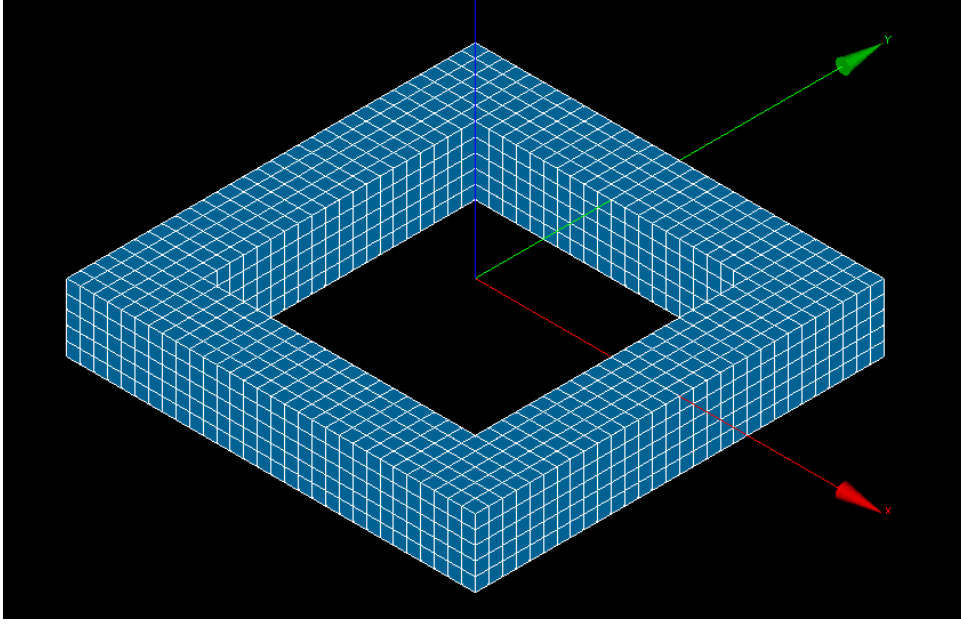


Figure 3.4: Meshed Barge with moonpool 40mX40m

Although the barge height is 15 m, the model is designed and meshed exclusively in the wetted area, corresponding to the draft depth of 10 m. All normal vectors are oriented against the flow, with particular attention to their placement inside the moonpool to effectively capture piston and sloshing modes.

3.3.4 Viscous Damping

According to DNV standards, motion damping in large-volume structures primarily arises from wave radiation damping, hull skin friction, hull eddy formation, and viscous effects such as those generated by bilge keels, risers, and mooring systems. Wave radiation damping is determined using potential theory, while viscous damping is typically estimated using simplified hydrodynamic models or experimental data. In certain cases, Computational Fluid Dynamics (CFD) can be employed to evaluate viscous damping for simple geometries. However, for this study, focusing on a barge-type floater with a moonpool in the context of a Floating Offshore Wind Turbine (FOWT), viscous damping is excluded from the analysis due to the absence of experimental data or CFD calculations. Instead, the study utilizes the potential theory-based numerical tool DIFFRAC to assess wave radiation damping.

3.3.5 Damping lid factor

According to Pauw et al., the damping lid method introduces an artificial damping zone, called a “lid”, at the free surface within the computational domain. This lid is designed to reduce the amplitude of unrealistic standing waves caused by piston and sloshing modes which are represented inside moonpools or in the gap between ships and decks. A table of free boundary equations presented below:

Table 3.3: Comparison of conventional, rigid lid, and damping lid methods.[36]

Conventional	Rigid lid	Damping lid
-	Rigid	Semi permeable
Undamped wave elevation	No wave elevation	Damped/tuned wave elevation
$\frac{\partial \phi}{\partial z} - \frac{\omega^2}{g}\phi = 0$	$\frac{\partial \phi}{\partial n} = 0$	$\frac{\partial \phi}{\partial z}(1 - \varepsilon) - \frac{\omega^2}{g}\phi = 0$

This equation can be viewed as a boundary condition or a governing equation for fluid flow under wave action in a vertically stratified system. The first term,

$$\frac{\partial \phi}{\partial z}(1 - \varepsilon),$$

denotes the vertical gradient of the velocity potential adjusted by the factor $(1 - \varepsilon)$. This adjustment indicates that the gradient is influenced by a parameter, depending on the context of ε .

Parameter ε is the non-dimensional parameter related to the damping parameter. This parameter will take values from 0 (incorporating zero damping) to 0.1 to assess the relationship and affection to the moonpool resonances and the mooring loads and motions [36].

In the figure below, the damping area appears to be placed precisely at the sea level for the barge.

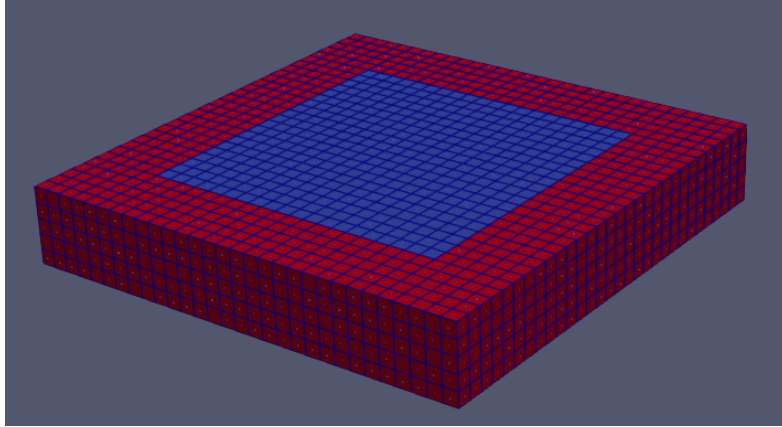


Figure 3.5: Damping lid area barge 40X40

At this point, by applying different values on the parameter ε , and running the model in DIFFRAC, new RAOs will be generated. These values will be used to create a graph showing the relationship and impact of the damping factor with resonance effects.

For the scope of this thesis, the damping lid factor, denoted as ε , was varied from 0 to 0.1 to examine its influence on the structural characteristics across a range of wave frequencies. The results will be presented and discussed in the Chapter 4.

3.4 aNySIM modeling

For this study, the aNySIM numerical tool of MARIN will be used in order to account for mooring system and wind turbine configuration. Tool aNySIM XMF is time-domain simulation software designed for the maritime industry. It is used to simulate the motions of both stationary offshore vessels and sailing ships, providing valuable insights for maritime contractors, shipyards and engineering consultants. The software helps in predicting the behavior of vessels under various conditions, which is crucial for planning and safety. By using aNySIM, professionals can optimize vessel or offshore structures and operations, ensuring efficiency and safety in maritime projects [29]. In this section, a detailed explanation is provided for each step carried out using aNySIM, including the assumptions made, theoretical basis applied, and limitations considered. For the scope of this thesis, a barge-type platform with a moonpool measuring 40 m $\times$ 40 m has been selected. Both the characteristics of the mooring system, the wind turbine and environmental conditions are analytically described to ensure accurate and focused answers to the primary research questions.

3.4.1 Mathematical model

The mathematical model in aNySIM is based on a time-step solution of the system of coupled differential equations of motion. The equations of motion for a floating structure can be expressed as:

$$(M + A)\ddot{x}(t) + B\dot{x}(t) + Cx(t) = F(t) \quad (3.30)$$

where:

- M is the mass matrix,
- A is the added mass matrix,
- B is the damping matrix,
- C is the stiffness matrix,
- $x(t)$ is the displacement vector, and
- $F(t)$ is the external force vector.

For the scope of this thesis, external forces account for first and second order hydrodynamic forces, mooring forces and wind forces.

3.4.2 Mooring configuration

Following the methodology, after the floater type has defined and hydrodynamic scheme has calculated from tool DIFFRAC, it is time to add the mooring configuration in the system. Two common types of mooring systems are used in offshore structures, and thus, many combinations can be produced. For example, a catenary system with a chain setup, a catenary system with a chain-wire-chain setup, etc.

A catenary mooring system is a system with a part lying on the seabed to tackle horizontal forces. The main characteristic that allows it to cope with the system's dynamic motions

is its self-weight. It is widely used, especially in shallow and medium depth water [30]. A configuration is shown.

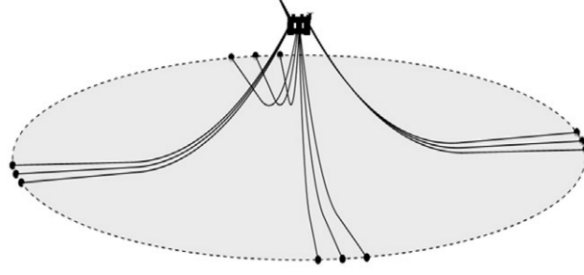


Figure 3.6: Catenary mooring system [30]

As opposes to catenary moorings, taut line moorings do not lie on the seabed but remain taut for their entire length, from the anchor to the fairlead. Some benefits of this system are that the anchor footprint is smaller than that of the catenary system and uses less material. It is suitable for deep- and ultra-deep-sea states.

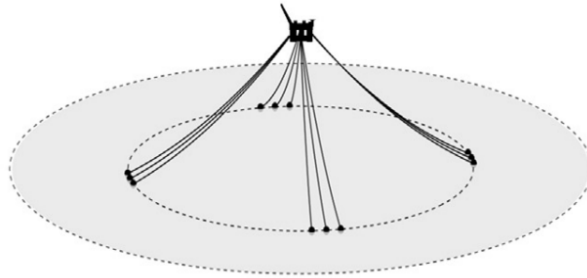


Figure 3.7: Taut line mooring system [30]

Static Catenary Equation

The catenary profile is expressed as [39]:

$$L_T = l_{\text{tot}} - \frac{T_0}{w} \sinh \left(\text{arcosh} \left(\frac{h_s w}{T_0} + 1 \right) \right) + \frac{T_0}{w} \text{arcosh} \left(\frac{h_s w}{T_0} + 1 \right). \quad (3.31)$$

where T_0 is the horizontal tension, w is the wet weight per unit length, h_s is the sea depth, and l_{tot} is the total length of the mooring line, as illustrated in the following figure.

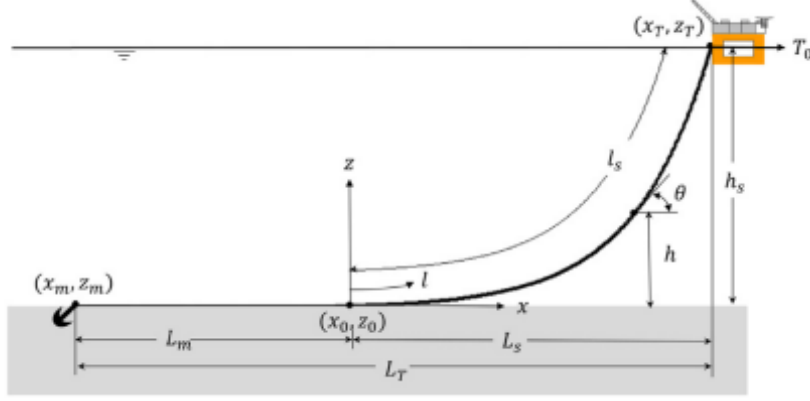


Figure 3.8: Catenary line illustration [30]

Catenary Mooring System Analysis

The time-domain simulation tool, *aNySIM*, employs two distinct methods to compute and model catenary mooring lines. The first method, known as the composite catenary line approach, characterizes the shape and tension distribution along the suspended chain. This method assumes the chain is in static equilibrium, subject to a uniform force field, with the option to incorporate uniform drag effects while neglecting bending stiffness. [29]

The approach involves constructing the mooring line shape in discrete segments, where the position and forces at the endpoint of each segment are computed based on the starting point's location and initial tension. Each segment's endpoint serves as the starting point for the subsequent segment, creating a chain of calculated interactions throughout the length of the mooring line. Submerged parts of the chain are accounted for separately from the non-submerged parts. [29]

To model seabed interactions, the tool implements a root-finding process. When a segment of the line becomes parallel to the seabed, it is assumed to represent the part lying along the bottom surface.

The second approach to constructing the mooring line in the *aNySIM* tool is the lumped mass line method, which incorporates inertia effects on the line loads. This method assumes that load elongation follows a linear relationship, hysteresis effects are absent, and bending stiffness is linear.

It is a dynamic modeling technique based on the lumped mass discretization of the line shape. The model accounts for axial restoring forces, axial damping, weight and buoyancy, Morison loads, and seabed interactions. As outlined, the mooring line is divided into discrete elements, with the mass allocated to nodes located between these elements. The first and last nodes are fixed to the body and seabed, respectively. Each node is connected to adjacent nodes by spring stiffness and optional damping, as illustrated in the figure below. [29]

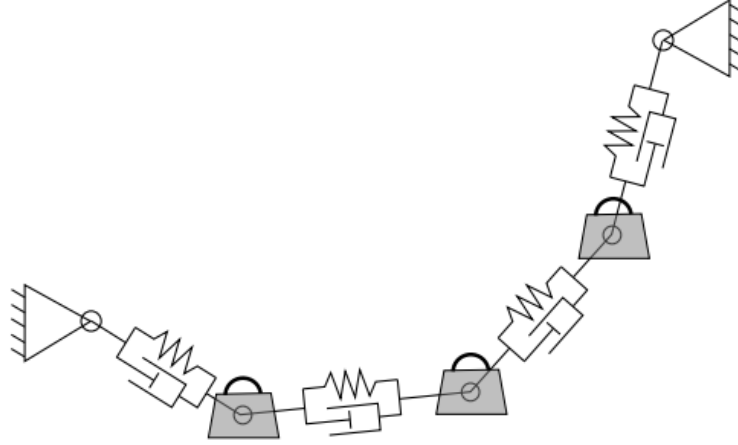


Figure 3.9: Lumped mass method. [29]

For the scope of the second research question, and due to computational considerations, the model used in this thesis is based on the composite line approach.

To account for the influence of the damping lid factor on mooring forces and motions of the coupled system in the time-domain analysis, it is assumed that the seabed is located at a depth of 50 meters.

3.4.3 Wind Turbine

The **NREL 5 MW reference wind turbine** is widely recognized as a benchmark model for offshore wind energy research. Developed by the National Renewable Energy Laboratory (NREL), this turbine serves as a standard for evaluating aerodynamic, hydrodynamic, and structural interactions in offshore wind systems. [22]

The key design parameters of the NREL 5 MW wind turbine are as follows [22]:

- **Rated Power:** 5 MW
- **Rotor Diameter:** 126 m
- **Hub Height:** 90 m
- **Cut-in Wind Speed:** 3 m/s
- **Rated Wind Speed:** 11.4 m/s
- **Cut-out Wind Speed:** 25 m/s
- **Blade Configuration:** Three-bladed, pitch-regulated

The NREL 5 MW wind turbine employs a variable-speed, pitch-controlled rotor to optimize power generation efficiency. The three-bladed design enhances stability and minimizes turbulence effects. Additionally, its aerodynamic properties are tailored to withstand offshore wind conditions, ensuring reliable energy production in dynamic environments. [22]

The turbine is frequently used in floating offshore wind platform studies due to its well-documented performance characteristics. When integrated into floating structures, such as barge-type platforms, the interaction between wind loads and platform dynamics becomes crucial. The design parameters of the turbine influence the overall stability and motion response of the floating system, making it an essential component in offshore wind energy research. [22]

The wind turbine tower is modeled as a rigid body with two joints: one at the base and one at the top. The top joint connects to the nacelle, which in turn is attached to the three blades. Each component—the tower, nacelle, and blades—is considered an independent body, characterized by its own mass and moment of inertia.

3.4.4 Environmental Conditions

At this section the environmental conditions will be described. The parameters to describe the environment are wind, waves, and currents. For the purpose of this research, only wind and waves will be considered.

Wind

In the aNySIM time-domain simulation tool, the wind model can represent either a constant wind or a gusting wind scenario originating from a single direction. The wind is described as a horizontal velocity vector, assuming spatial homogeneity and a constant height throughout the simulation domain.

More specifically, the wind velocity vector defines both the wind speed and direction. It can be implemented as:

- A **constant wind**, where the velocity remains unchanged over time.
- A **wind spectrum**, capturing statistical variations in wind characteristics.
- A **wind time trace**, incorporating temporal changes in both speed and direction.

For this study, the wind is modeled as a horizontal velocity vector and assumed to remain constant throughout the analysis.

Waves

The simplest way to describe a wave is the Airy regular wave model. The wave elevation can be described from the following equation: [2]

$$\zeta = \zeta_a \cos(kx - \omega t) \quad (3.32)$$

and the wave potential:

$$\Phi_w(x, z, t) = \zeta_a \frac{g}{\omega} \frac{\cosh(k(h + z))}{\cosh(kh)} \sin(kx - \omega t). \quad (3.33)$$

where,

- ω : Angular frequency (rad/s)
- g : Acceleration due to gravity (9.81 m/s^2)
- ζ_a : Wave amplitude
- k : Wave number ($k = \frac{2\pi}{\lambda}$)
- h : Water depth (meters)
- λ : wave length (meters)

To accurately represent real sea states, spectral models such as the JONSWAP spectrum can be utilized. The aNySIM tool is capable of incorporating these types of spectra, enabling realistic wave condition modeling.

In particular, the JONSWAP spectrum is one of the most commonly used spectral representations in offshore operations. It is derived from empirical studies and accounts for the irregular nature of ocean waves by summing multiple regular wave components. This approach allows for a more realistic description of wave behavior, improving the accuracy of simulations and analyses related to offshore engineering [29].

The shape of the wave spectrum is defined as follows:

$$S(\omega) = \omega^{-5} e^{-1.25 \left(\frac{\omega_p}{\omega} \right)^4} \gamma^r, \quad (3.34)$$

$$r = e^{-\frac{(\omega - \omega_p)^2}{2\sigma^2 \omega_p^2}}, \quad (3.35)$$

$$\sigma = \begin{cases} \sigma_1, & \omega < \omega_p \\ \sigma_2, & \omega > \omega_p \end{cases} \quad (3.36)$$

Chapter 4

Model Observations

This chapter presents the outcomes of the numerical analysis performed using the potential theory-based tool DIFFRAC on the barge-type floater with a moonpool Floating Offshore Wind Turbine (FOWT). The findings are structured to demonstrate the influence of various wave frequencies and the selected damping lid factor, ε , on the dynamic behavior and structural characteristics of the floater. Key observations and trends are discussed in detail, offering insight into the effectiveness of the applied methodology and the implications for the floater's performance under operational conditions.

As discussed in the previous chapters, the damping lid factor was introduced to correct the over-prediction of the wave elevation inside the moonpool of the barge. This issue arises from the absence of viscous damping calculations in the potential theory diffraction tool *DIFFRAC*. At this point, it is crucial to further investigate how the damping lid factor affects the hydrodynamic coefficients and the motions of the floater.

4.1 Hydrodynamics of barge Moonpool of $40\text{ m} \times 40\text{ m}$

In this section, the effects of the damping lid factor, ε , on the six degrees of freedom (DOFs) of the floater, as well as on the hydrodynamic coefficients, are analyzed. The values of the damping lid factor, ε , considered in this thesis are 0, 0.01, 0.02, 0.025, 0.03, 0.04, 0.06, 0.07 and 0.1.

4.1.1 Added mass coefficients

In the figures below, the added mass coefficients for the surge, sway, heave, roll, pitch, and yaw motions of the barge with a moonpool of these dimensions are illustrated. For every mode, except for the heave, an unexpected high peak is observed at the frequencies of 1.25 rad/s and 2.17 rad/s, which align closely with the expected first and third sloshing modes as described by Molin (2018) [34]. According to Molin's theory, the resonance frequencies for the sloshing modes are anticipated to occur at 1.25 rad/s and 2.15 rad/s, respectively.

The first six sloshing mode frequencies are clearly discernible. Notably, the first (0.93 rad/s), third (1.52 rad/s), and fifth (1.96 rad/s) modes demonstrate a more pronounced impact. On the contrary, the second (1.25 rad/s), fourth (1.75 rad/s), and sixth (2.15

rad/s) modes exhibit a less significant influence in surge and sway. These frequencies align closely with Molin's theoretical calculations, indicating strong convergence.

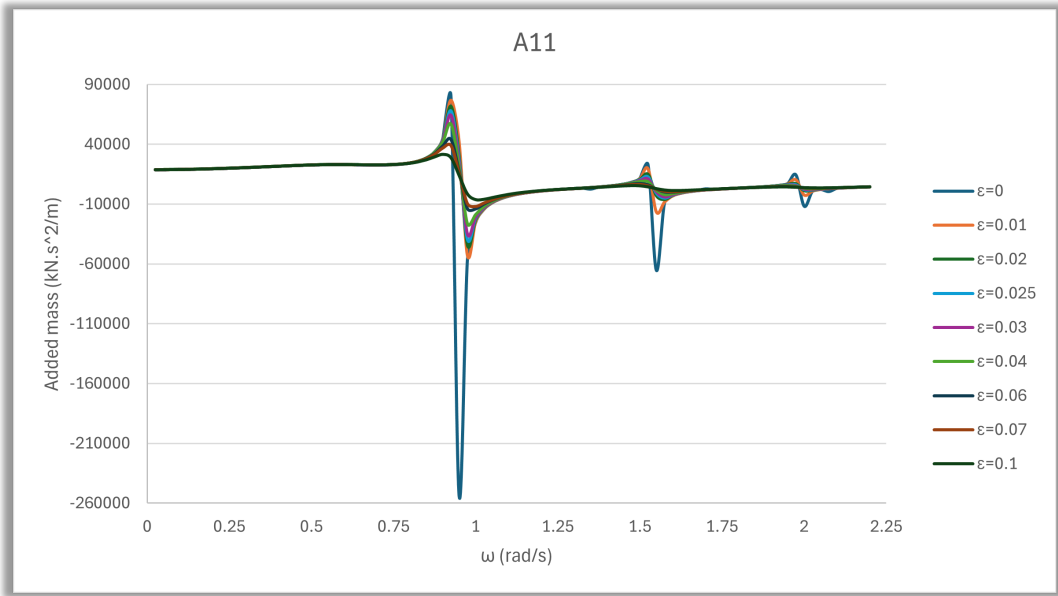


Figure 4.1: Added mass coefficient in surge

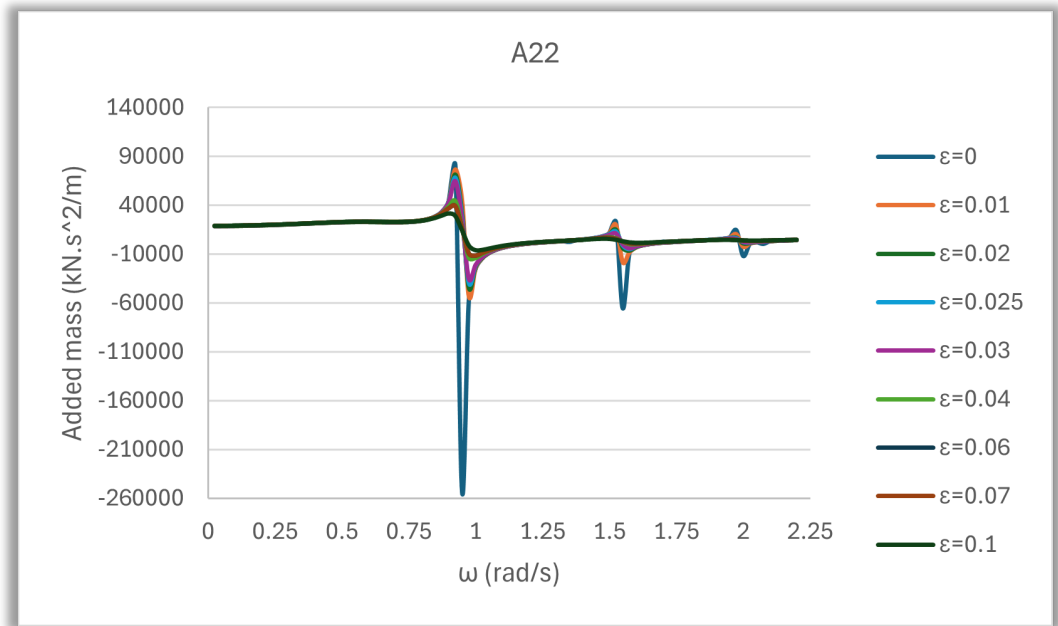


Figure 4.2: Added mass coefficient in sway

In this type of barge, with a moonpool measuring $40\text{ m} \times 40\text{ m}$, it is observed that in heave mode, the added mass coefficients are influenced not only by the piston mode at 0.67 rad/s but also by the second sloshing mode at 1.25 rad/s . This phenomenon arises due to the substantial dimensions of the moonpool, which provide ample space for wave reflections to occur. These reflections lead to the development of standing waves that, in

turn, amplify the heave motion. Added mass coefficient RAO is represented in the figure below. It is notably again that the frequencies of resonance align perfectly with Molin's formula predictions.

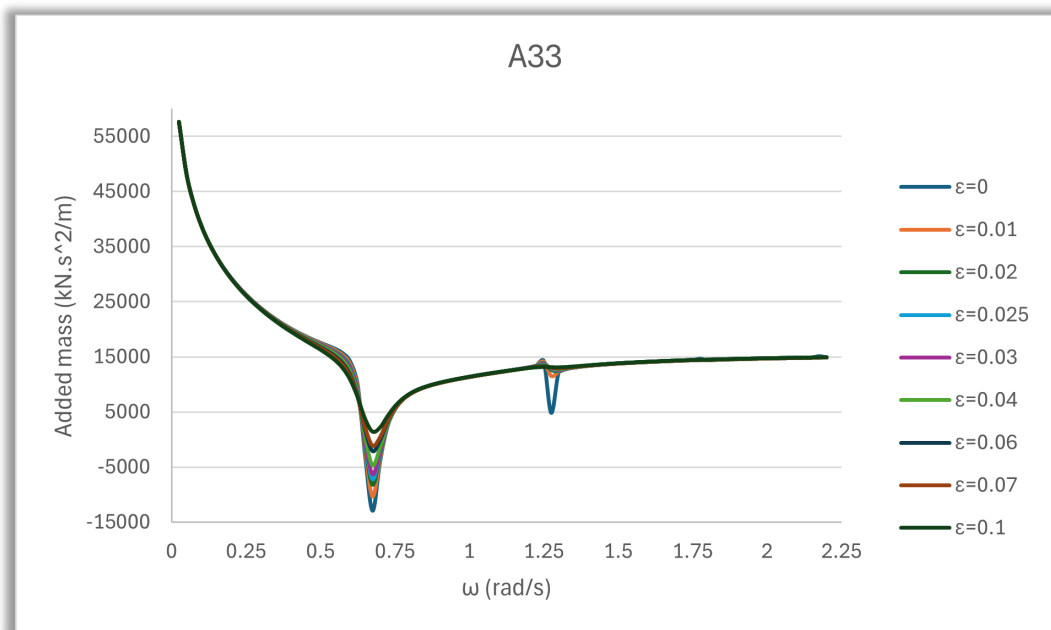


Figure 4.3: Added mass coefficient in heave

In the roll and pitch modes, it is observed that the first sloshing mode is dominant at a frequency of 0.93 rad/s. However, this peak appears to be unrealistic.

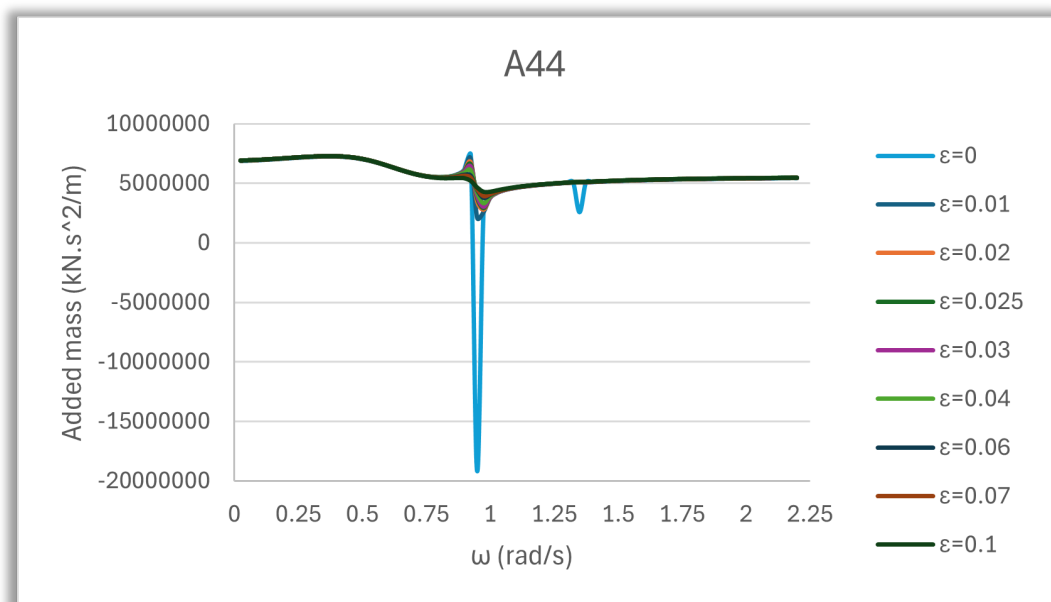


Figure 4.4: Added mass coefficient in roll

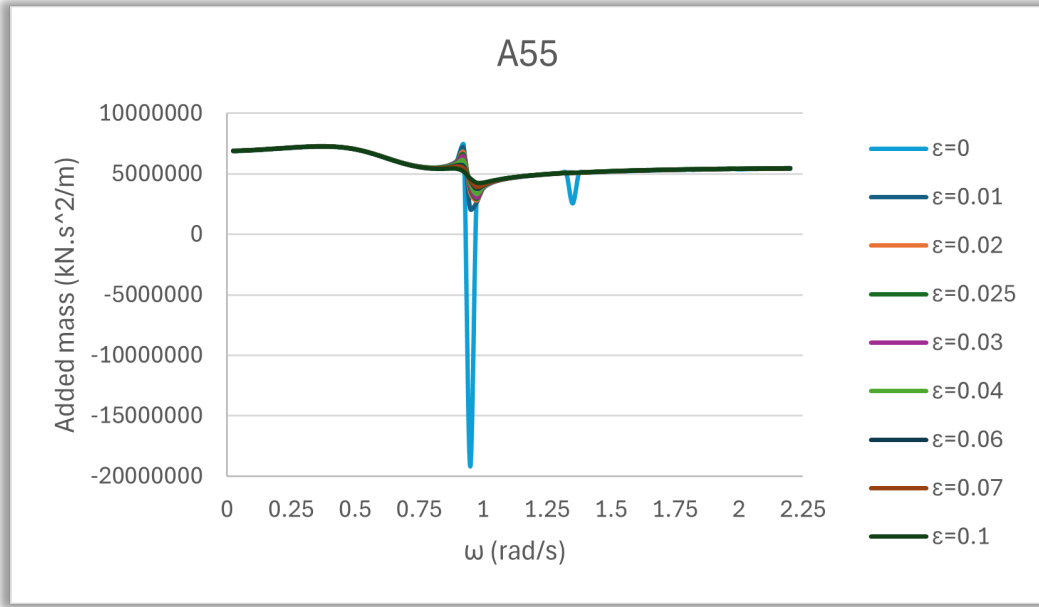


Figure 4.5: Added mass coefficient in pitch

In the figure below, the yaw mode is presented, in which the third mode and sixth mode of sloshing resonate at 1.53 rad/s and 2.15 rad/s respectively.

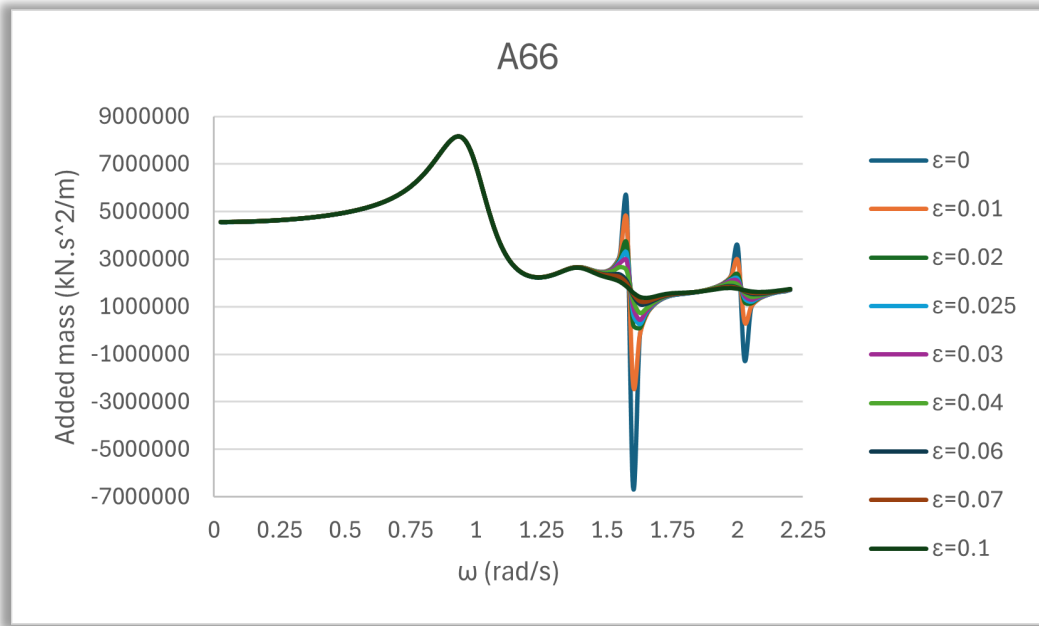


Figure 4.6: Added mass coefficient in pitch

A detailed evaluation of the graphs reveals that the damping lid method effectively reduces unexpected and unrealistic peaks in added-mass RAOs while maintaining consistent behavior across other frequencies.

It is essential to highlight that negative values of added mass are observed in this case. Extensive research has been conducted on this phenomenon, and as noted by P. McIver

and D. V. Evans [32], when the depth of submergence is small compared to the free surface area, the effects of the free surface become significant. In addition, due to the symmetric geometry of the barge, surge and sway, as well as pitch and roll, exhibit similar responses in all wave directions, as illustrated in the figures below.

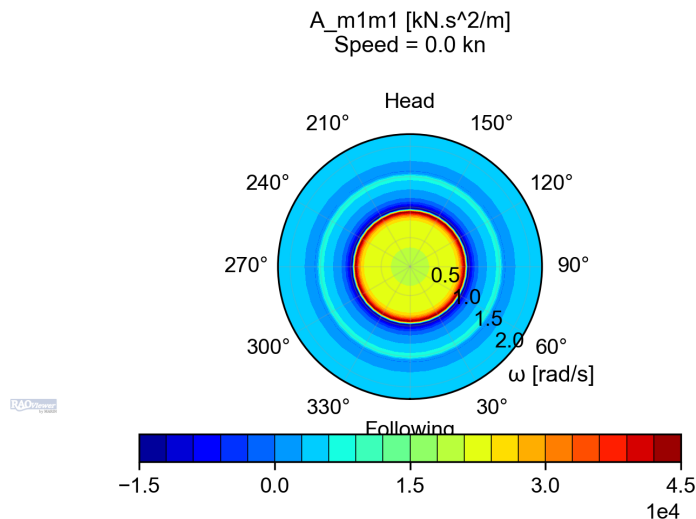


Figure 4.7: Added mass coefficient in surge for all directions.

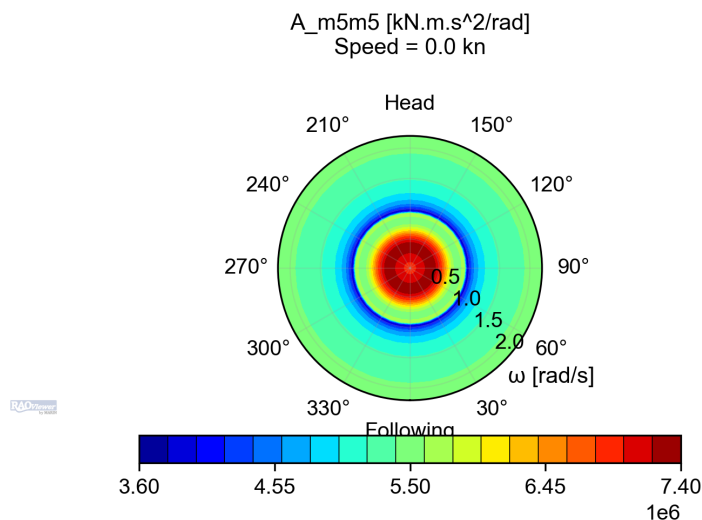


Figure 4.8: Added mass coefficient in pitch for all directions.

4.1.2 Radiation damping coefficients

Radiation damping behaves similarly to added mass coefficients. It is evident that in surge and sway modes, radiation damping is primarily dominated by the first sloshing mode at 0.937 rad/s and in less grade by the third and sixth mode at 1.53 rad/s and 2.15 rad/s respectively.

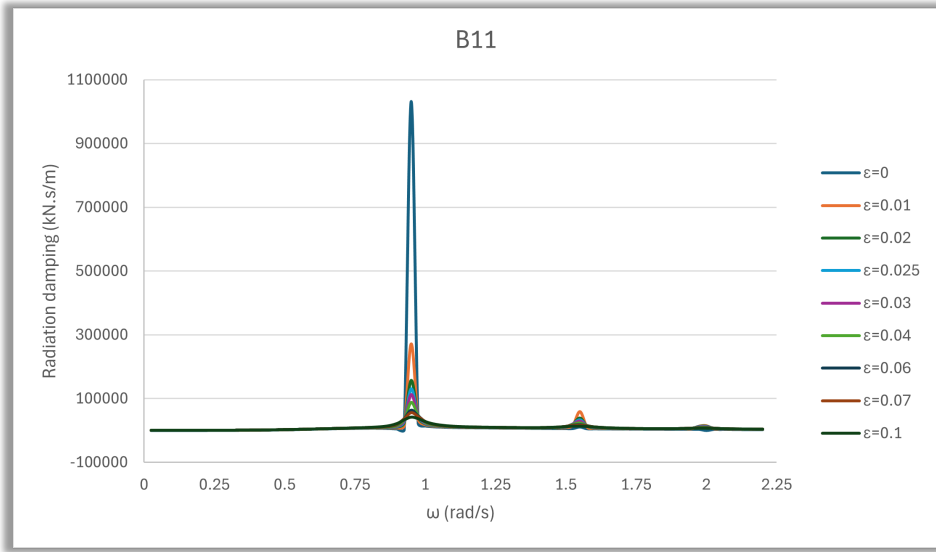


Figure 4.9: Radiational damping coefficient in surge.

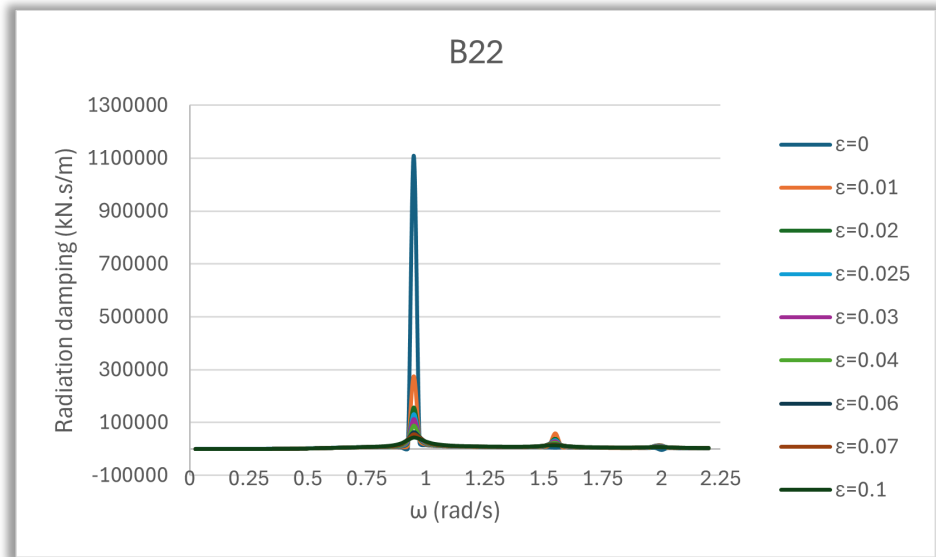


Figure 4.10: Radiational damping coefficient in sway.

In heave mode, activation occurs through both the piston mode at 0.67 rad/s and the second sloshing mode at 1.25 rad/s.

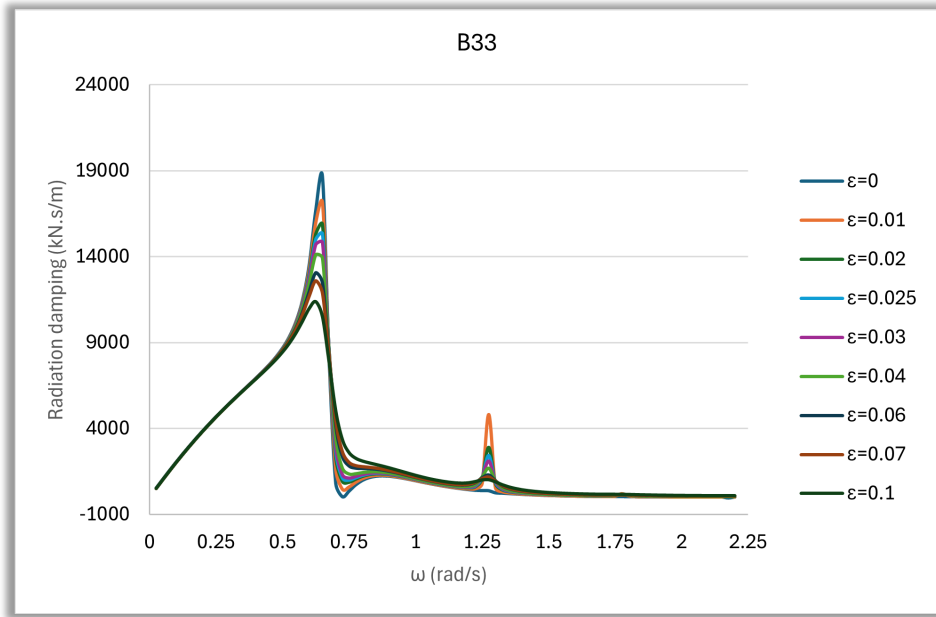


Figure 4.11: Radiational damping coefficient in heave.

Finally, roll and pitch modes are distinctly activated by the first sloshing mode at 0.937 rad/s.

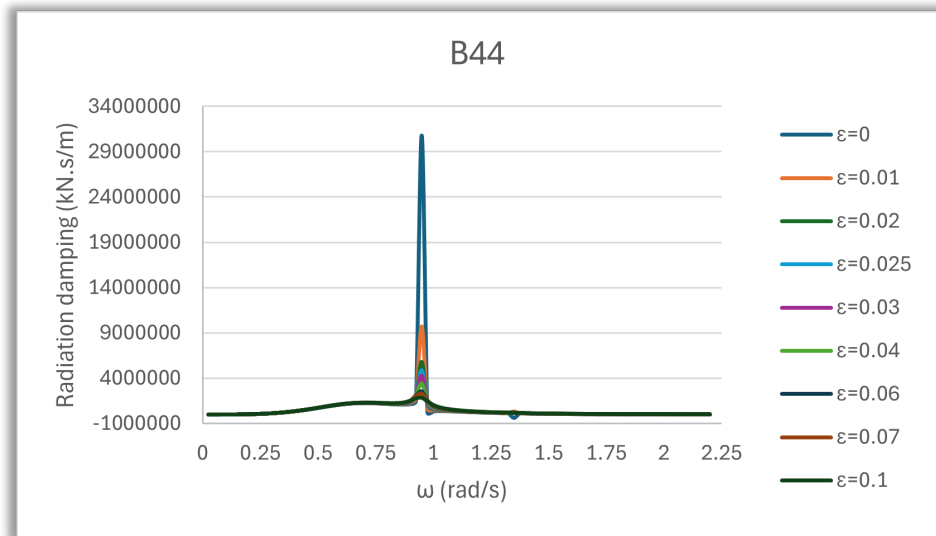


Figure 4.12: Radiational damping coefficient in roll.

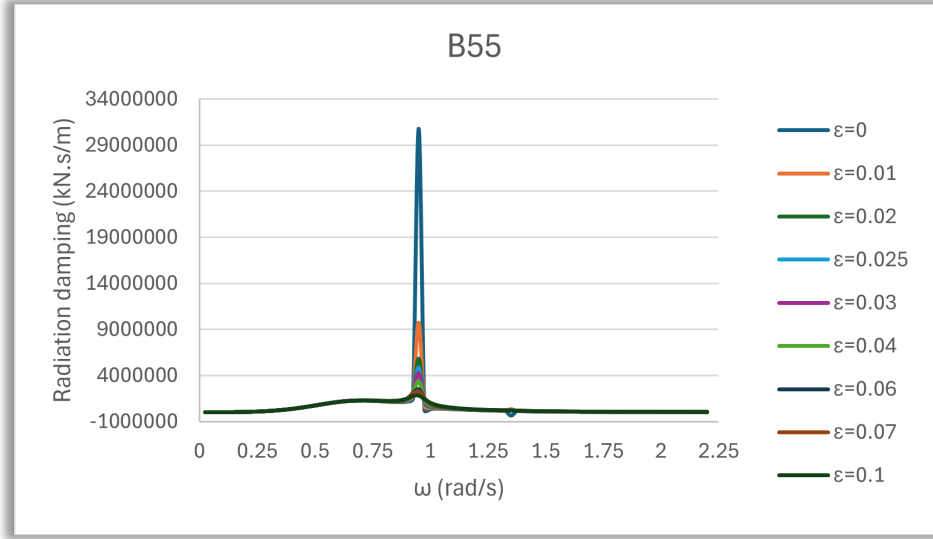


Figure 4.13: Radiational damping coefficient in pitch.

Same as surge and sway modes, but also similar to added mass coefficients, yaw mode radiational damping coefficient present peaks at third and sixth sloshing mode at 1.52 rad/s and 2.15 rad/s respectively.

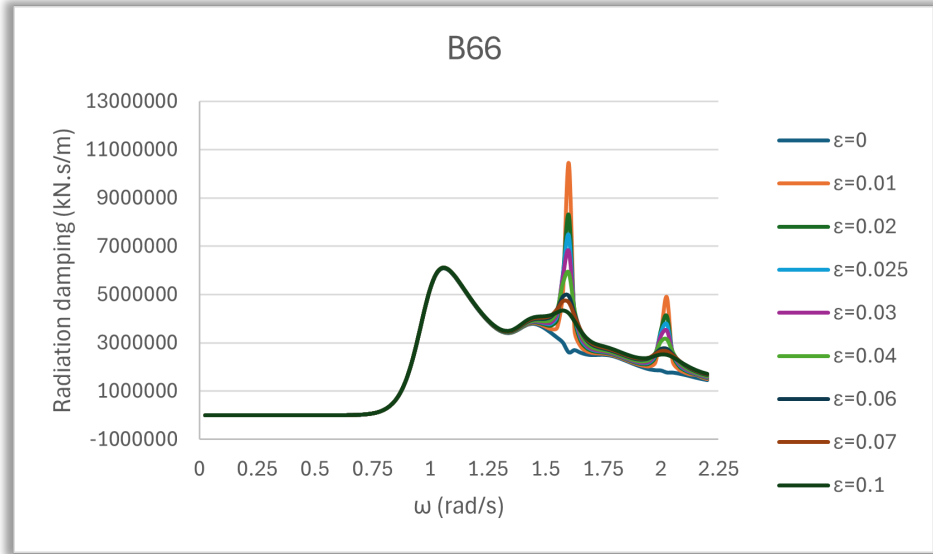


Figure 4.14: Radiational damping coefficient in yaw.

For both coefficients, it is observed that the third mode of sloshing is damped by applying a shorter damping lid factor compared to the first mode resonance. Therefore, further investigation is required to determine the optimal selection of the lid factor value.

4.1.3 Motion RAOs

In this section the surge, sway, heave, roll, pitch and yaw modes in motion RAOs of the barge are presented.

In surge mode, the presence of the moonpool influences the motions at the first and third sloshing mode frequencies. The first sloshing mode appears to shift slightly towards higher frequencies, from 0.93 rad/s to 1.05 rad/s. This variation arises due to differences in calculation methods between potential theory and Molin's assumption in his formulation. Molin assumes that the barge is fixed relative to an Earth-based reference frame, whereas diffraction analysis considers the barge as freely floating in water. Consequently, the interaction between the water inside the moonpool and the barge's motion induces a coupled effect, leading to this frequency shift.

The shift in resonance frequencies observed in diffraction analysis compared to radiation analysis can be explained by the fluid-structure interaction and the dynamic pressure redistribution around the moonpool [42]. In radiation analysis, the barge is assumed to be fixed, meaning the added mass of the moonpool fluid remains constant, leading to resonance frequencies close to Molin's formulation [33]. However, diffraction analysis accounts for the motion of the floating body and hydrodynamic coupling effects, which modify the effective inertia of the water column inside the moonpool.

As the barge oscillates, the moonpool fluid experiences varying inertial forces, altering the natural frequencies of both the piston and the sloshing modes [35]. Studies have shown that free-floating conditions significantly shift resonance frequencies due to increased added mass effects, as observed in experiments by Han et al. (2023) [19]. Huang et al. (2019) extended Molin's method to account for the dynamic influence of the barge, demonstrating that wave diffraction alters pressure distribution and resonance characteristics [20]. Overall, these results confirm that wave-structure interactions fundamentally reshape the hydrodynamic behavior of offshore platforms with moonpools.

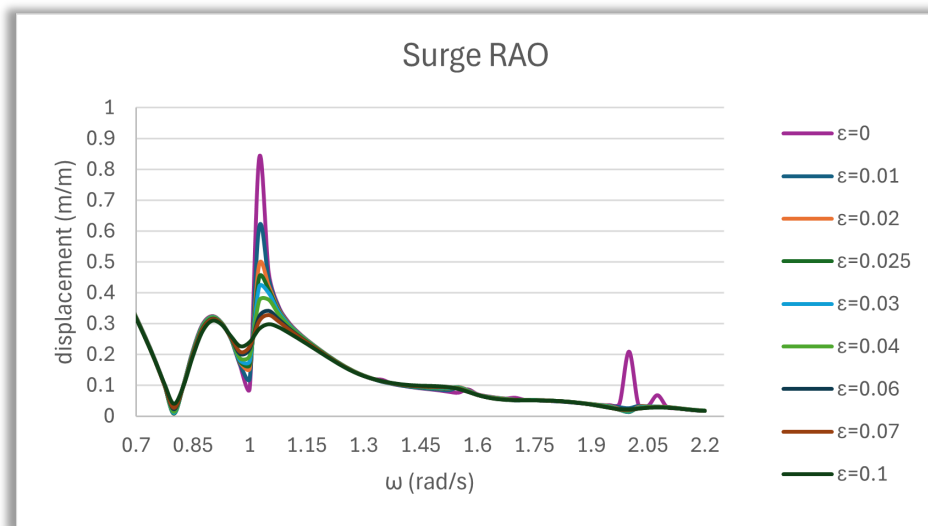


Figure 4.15: Surge RAO.

The graph below illustrates the sway motion. It is observed that the sway motion is influenced by several modes of piston and sloshing, yet its magnitude remains close to zero. As a result, it is considered negligible.

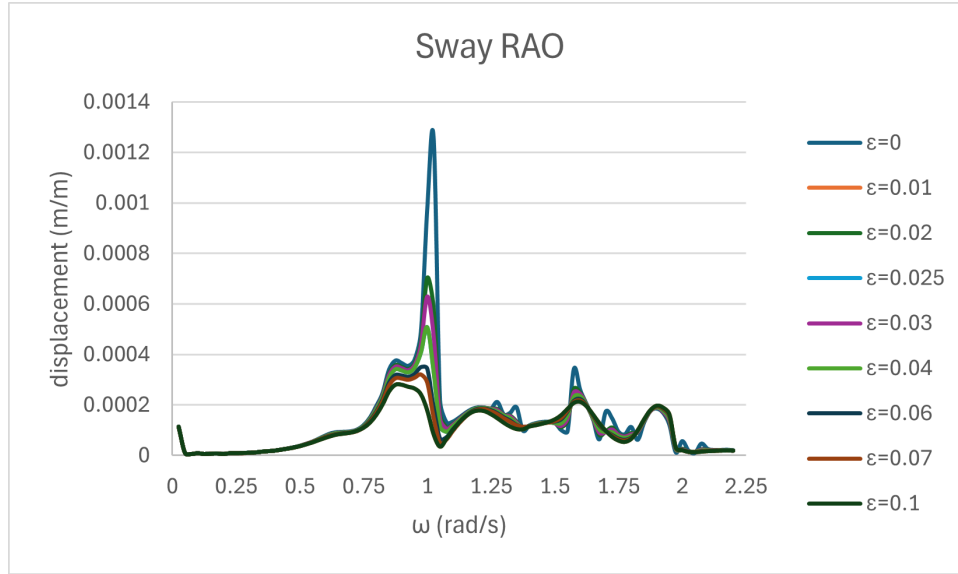


Figure 4.16: Sway RAO.

In the graph below, a resonance due to the piston mode of the moonpool is observed at a frequency close to the piston mode resonance frequency. Specifically, it occurs at 0.825 rad/s, although the calculated frequency is 0.67 rad/s. This difference arises because Molin's formula assumes the barge is constrained on the water, allowing only the water to move through it. However, in this case, the motions were derived under the assumption of a free-floating structure. Furthermore, it is evident that the damping lid factor ε effectively reduces the unexpected motions of the barge at the piston mode resonance frequency.

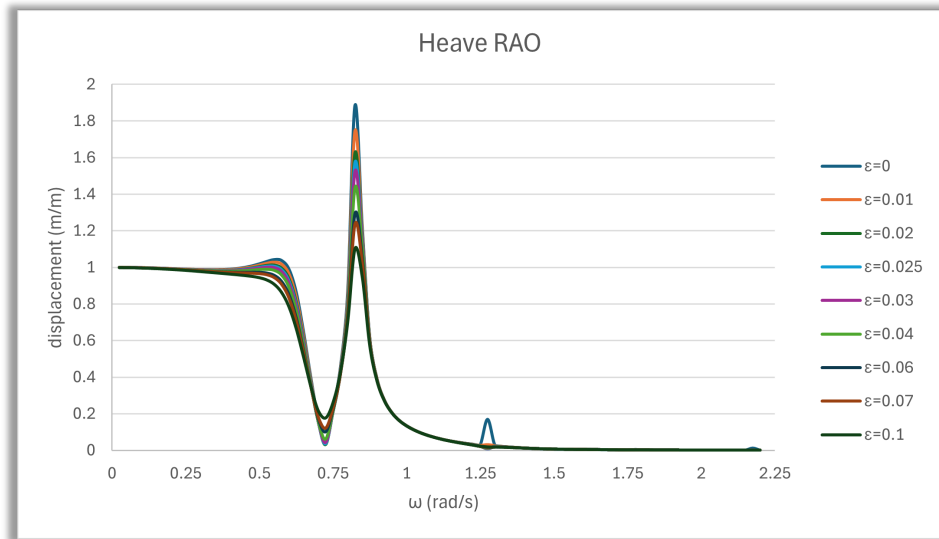


Figure 4.17: Heave RAO.

In roll motion, it is observed that the first, second, fourth, fifth, and sixth sloshing modes influence the motion. However, similar to the sway mode, the displacement magnitudes are extremely low and approach zero; therefore, they are considered negligible.

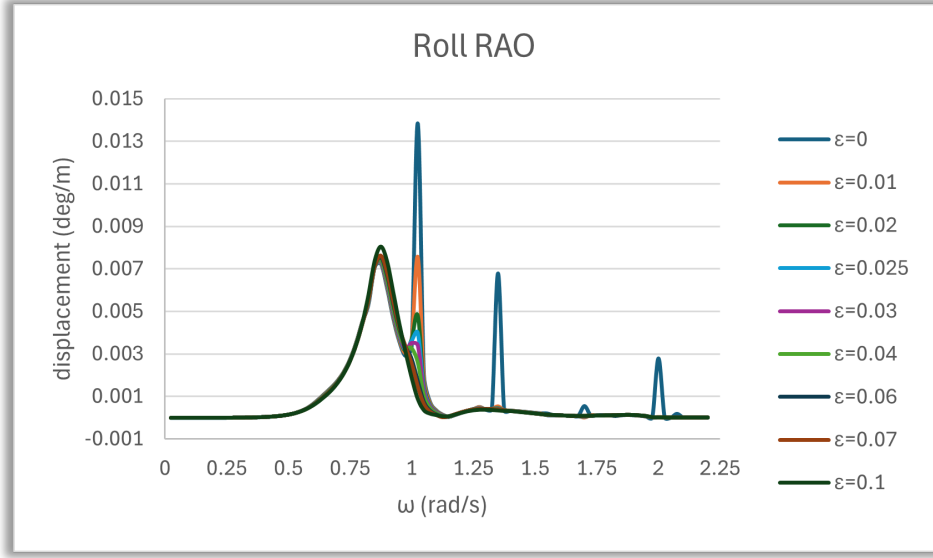


Figure 4.18: Roll RAO.

The pitch mode appears to be unaffected by the presence of the moonpool, as shown in the pitch RAO below. Similar to the surge mode, it also exhibits excitation close to the first sloshing mode frequency 1.25 rad/s. However, compared to the behavior of the entire system, this effect is negligible.

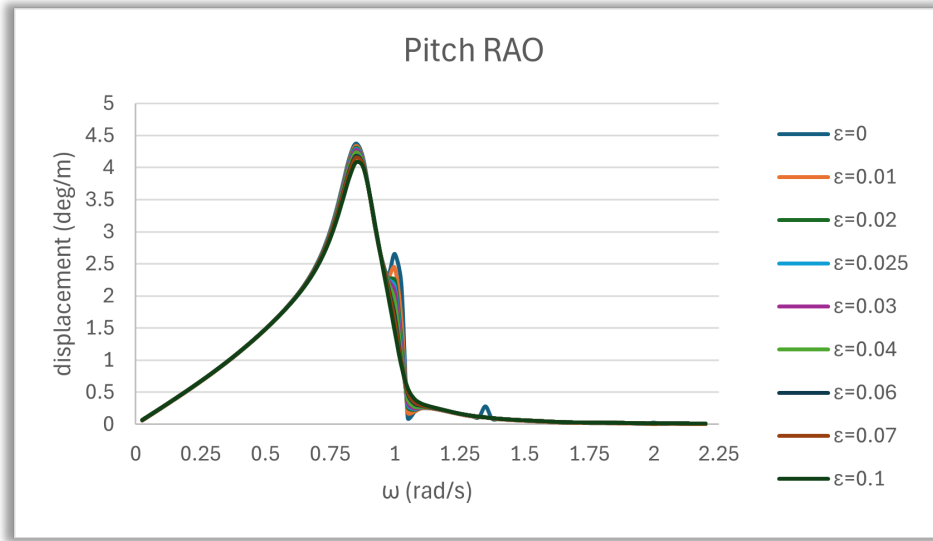


Figure 4.19: Pitch RAO.

4.1.4 Wave surface elevation

Going back to theory, water inside the moonpool can move in two primary directions, categorized into two distinct types of motion. The first involves up-and-down movement, similar to a water column, which is referred to as piston mode. The second entails forward-and-backward motion, known as sloshing mode. In these two oscillation modes, standing waves may develop, leading to significant elevations at the free surface. In this section, the free surface elevations will be calculated.

Easily is noted the piston mode resonance at 0.825 rad/s, the first sloshing mode at 1.05 rad/s and all the other sloshing modes till sixth at 2.15 rad/s. As illustrated in the figure below, the damping lid effectively reduces the wave amplitude, and for higher values of the damping lid factor, it can diminish the amplitude entirely.

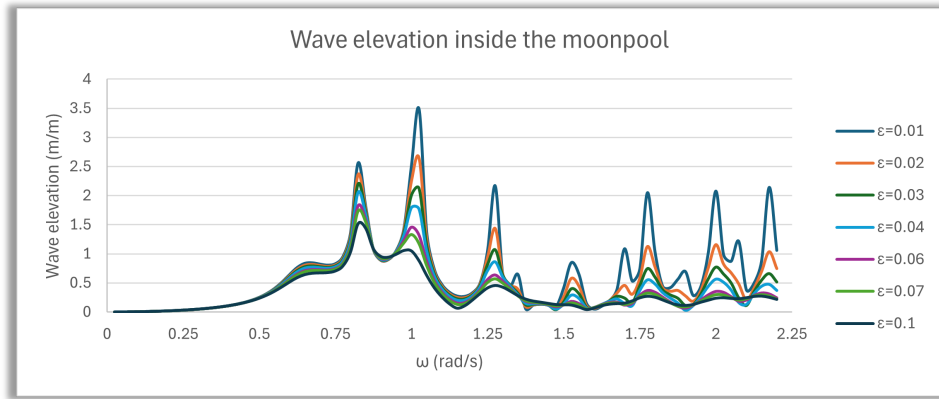


Figure 4.20: Wave elevation inside the moonpool.

In order to understand better the behavior of the damping lid to the wave elevation inside the moonpool two graphs are constructed to show the relation between the relative wave elevation and the damping lid factor ε .

The first graph illustrates the relationship in piston mode, where a nearly constant slope is observed. Consequently, determining a precise value for ε proves challenging. As expected, an increase in ε results in a consistent decrease in wave elevation.

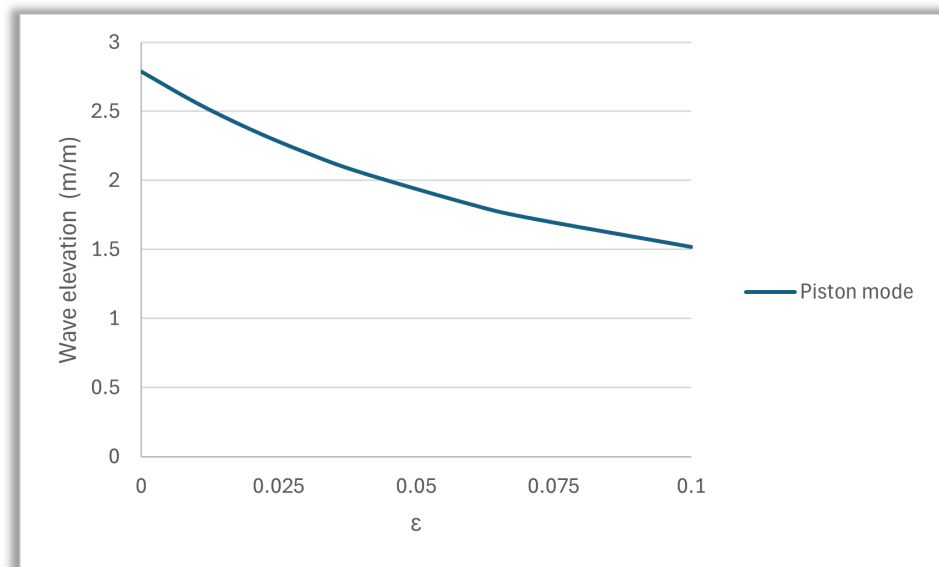


Figure 4.21: Wave elevation - ε relation at $\omega = 0,825$ rad/s.

In sloshing mode, similar to piston mode, the damping factor ε reduces the free surface elevation at the corresponding sloshing mode frequencies. The graph below illustrates the wave elevation in the moonpool under sloshing mode excitation. It is observed that beyond a damping lid factor of 0.030, the slope of the graph becomes more smooth.

However, at lower values of ε , the slope remains steep, indicating a strong influence of damping on wave elevation.

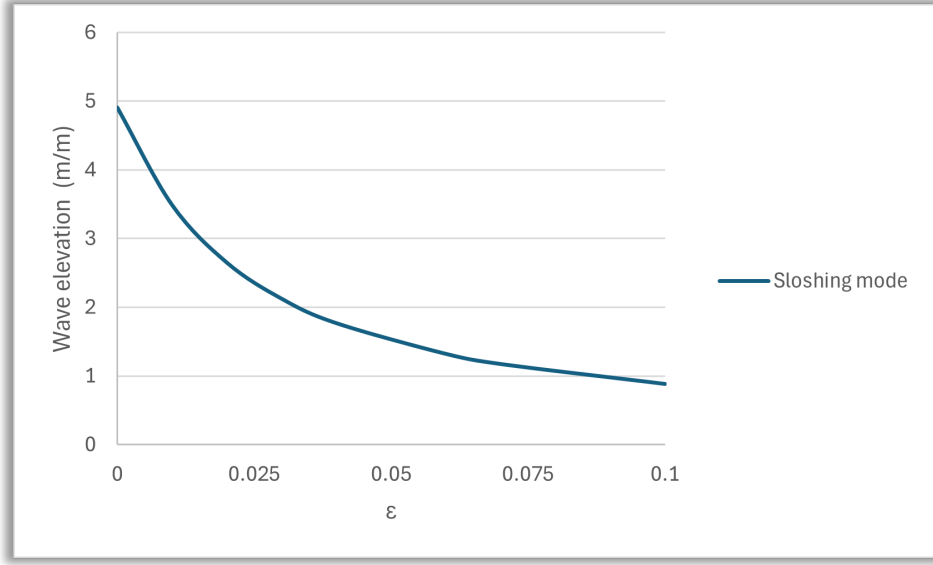


Figure 4.22: Wave elevation - ε relation at $\omega = 1.05$ rad/s.

4.2 Loads on Mooring System

This section examines the effect of the damping lid factor ε on the mooring system under both regular and irregular wave conditions. According to the adopted methodology, the mooring tension must be considered in the analysis, requiring the evaluation of the complete system. Specifically, a 5MW wind turbine is added to the right side of the barge, along with the installation of eight catenary mooring lines, as described in more detail in the previous chapter.

Regular waves

Initially, the mooring line tension is examined under regular wave excitation. For this analysis, the mooring system and the nonlinearities introduced by the mooring lines and the wind turbine are considered. Time-domain simulations are conducted using a wave height of 1 m and wave frequencies of 0.4 rad/s, 0.825 rad/s, 1.05 rad/s, and 1.4 rad/s. Two of these frequencies are outside the moonpool's resonance range, while the other two coincide with the moonpool's resonance frequencies. This selection allows for an assessment of the moonpool's influence on various incoming wave conditions.

Subsequently, the Fourier transform formula is employed to convert the time-domain signals into the frequency domain. According to Du, J., Li, H., Zhang, M., & Wang, S. (2015) [7], the minimum required simulation duration for an accurate transformation into the frequency domain must satisfy the condition:

$$time \geq \frac{2\pi}{\delta\omega}$$

where *time* represents the total simulation time, and $\delta\omega$ denotes the frequency resolution.

For the scope of this thesis, a frequency resolution $\delta\omega$ of 0.04 is selected. Based on this value, the minimum required simulation time is:

$$time \geq \frac{2\pi}{\delta\omega} = \frac{2\pi}{0.04} = 156s$$

To ensure greater accuracy in the results, a total simulation time of 300 s was chosen. After conducting a time-domain simulation to identify the mooring line with the maximum tension, the following results were obtained:

Table 4.1: Mooring Tensions

Mooring number	Tension(N)
1	1549410
2	1553720
3	1614230
4	1617290
5	1617300
6	1553730
7	1549410
8	1553720

As it is observed, the maximum tension is on mooring line 5, so from now and then only the results for this line will be presented.

The first frequency of interest is 0.405 rad/s, which lies in the lower range of the frequency spectrum and is far from the dominant piston and first sloshing modes. The graph below illustrates the time-domain tension response of mooring line 5 for different damping lid factors ε .

It is observed that some variations appear in the signal, affect the main peaks. A frequency-domain analysis will provide a clearer perspective and deeper understanding of the system's behavior.

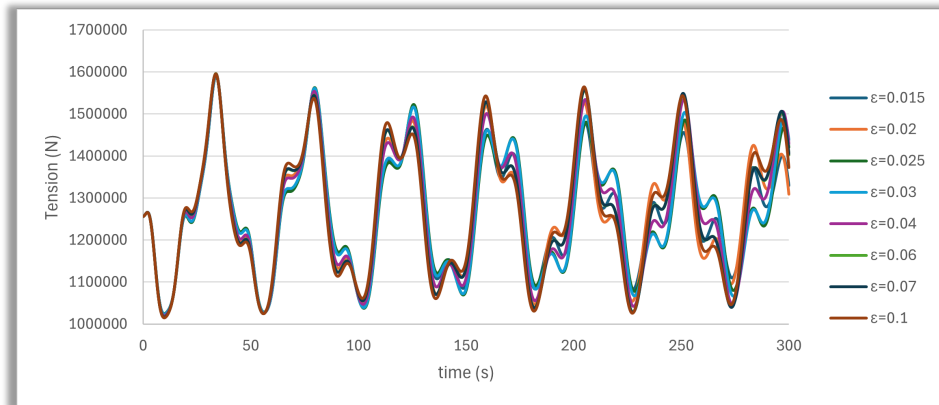


Figure 4.23: Tension signals over time at $\omega = 0.405$ rad/s.

By plotting the frequency-domain curve, it is observed that the system oscillates at the natural frequency of the mooring system as well as at the frequency of the propagating waves. However, at the given scale, other resonance frequencies are not distinctly visible.

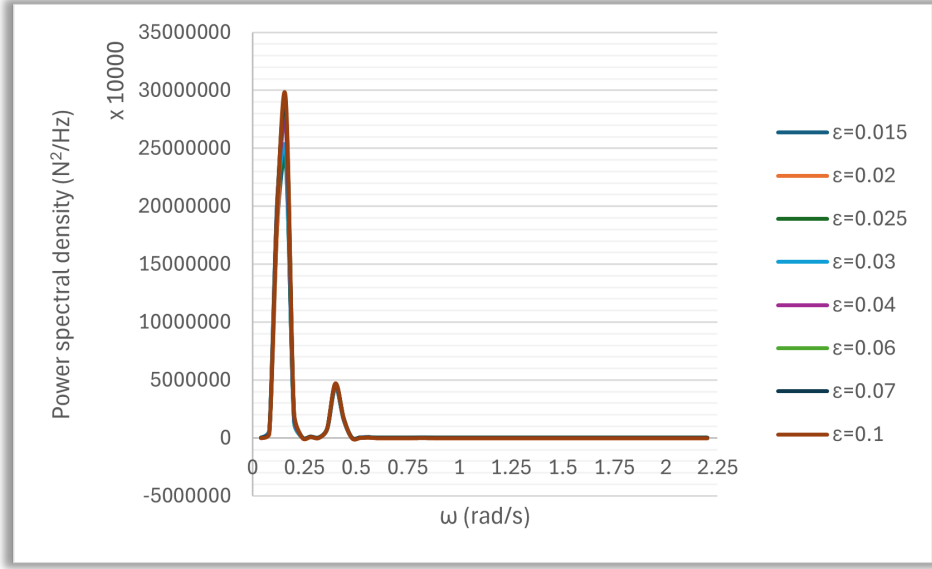


Figure 4.24: PSD signals over frequencies at $\omega = 0.405$ rad/s.

To obtain a clearer representation, the scale is adjusted to a logarithmic scale, which provides a more comprehensive view of the system's response. This adjustment reveals the presence of piston mode and various sloshing modes, with piston mode at 0.82 rad/s being the dominant frequency, as it is closest to the incoming wave frequency.

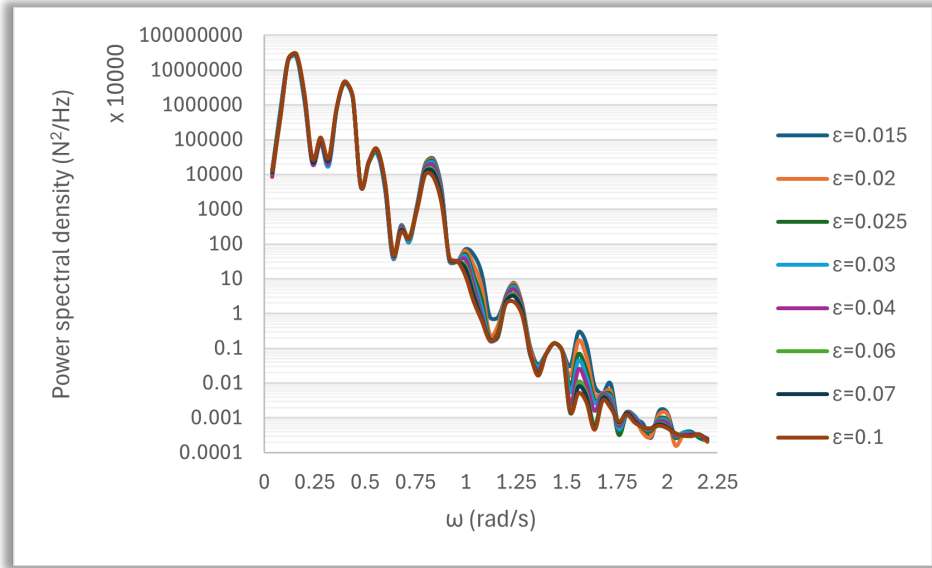


Figure 4.25: PSD signals over frequencies at $\omega = 0.405$ rad/s.

In the graph below the time signals of mooring line 1 tension under the wave frequency of 0.825 rad/s is illustrated for different damping lid factors ε . From this graph it can not said much about the effect of ε , so there is the need of tranfered it to frequency domain in order to see the resonance frequencies.

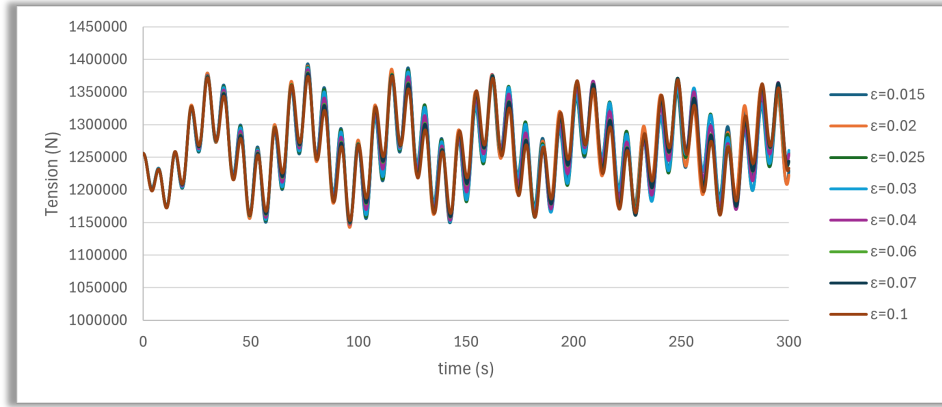


Figure 4.26: Time series at $\omega = 0.825$ rad/s.

In the frequency domain, it is observed that the dominant frequency remains the incoming wave frequency of 0.82 rad/s. However, at this stage, the piston mode of the moonpool is also activated. Additionally, it is evident that the damping lid factor mitigates these unrealistic peaks, contributing to a more stable system response.

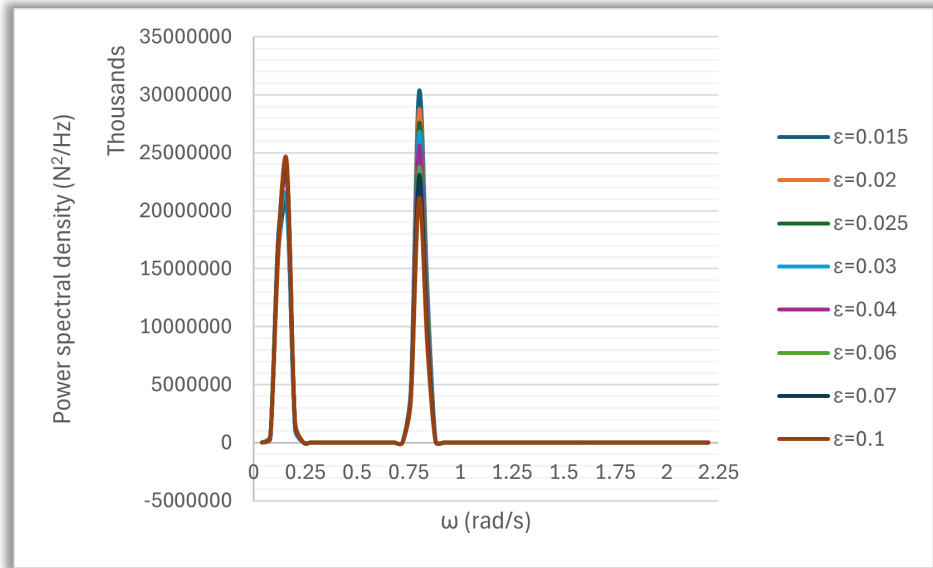


Figure 4.27: PSD signals over frequencies at $\omega = 0.825$ rad/s.

Upon closer examination in the logarithmic scale, it is observed that, alongside the piston mode, the first sloshing mode also plays a dominant role at 1.05 rad/s. Moreover, at higher frequencies, the second, fourth, fifth, and sixth sloshing modes are clearly identified, albeit with comparatively lesser significance.

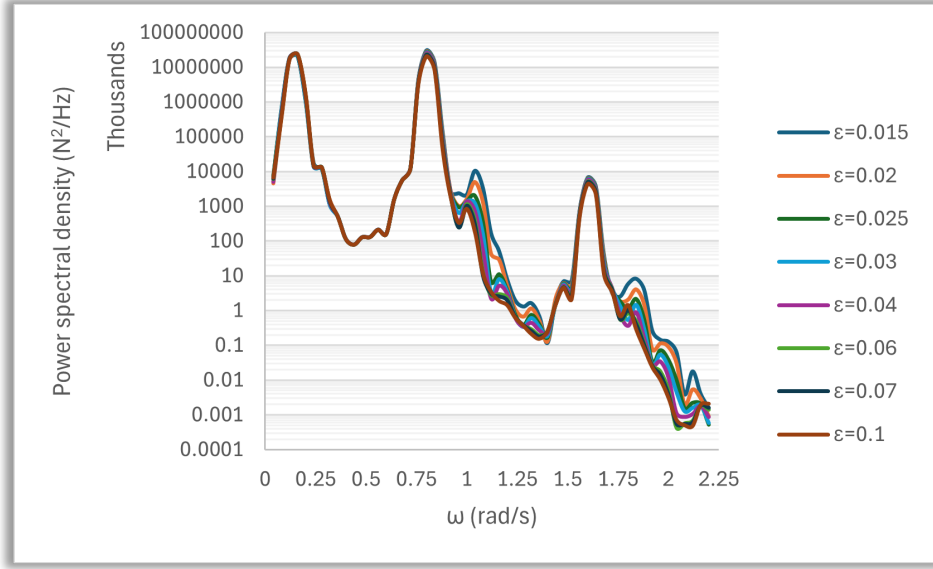


Figure 4.28: PSD signals over frequencies at $\omega = 0.825$ rad/s.

The third incoming wave frequency coincides precisely with the first sloshing resonance frequency at 1.05 rad/s. Observing the time-series graph, it is evident that applying a larger damping lid factor results in a more stable response. By translating the time series once again into the frequency domain, a clearer representation of all affected frequencies is obtained.

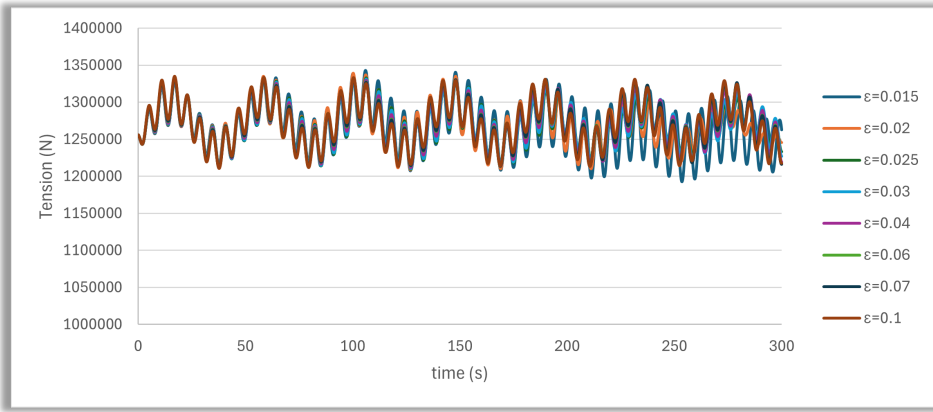


Figure 4.29: Time series at $\omega = 1.05$ rad/s.

The incoming frequency of 1.05 rad/s not only influences the first sloshing mode but also affects the piston mode at 0.82 rad/s, as well as the third, fourth and sixth sloshing modes at 1.53 rad/s, 1.77 rad/s and 2.13 rad/s, respectively.

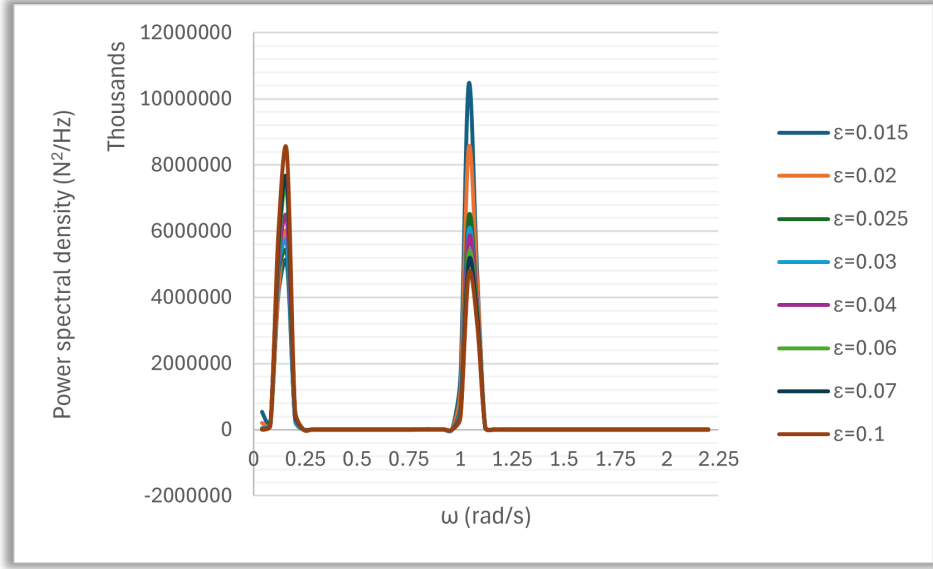


Figure 4.30: PSD signals over frequencies at $\omega = 1.05$ rad/s.

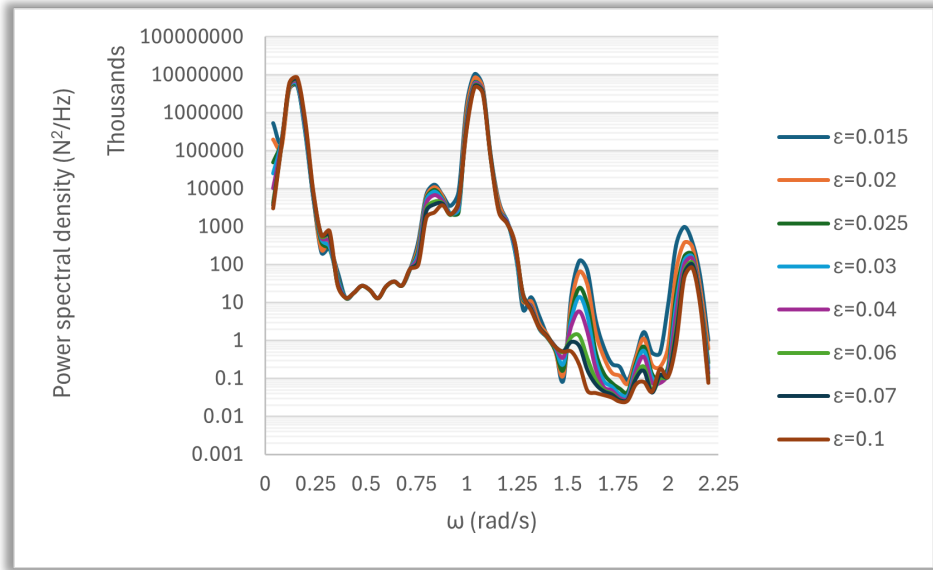


Figure 4.31: PSD signals over frequencies at $\omega = 1.05$ rad/s.

The last examined incoming frequency is 1.4 rad/s, which is higher than the first sloshing mode, the dominant frequency in the system. From the time-series data, it is challenging to clearly observe and analyze the specific influence of the moonpool's presence.

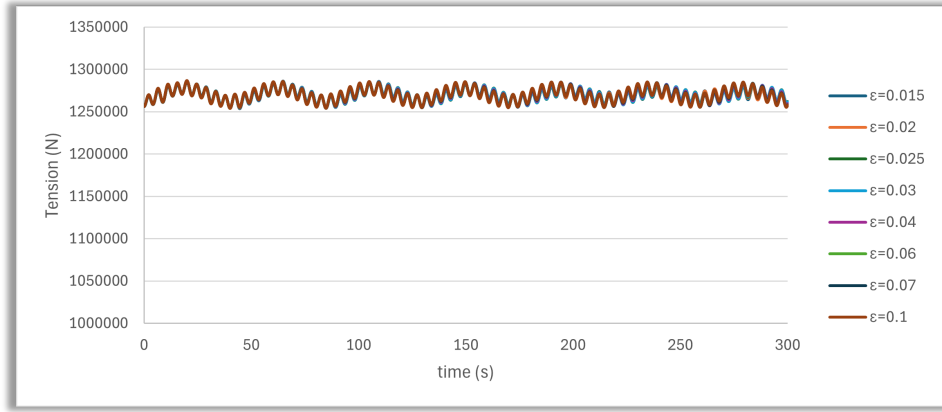


Figure 4.32: Time series at $\omega = 1.4$ rad/s.

However, in the frequency domain, as illustrated in the two graphs below, the incoming wave frequency at 1.4 rad/s is observed, along with the first, third, fourth, and sixth sloshing modes at 1.05 rad/s, 1.52 rad/s, 1.77 rad/s, and 2.13 rad/s, respectively.

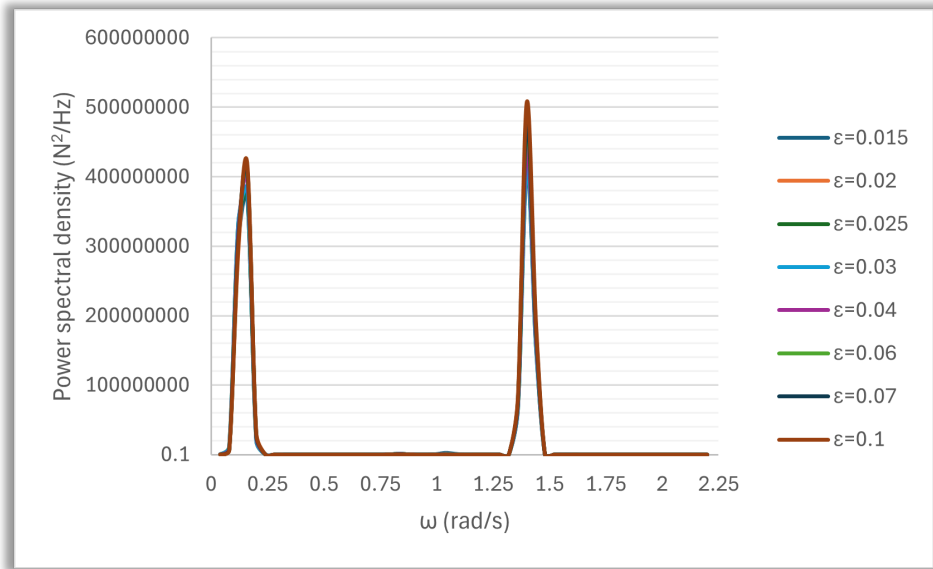


Figure 4.33: PSD signals over frequencies at $\omega = 1.4$ rad/s.

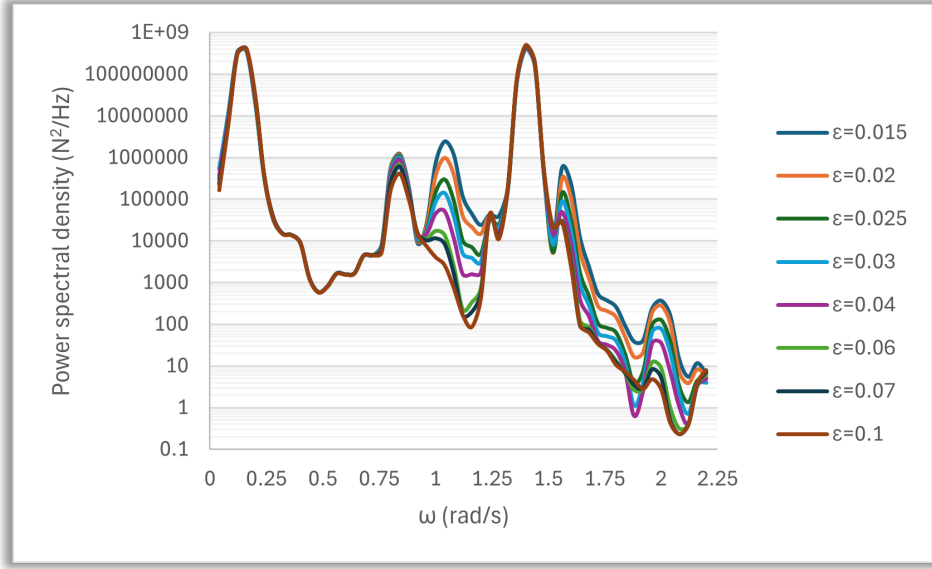


Figure 4.34: PSD signals over frequencies at $\omega = 1.4$ rad/s.

According to previous graphs it is noted that every incoming frequency of regular waves activates more or less all the resonance mode of the moonpool. The dominant resonance mode is still the piston and first sloshing mode so at this point it is essential to plot the relation with ε and examine the results. Below figures illustrates this relation for piston and first sloshing mode for each incoming wave frequency.

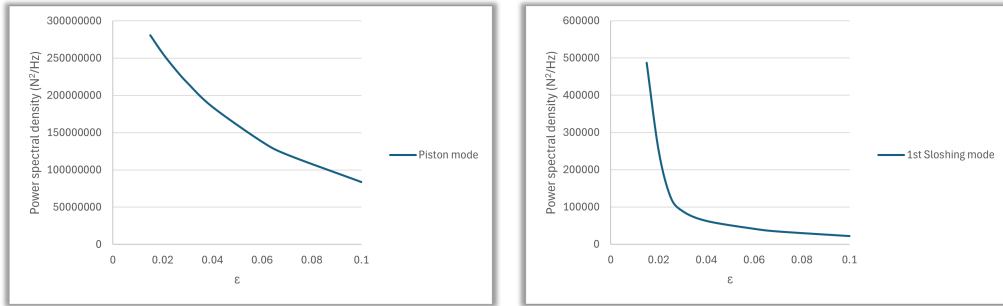


Figure 4.35: Piston mode (left) and 1st Sloshing mode (right) at $\omega = 0.4$ rad/s.

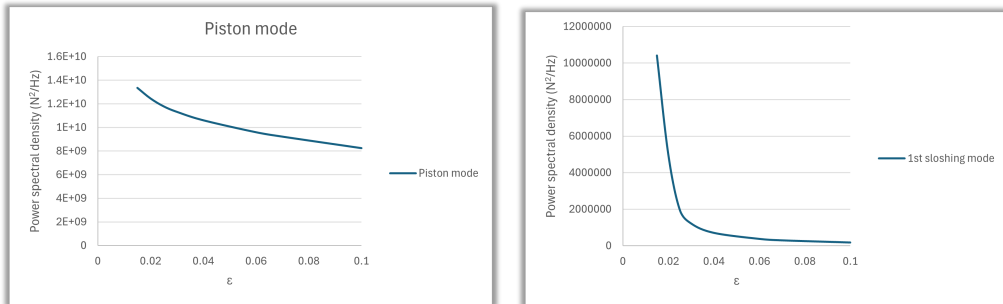


Figure 4.36: Piston mode (left) and Sloshing mode (right) at $\omega = 0.82$ rad/s.

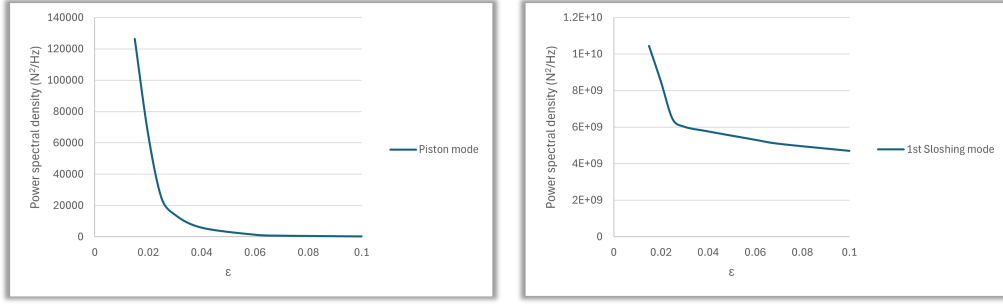


Figure 4.37: Piston mode (left) and Sloshing mode (right) at $\omega = 1.05$ rad/s.

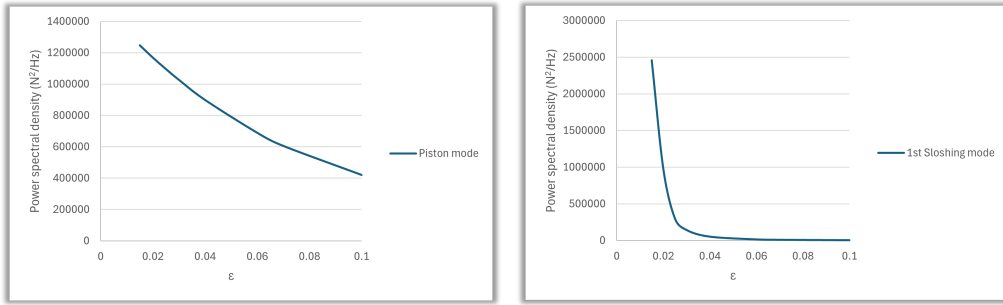


Figure 4.38: Piston mode (left) and Sloshing mode (right) at $\omega = 1.4$ rad/s.

By analyzing the graphs above, it becomes clear that the damping lid factor ε has a strong influence on the behavior of the moonpool, especially at resonance frequencies. Looking at the curves, it is observed that small increases in ε , particularly up to values around 0.030 to 0.035, lead to a large reduction of the resonance effects inside the moonpool. After this range, further increases in the damping lid factor result in smaller changes in the response, indicating that the system becomes less sensitive to ε . This means that applying a damping lid factor larger than 0.030 is enough to significantly decrease the unrealistic peaks caused by the limitations of potential flow theory.

Irregular waves

According to the methodology, the next step in understanding the influence of the moonpool's presence on the system is to evaluate its behavior under irregular incoming waves. For this purpose, the JONSWAP spectrum was selected, with a wave height of 1 m corresponding to regular waves and peak frequencies at 0.4 rad/s, 0.82 rad/s, 1.05 rad/s, and 1.4 rad/s, allowing for direct comparison with previous results and evaluate possible differences between tow types of waves.

The figure below represents the time-series signals for the peak wave period of 0.4 rad/s. It is observed that the damping lid factor influences the system's response.

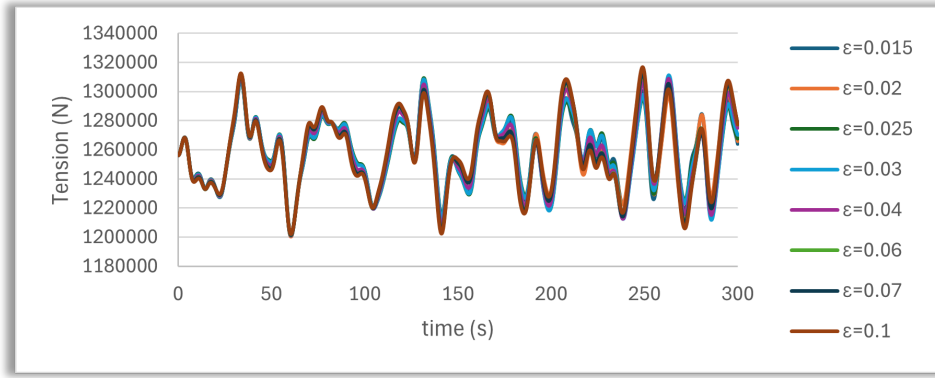


Figure 4.39: Time series at $\omega = 0.4$ rad/s.

As in previous cases, resonance at the moonpool's modes is observed, with dominant resonances occurring at 0.82 rad/s for the piston mode and 1.05 rad/s for the first sloshing mode.

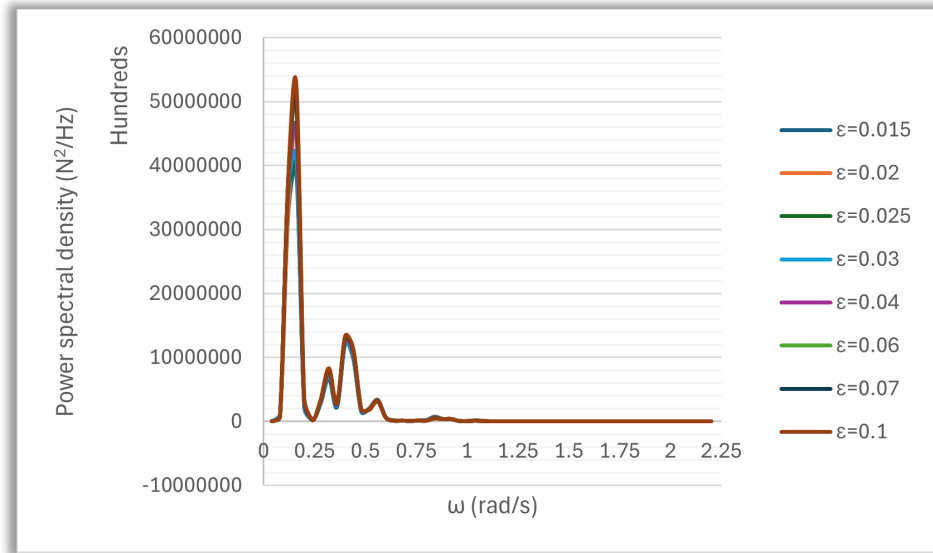


Figure 4.40: PSD signals over frequencies at $\omega = 0.4$ rad/s.

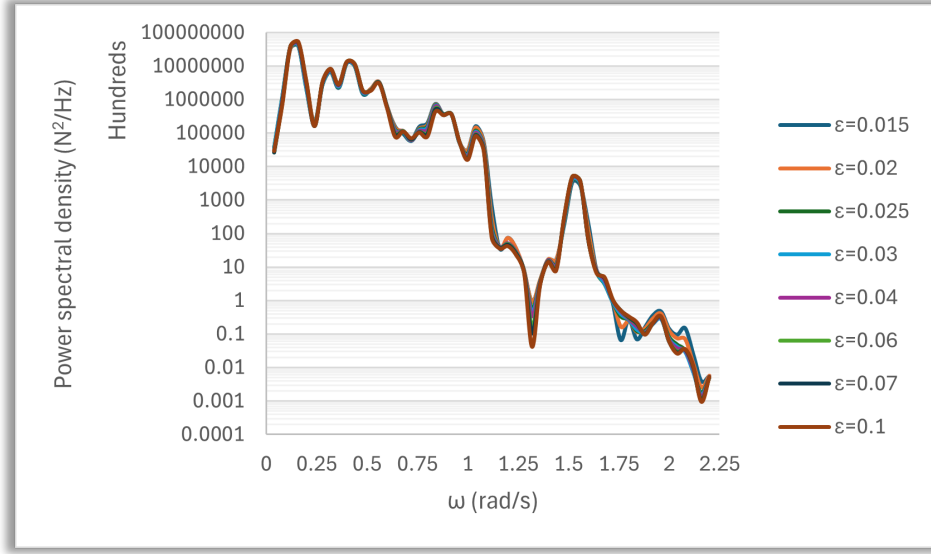


Figure 4.41: PSD signals over frequencies at $\omega = 0.4$ rad/s.

As in previous cases, using the same methodology, the time-series data for the incoming wave spectrum with a peak frequency of 0.82 rad/s is presented below.

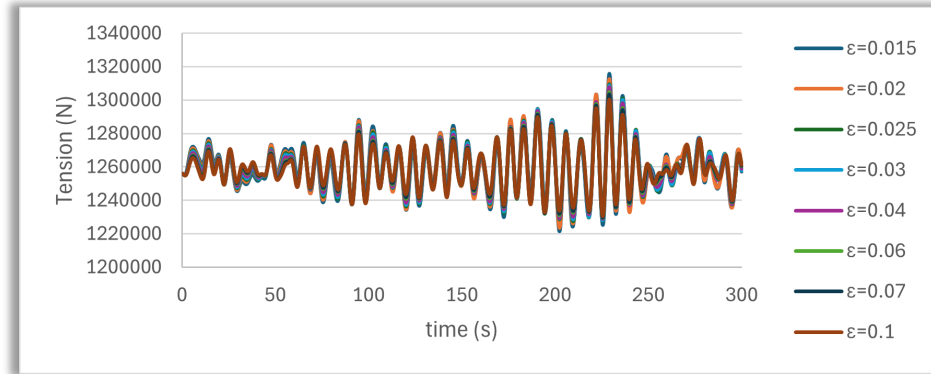


Figure 4.42: Time series at $\omega = 0.82$ rad/s.

Compared to regular waves, a greater number of frequencies near the peak frequency of the spectrum are activated, as expected. Once again, the dominant frequencies of moonpool resonance are the piston mode and the first sloshing mode.

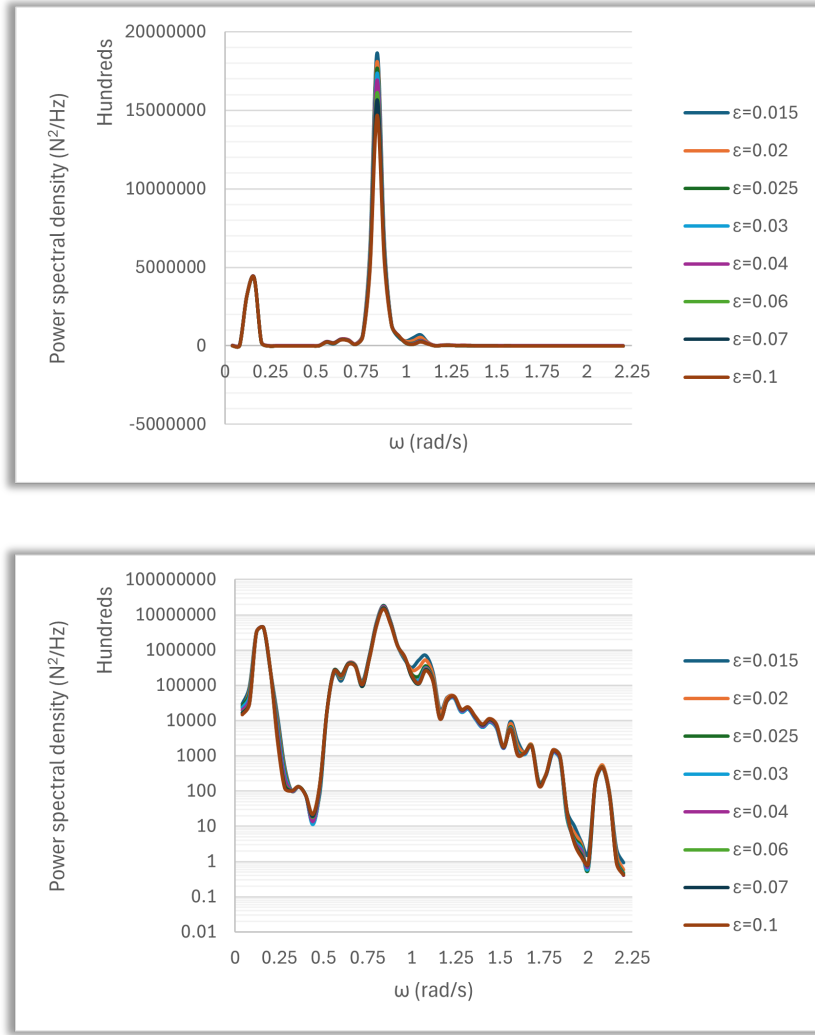


Figure 4.43: PSD signals over frequencies at $\omega = 0.82$ rad/s.

At the incoming wave frequency of 1.05 rad/s which is the 1st sloshing resonance mode, it is observed that

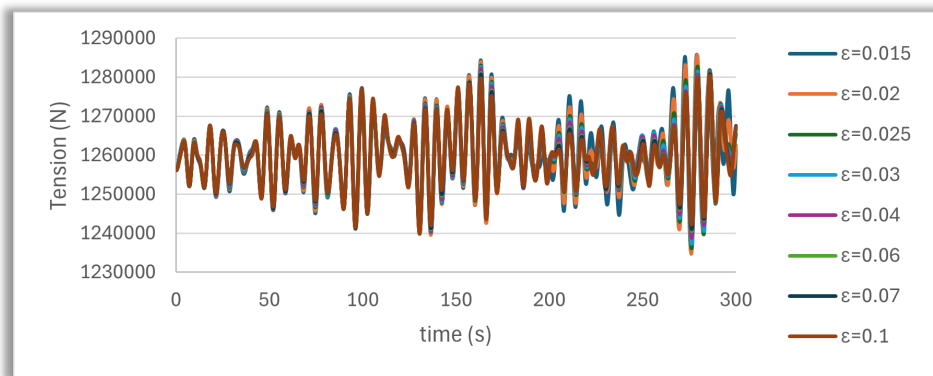


Figure 4.44: Time series at $\omega = 1.05$ rad/s.

At the incoming wave frequency of 1.05 rad/s, which corresponds to the first sloshing resonance mode, it is observed that the spectrum significantly influences the piston mode

due to its proximity to the peak frequency. This indicates that when a spectrum is near the first sloshing mode, it simultaneously affects and activates the dominant resonant mode of the moonpool. Additionally, the damping lid factor appears to effectively mitigate both of these peaks.

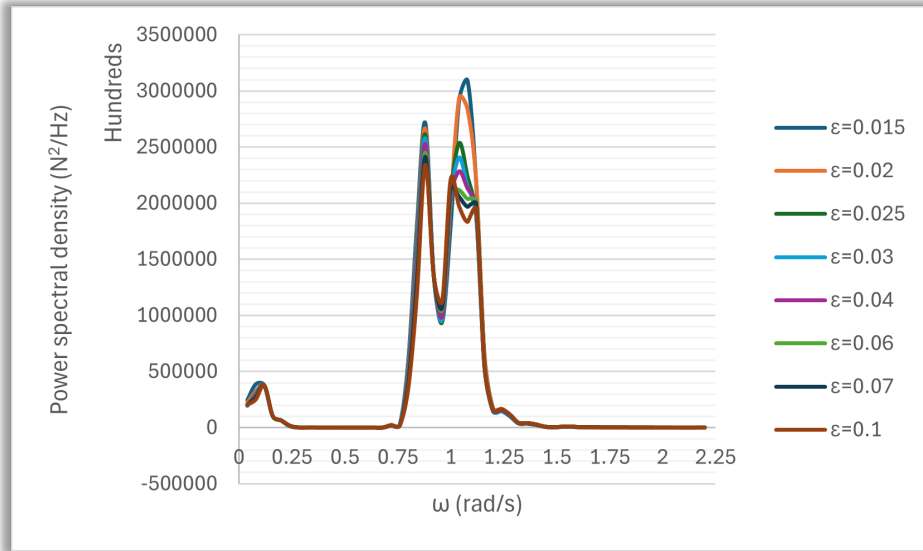


Figure 4.45: PSD signals over frequencies at $\omega = 1.05$ rad/s.

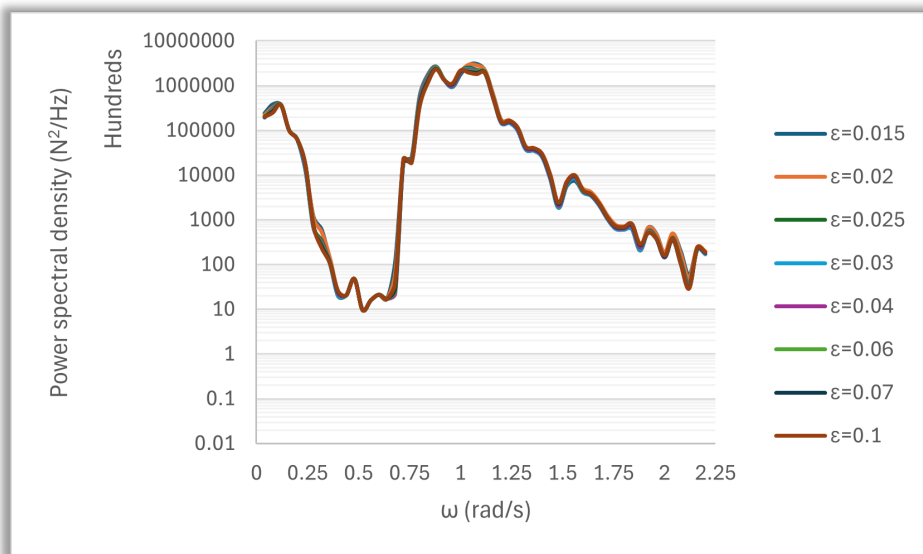


Figure 4.46: PSD signals over frequencies at $\omega = 1.05$ rad/s.

Finally, for the incoming wave frequency of 1.4 rad/s, it is observed that the time-series data resembles that of a regular wave. This occurs because the peak frequency is significantly distant from the natural frequency of the mooring system, resulting in a reduced impact on the system's response.

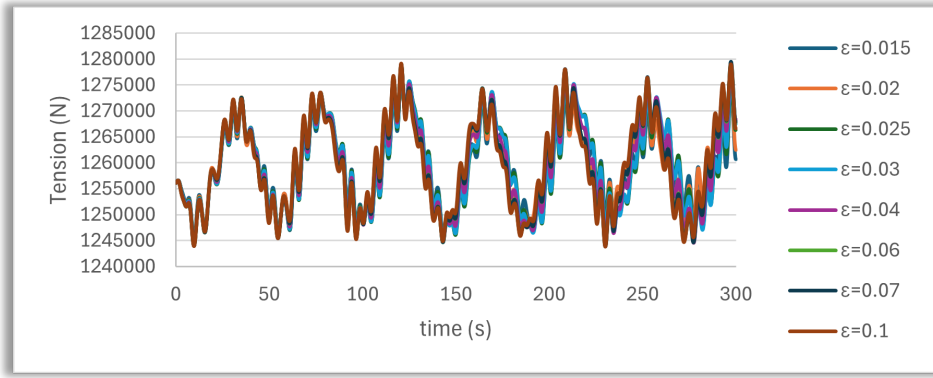


Figure 4.47: Time series at $\omega = 1.4$ rad/s.

Once again, the dominant resonance modes observed are the piston mode at 0.82 rad/s and the first sloshing mode at 1.05 rad/s. The damping lid factor appears to have a significant influence on these modes.

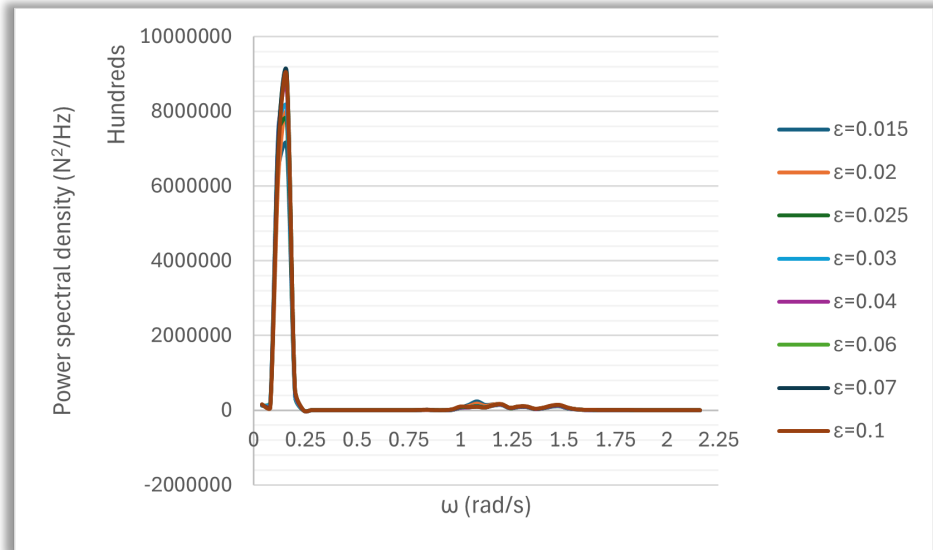


Figure 4.48: PSD signals over frequencies at $\omega = 1.4$ rad/s.

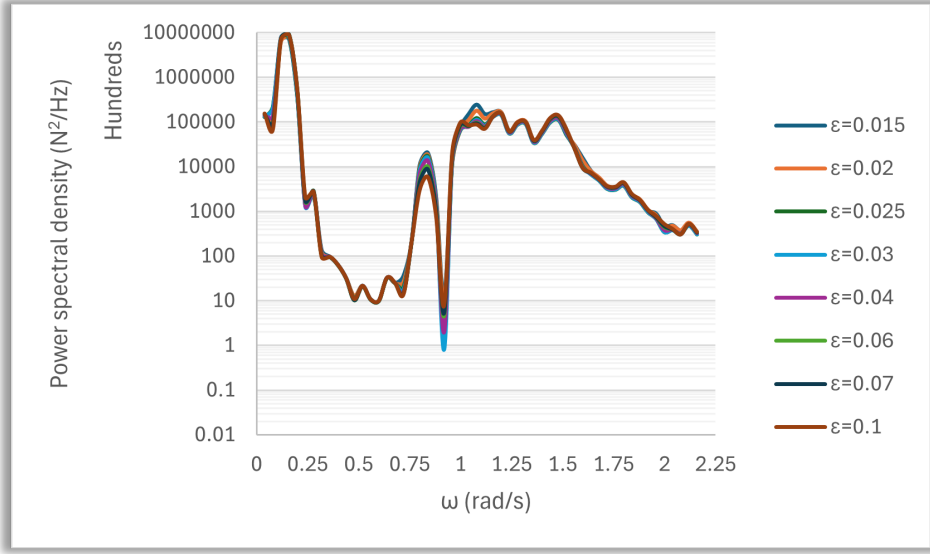


Figure 4.49: PSD signals over frequencies at $\omega = 1.4$ rad/s.

At this stage, it is essential to analyze and visualize the relationship between the damping lid factor ε and the mooring tension. The following graphs illustrate these dependencies across different incoming wave spectra. In all cases, comparisons are based on the highest observed force tension.

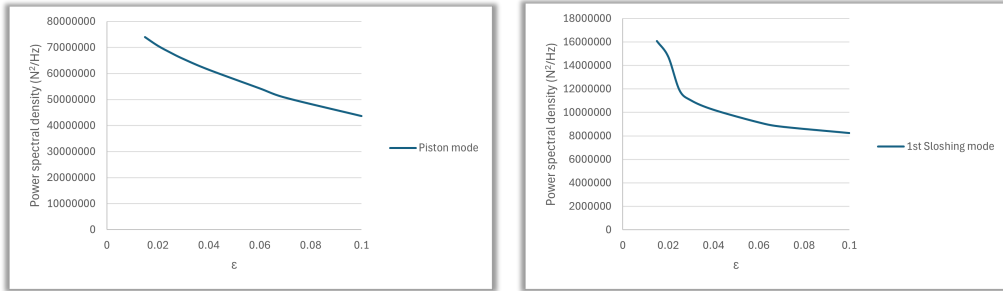


Figure 4.50: Piston mode (left) and 1st Sloshing mode (right) at $\omega = 0.4$ rad/s.

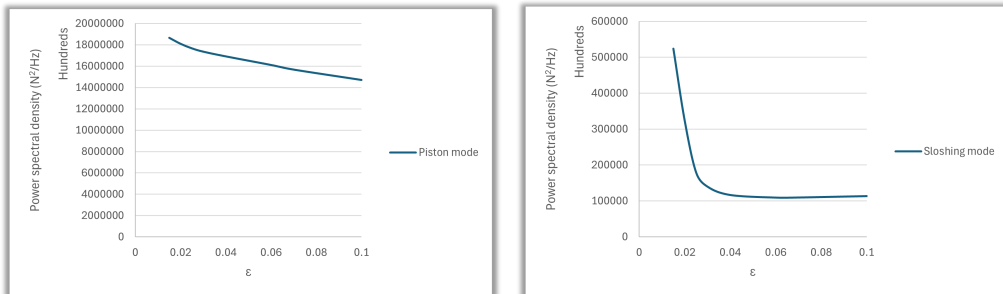


Figure 4.51: Piston mode (left) and Sloshing mode (right) at $\omega = 0.82$ rad/s.



Figure 4.52: Piston mode (left) and Sloshing mode (right) at $\omega = 1.05$ rad/s.

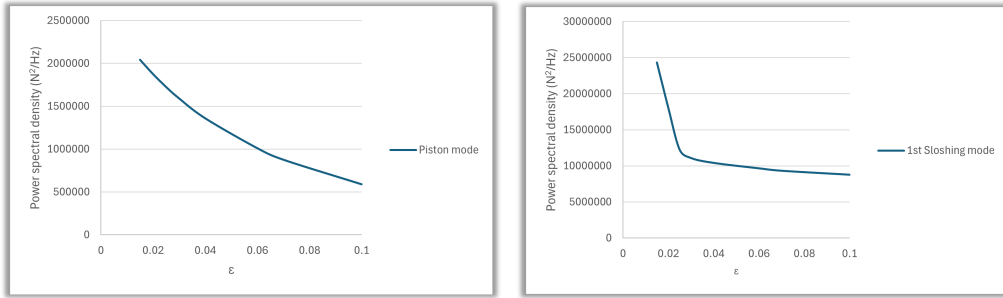


Figure 4.53: Piston mode (left) and Sloshing mode (right) at $\omega = 1.4$ rad/s.

It is observed that, particularly in the first sloshing mode, once the damping lid factor ε exceeds 0.03, the slope of the graph gradually smooths out, and at certain incoming wave frequencies, it stabilizes and remains constant.

From a theoretical perspective, potential flow theory does not account for the viscosity of water or fluid separation, and therefore, it neglects the additional damping effects produced by these two mechanisms. As presented in this study, after the damping lid factor reaches 0.03, the slope of the graph—both for surface elevation and mooring tension—becomes smoother and, in some instances, constant. Consequently, this suggests that the damping lid effectively mitigates unrealistic values.

To further investigate the precise value of the damping lid factor at which the system yields realistic results, the following graphs illustrate the three-hour test of the structure under irregular wave conditions with wave height of 1 meter for damping lid factor values of 0.03, 0.032, and 0.035.

The first figure presents the time series for a damping lid factor value of 0.03. It reveals unrealistic values in the heave motion, where the magnitude reaches four times the wave height. This indicates that the excessive peaks in surface elevation within the moonpool were not effectively damped.

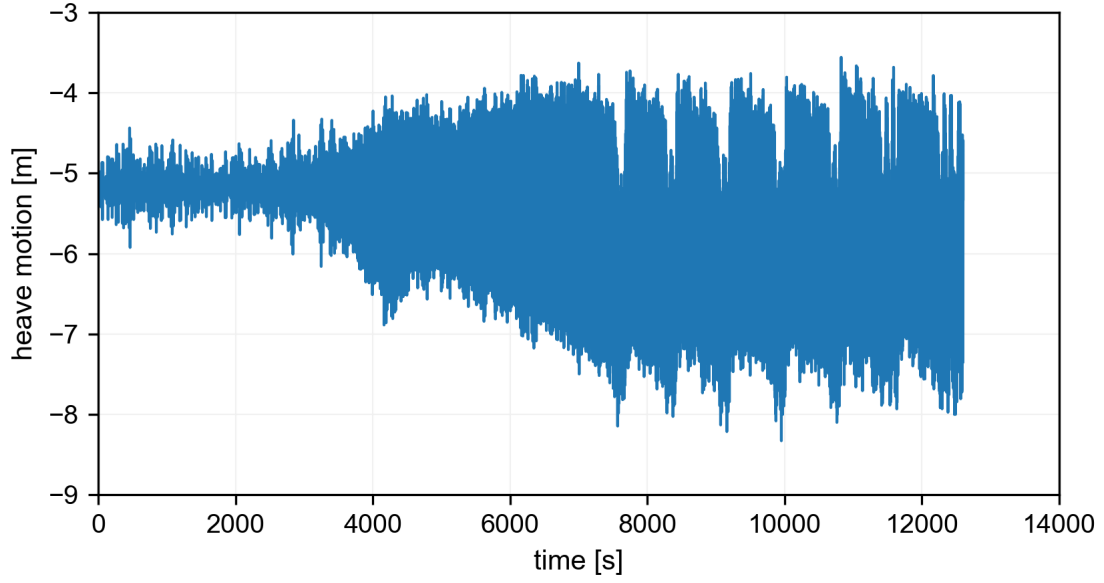


Figure 4.54: Time series - heave motion - $\varepsilon = 0.03$

The second figure presents the time series for the damping lid factor value of 0.032. It reveals again unrealistic values in the heave motion, where the magnitude reaches one and a half times the wave height.

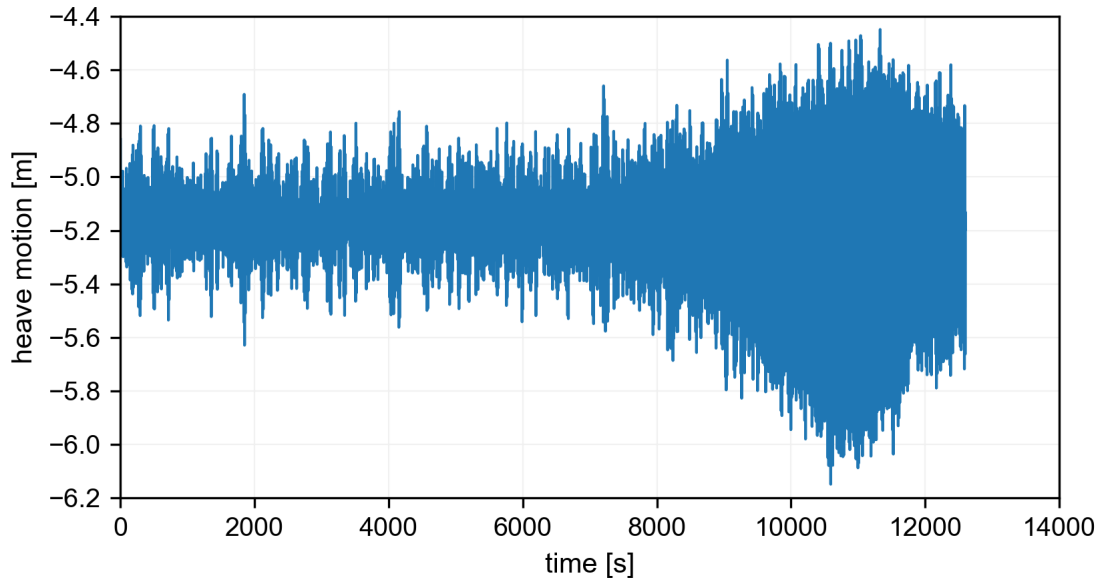


Figure 4.55: Time series - heave motion - $\varepsilon = 0.032$

For a damping lid factor value of 0.035, no unrealistic values are observed in the heave motion. This suggests that at this value, the excessive peaks in free surface elevation within the moonpool are effectively damped, bringing the system into a realistic and stable condition.

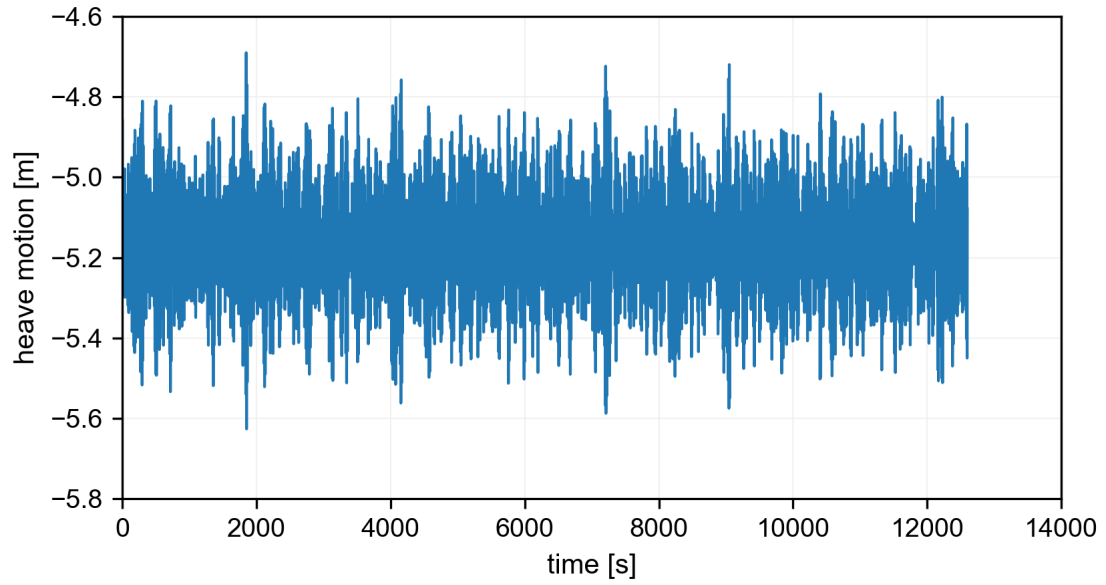


Figure 4.56: Time series - heave motion - $\varepsilon = 0.035$

One remaining issue is that for damping lid factor values greater than 0.035, there is no definitive proof that the realistic value is 0.035 or 0.04, rather than, for instance, 0.1.

Chapter 5

Case study

This chapter presents a case study to gain a deeper understanding of the effects of the moonpool's presence in a barge-type platform and the influence of the damping lid factor on an operational system under real sea states, the final section of this thesis identifies a viable location for deploying this type of floating offshore wind turbine and presents a corresponding case study.

According to DNV standards [6] and serviceability limit state, a wind turbine can generate electricity when the maximum pitch motion is less than 10 degrees and the maximum acceleration on the top of the tower is less than 0.3g. So, the scope of this case study and last part of this thesis is to investigate the affection of damping lid factor ε to these above mentioned conditions. In parallel with that, a check on ultimate limit state

According to previous chapter observations, a damping lid of 0.035 and 0.1 used to examine the affection on operability limits of the wind turbine.

5.1 Inputs - Assumptions

This section explains all the inputs and assumptions used in this case study. Each detail is carefully described to make sure the analysis is accurate and reliable.

Location

According to Edwards et al. (2023) [8], upcoming projects featuring this type of floater are planned for north Scotland and the Bay of Biscay in western France. For the purposes of this research, north Scotland was selected as the case study location. Additionally, to enhance this choice, a review of existing literature identified a specific area in North Scotland—referenced in Edwards et al.'s paper—where there is future interest in installing 60 MW of floating wind turbines, as illustrated in the following figure with red color. This location is placed at 30 km from the coast of Crosskirk with a depth of 50 m.



Figure 5.1: Location — North Scotland [1]

Wind Turbine configuration

As illustrated in the figures below, the wind turbine configuration consists of a barge-type floater with dimensions of $60 \text{ m} \times 60 \text{ m} \times 15 \text{ m}$ ($L \times W \times T$). The barge features a centrally positioned square moonpool measuring $40 \text{ m} \times 40 \text{ m}$.

A 5 MW NREL-type wind turbine with a tower height of 90 m is installed on the right side of the barge. The rotor comprises three blades with a diameter of 126 m. Additionally, eight catenary moorings are arranged as detailed in Chapter 2.

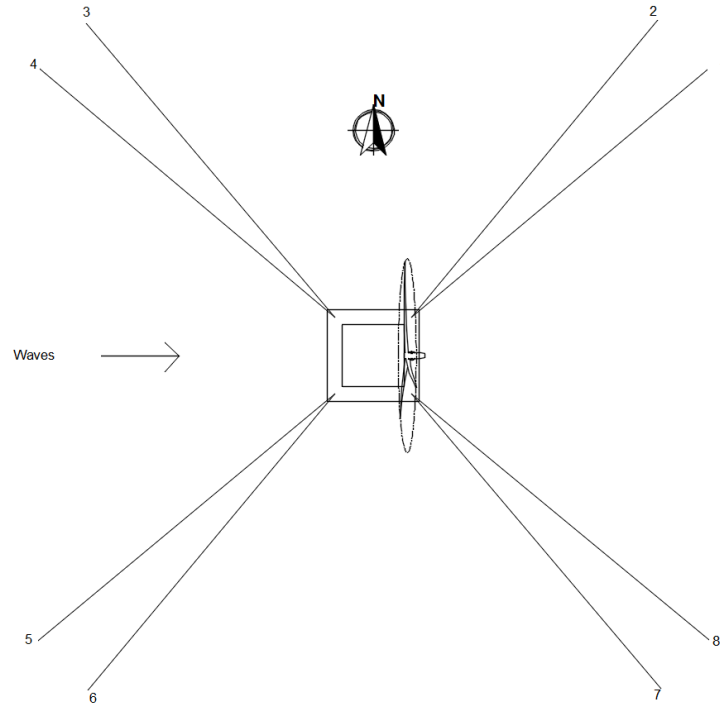


Figure 5.2: Top view — Wind turbine configuration.

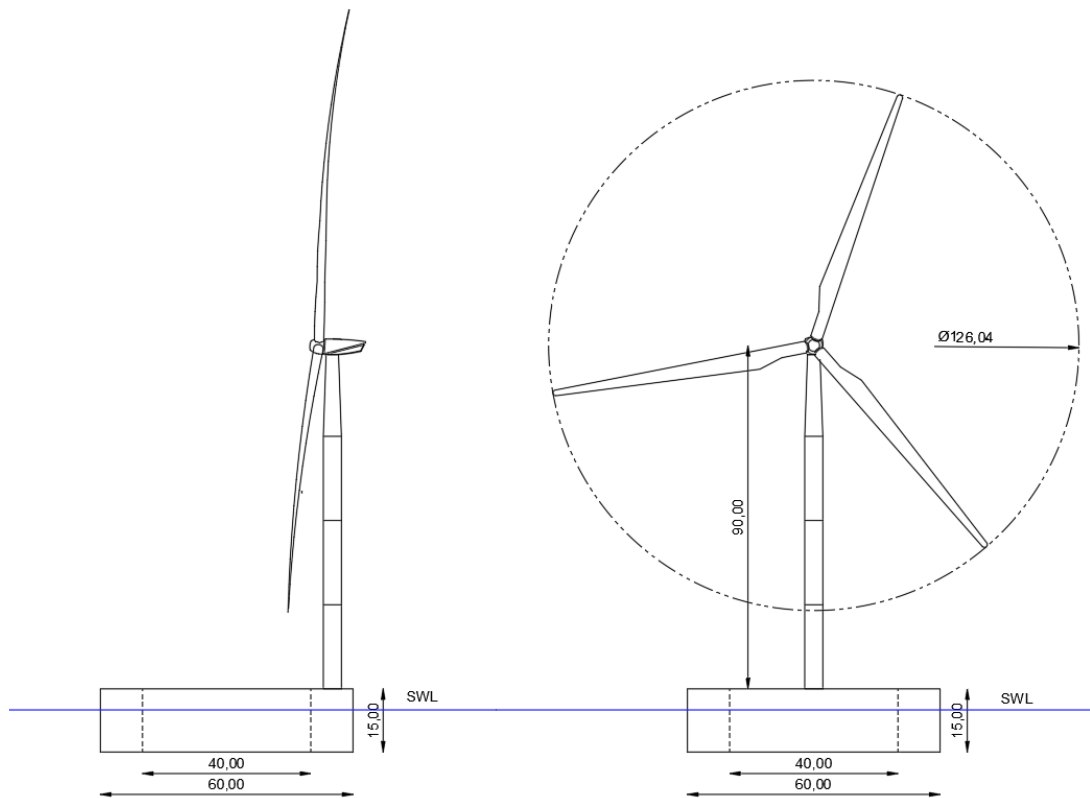


Figure 5.3: Side view (left) , front view (right)

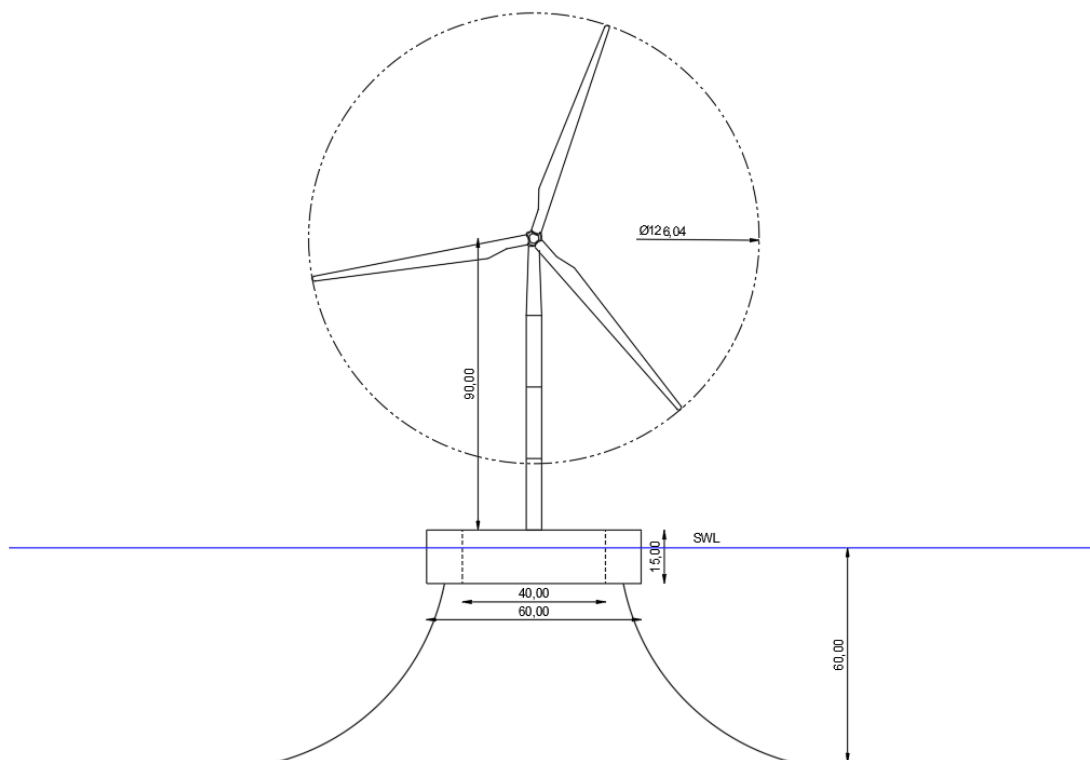


Figure 5.4: Front view with moorings

Moreover, it is essential to be refereed that the cut in, rated and cut out wind speed is 3 m/s, 11.4 m/s and 25 m/s respectively which translated to 6.9 rpm and 12.1 rpm rotor speed. At this research is assumed that the rotor speed is constant at rated speed of 12.1 rpm.

Environment

Environmental data spanning forty years, with observations recorded every three hours, was analyzed to derive the following table, which presents the probability of wave height occurrence across different directions. The analysis indicates that the most probable wave height is 1.5 m, predominantly originating from the west with a probability of 12.53 %.

Hm0 m	N 0	NE 45	E 90	SE 135	S 180	SW 225	W 270	NW 315	All classes
0.5	0.397	0.018	0.027	0.005	0.002	0.011	2.867	0.363	3.689184
1	2.824	0.100	0.283	0.174	0.000	0.070	11.969	1.276	16.6965
1.5	3.701	0.097	0.263	0.220	0.016	0.063	12.527	1.661	18.54789
2	3.542	0.091	0.163	0.159	0.011	0.048	10.705	1.278	15.99628
2.5	2.665	0.077	0.163	0.145	0.005	0.018	8.516	1.144	12.73312
3	1.824	0.059	0.068	0.161	0.000	0.011	6.916	0.929	9.968501
3.5	1.267	0.036	0.048	0.052	0.000	0.002	5.287	0.523	7.21521
4	0.872	0.011	0.014	0.011	0.002	0.000	3.558	0.344	4.813161
4.5	0.560	0.000	0.007	0.009	0.002	0.000	2.579	0.238	3.394593
5	0.354	0.000	0.002	0.005	0.000	0.000	1.949	0.168	2.476829
5.5	0.265	0.002	0.000	0.002	0.000	0.000	1.289	0.116	1.674636
6	0.177	0.000	0.000	0.000	0.000	0.000	0.845	0.052	1.074124
6.5	0.111	0.000	0.000	0.000	0.000	0.000	0.503	0.025	0.639036
7	0.075	0.000	0.000	0.000	0.000	0.000	0.347	0.050	0.471345
7.5	0.050	0.000	0.000	0.000	0.000	0.000	0.229	0.014	0.292325
8	0.027	0.000	0.000	0.000	0.000	0.000	0.156	0.011	0.194883
8.5	0.002	0.000	0.000	0.000	0.000	0.000	0.036	0.005	0.043056
9	0.014	0.000	0.000	0.000	0.000	0.000	0.029	0.002	0.045322
9.5	0.000	0.000	0.000	0.000	0.000	0.000	0.020	0.000	0.020395
10	0.002	0.000	0.000	0.000	0.000	0.000	0.009	0.000	0.01133
10.5	0.000	0.000	0.000	0.000	0.000	0.000	0.002	0.000	0.002266
Total	18.72691	0.49174	1.037866	0.942691	0.038523	0.224342	70.33923	8.19869	100

Figure 5.5: Probability of occurrence

As the spectral analysis of the data is out of this thesis scope, JONSWAP spectrum used to calculate for each cell of table above the force tension, pitch motion and tower top acceleration.

Assumptions

At this point all the assumptions considered at this analysis are presented in following table.

Table 5.1: Assumptions

Type	Assumption
Platform (barge L * W * T)	60 * 60 * 10
Moonpool (L * W * T)	40 * 40 * 10
Wind Turbine	5MW NREL
Rotor diameter	126 m
Rotor speed (Rated)	12.1 rpm constant
Mooring line — Water depth	8 * Catenary — 50 m
Wave spectrum	JONSWAP
Swell	No
Wind	aligned with waves
Wind spectrum	No - constant
Current	No

5.2 Results

This section presents the results of the case study, beginning with the damping lid factor of 0.035 and ending 0.1. The analysis includes tables detailing pitch motion, tower top acceleration, and mooring tensions.

The following tables show the maximum mooring tension for a three-hour simulation for each wave height wave period wind and direction. It is essential to note here that the nacelle and blades look towards wind and waves in every direction.

Hm0 m	N	NE	E	SE	S	SW	W	NW
	0	45	90	135	180	225	270	315
0.5	1291550	1297890	1323600	1278260	1227010	1233440	1245990	1235010
1	1340870	1444200	1442510	1284980		1257560	1258550	1239010
1.5	1460830	1625240	1770370	1315370	1268290	1420090	1279830	1257720
2	1549680	1990320	2225150	1349640	1376720	1472720	1309200	1260100
2.5	1929750	2190100	2586530	1428770	1631990	1571790	1346100	1283040
3	1821840	2427010	3975880	1453280		1828060	1408860	1301540
3.5	1972900	2769750	2628640	1659030		1933530	1459110	1338450
4	2181630	3257010	3086210	1598830	1726250	0	1573190	1370690
4.5	2335850		3735210	1743440	1864120	0	1672440	1353230
5	2676260		4288170	1936020	0	0	1767470	1401580
5.5	2842570	5809310	0	1954300	0	0	1830660	1424970
6	3286860	0	0	0	0	0	2221390	1525400
6.5	3983480	0	0	0	0	0	2289720	1514840
7	4048660	0	0	0	0	0	2531730	1637970
7.5	5340220	0	0	0	0	0	2643440	1811830
8	4912030	0	0	0	0	0	2739000	1858330
8.5	4957390	0	0	0	0	0	3385600	2141550
9	5643360	0	0	0	0	0	3410880	2438710
9.5	0	0	0	0	0	0	4264790	
10	6351720	0	0	0	0	0	4471280	
10.5	0	0	0	0	0	0	4635580	

Hm0 m	N	NE	E	SE	S	SW	W	NW
	0	45	90	135	180	225	270	315
0.5	1308900	1303800	1303330	1211720	1192560	1214780	1234790	1247100
1	1371720	1469270	1402580	1214450		1222350	1245890	1259750
1.5	1546510	1659770	1666400	1222400	1179020	1366250	1261800	1296510
2	1649460	2036120	2019580	1240670	1285390	1426520	1280440	1335320
2.5	2153400	2262720	2298480	1301700	1526150	1477350	1306310	1389150
3	2014440	2479240	3349250	1321890		1742160	1356040	1425000
3.5	2201940	2831800	2337190	1435610		1827120	1392490	1465280
4	2488750	3284660	2717080	1446430	1636010	0	1487340	1564410
4.5	2653830		3265060	1516800	1776740	0	1559550	1570410
5	3111540		3672010	1793370	0	0	1642860	1630760
5.5	3365140	5961170	0	1722920	0	0	1699900	1769990
6	3681540	0	0	0	0	0	2011240	1886590
6.5	4589110	0	0	0	0	0	2061070	1850940
7	4915720	0	0	0	0	0	2236980	1995470
7.5	6315370	0	0	0	0	0	2324450	2091810
8	5994450	0	0	0	0	0	2388360	2235880
8.5	6099590	0	0	0	0	0	2889540	2295350
9	7009590	0	0	0	0	0	2881390	2556400
9.5	0	0	0	0	0	0	3564540	0
10	7769710	0	0	0	0	0	3643910	0
10.5	0	0	0	0	0	0	3859190	0

Figure 5.6: Mooring line 1 (left) and Mooring line 2 (right) for $\varepsilon=0.035$

Hm0 m	N	NE	E	SE	S	SW	W	NW
	0	45	90	135	180	225	270	315
0.5	1305550	1243950	1198610	1145300	1227370	1254800	1312840	1325090
1	1357430	1251230	1190590	1163980		1284180	1363020	1394390
1.5	1499910	1260530	1304280	1196160	1260110	1332390	1481440	1567390
2	1590940	1290510	1487150	1246600	1415120	1328350	1596520	1774730
2.5	2005890	1316190	1502150	1535390	1703530	1396430	1728770	2030190
3	1897740	1347880	2302770	1659600		1425430	1921160	2233570
3.5	2008710	1362560	1526650	1951280		1456640	2066120	2554620
4	2301180	1484040	1783560	1888660	2317660	0	2132120	2695780
4.5	2370370		1875910	2447630	2345880	0	2316350	3069660
5	2857980		2222970	2613470	0	0	2784900	3645770
5.5	3206450	1773670	0	2996220	0	0	3018850	4521180
6	3030400	0	0	0	0	0	3575770	5065120
6.5	3976630	0	0	0	0	0	4337970	4777960
7	4436780	0	0	0	0	0	4724490	5219730
7.5	5821190	0	0	0	0	0	5182450	5403120
8	5449490	0	0	0	0	0	5213950	8616470
8.5	5545270	0	0	0	0	0	5994020	9665600
9	6606960	0	0	0	0	0	5671900	11020800
9.5	0	0	0	0	0	0	6866920	
10	7349080	0	0	0	0	0	7487040	
10.5	0	0	0	0	0	0	8475840	

Hm0 m	N	NE	E	SE	S	SW	W	NW
	0	45	90	135	180	225	270	315
0.5	1303790	1241450	1184990	1106400	1202000	1247690	1317820	1330020
1	1354220	1251370	1176760	1122650		1281780	1367460	1403880
1.5	1478660	1265280	1319350	1131630	1181480	1326500	1487030	1586640
2	1565060	1299780	1545790	1163400	1294550	1350530	1602360	1807990
2.5	1904810	1356990	1565560	1397280	1494480	1390680	1734140	2074550
3	1829920	1379430	2624190	1488340		1428490	1926160	2288530
3.5	1924830	1407620	1598780	1745810		1448900	2066910	2634790
4	2172680	1540660	1909400	1697010	1997910	0	2128330	2811960
4.5	2228910		2023320	2208770	1945370	0	2325590	3168930
5	2609170		2461300	2400020	0	0	2798610	3816820
5.5	2924980	2089050	0	2678290	0	0	3037720	4630250
6	2924840	0	0	0	0	0	3596820	5216240
6.5	3326810	0	0	0	0	0	4359340	4889450
7	3854930	0	0	0	0	0	4726930	5262830
7.5	5227660	0	0	0	0	0	5192610	6019050
8	4708360	0	0	0	0	0	5218070	8508410
8.5	4825540	0	0	0	0	0	5973550	9712480
9	5642830	0	0	0	0	0	5674220	11056600
9.5	0	0	0	0	0	0	6871000	0
10	6093320	0	0	0	0	0	7477580	0
10.5	0	0	0	0	0	0	8511070	0

Figure 5.7: Mooring line 3 (left) and Mooring line 4 (right) for $\varepsilon=0.035$

Hm0 m	N 0	NE 45	E 90	SE 135	S 180	SW 225	W 270	NW 315
0.5	1230290	1194500	1183560	1247060	1310450	1340110	1317820	1244610
1	1230110	1178620	1173810	1250400		1632330	1367460	1255000
1.5	1280580	1181070	1324500	1263310	1562800	1885560	1487030	1276550
2	1248730	1280880	1541810	1287660	1805330	2019850	1602360	1292910
2.5	1453070	1318040	1548280	1348570	2333640	2823080	1734140	1320410
3	1375120	1455340	2567860	1418450		3086850	1926160	1352920
3.5	1319040	1519730	1580170	1608150		3795160	2066910	1396010
4	1376280	1757060	1868000	1567850	2741470	0	2128330	1413270
4.5	1489220		2116270	1690240	2865180	0	2325590	1419460
5	1587370		2432230	1978140	0	0	2798610	1464250
5.5	1684160	2532730	0	2032550	0	0	3037720	1473330
6	2035540	0	0	0	0	0	3596820	1527950
6.5	2347310	0	0	0	0	0	4359340	1597680
7	2371840	0	0	0	0	0	4726930	1653490
7.5	3414450	0	0	0	0	0	5192610	1850210
8	2554090	0	0	0	0	0	5218070	1916920
8.5	2643370	0	0	0	0	0	5973550	1992020
9	2978550	0	0	0	0	0	5674220	2408970
9.5	0	0	0	0	0	0	6871000	0
10	3773240	0	0	0	0	0	7477580	0
10.5	0	0	0	0	0	0	8511070	0

Hm0 m	N 0	NE 45	E 90	SE 135	S 180	SW 225	W 270	NW 315
0.5	1247450	1210290	1194700	1235200	1299310	1329060	1307550	1249330
1	1258340	1213980	1186170	1240520		1600830	1350970	1258720
1.5	1339830	1236220	1312590	1254070	1533370	1845510	1449410	1278950
2	1309460	1370010	1487850	1281360	1810990	1974300	1542890	1284230
2.5	1588480	1427210	1490160	1339760	2375780	2723500	1652310	1312730
3	1504520	1590060	2297070	1367390		3000130	1812690	1330260
3.5	1445450	1690150	1518460	1462380		3685890	1925610	1356490
4	1545610	1919460	1745280	1482250	2693010	0	1971290	1415650
4.5	1735420		1945410	1564830	3229380	0	2115930	1437150
5	1865570		2185500	1812380	0	0	2489070	1452690
5.5	1993180	2612440	0	1743640	0	0	2675080	1507760
6	2525900	0	0	0	0	0	3104850	1624270
6.5	2998890	0	0	0	0	0	3709770	1636140
7	3052860	0	0	0	0	0	4001600	1713700
7.5	4269310	0	0	0	0	0	4392320	1972030
8	3417230	0	0	0	0	0	4420080	2015380
8.5	3250500	0	0	0	0	0	5004750	2158890
9	3966110	0	0	0	0	0	4758720	2776440
9.5	0	0	0	0	0	0	5674880	0
10	4933470	0	0	0	0	0	6175580	0
10.5	0	0	0	0	0	0	7009710	0

Figure 5.8: Mooring line 5 (left) and Mooring line 6 (right) for $\varepsilon=0.035$

Hm0 m	N 0	NE 45	E 90	SE 135	S 180	SW 225	W 270	NW 315
0.5	1221510	1224180	1302330	1410560	1318730	1256940	1237760	1211910
1	1225370	1221230	1399230	1469990		1321130	1249790	1233640
1.5	1291270	1222700	1644400	1658800	1662700	1377870	1264480	1286980
2	1258410	1241610	2016090	1927550	2046050	1390860	1283980	1273310
2.5	1470040	1269490	2298070	2553000	2625290	1501780	1308510	1334870
3	1404450	1301120	3359610	2742530		1551720	1359580	1403300
3.5	1414280	1323400	2330470	3407330		1626600	1392730	1498400
4	1437170	1438500	2812720	3212970	3520580	0	1478550	1554900
4.5	1605490		3130280	3985590	3885730	0	1547730	1602110
5	1653320		3683880	5051410	0	0	1634640	1917640
5.5	1834870	1786240	0	5619180	0	0	1673800	2291370
6	2038240	0	0	0	0	0	1992620	2318350
6.5	2376150	0	0	0	0	0	2039060	2486540
7	2825770	0	0	0	0	0	2226370	2949780
7.5	4289160	0	0	0	0	0	2319710	3299380
8	2885770	0	0	0	0	0	2374120	4865740
8.5	3348740	0	0	0	0	0	2892850	5547140
9	3547660	0	0	0	0	0	2854770	6284650
9.5	0	0	0	0	0	0	3555190	0
10	4573540	0	0	0	0	0	3619820	0
10.5	0	0	0	0	0	0	3856120	0

Hm0 m	N 0	NE 45	E 90	SE 135	S 180	SW 225	W 270	NW 315
0.5	1241470	1249300	1319350	1372600	1286150	1236020	1242410	1226960
1	1251620	1274060	1437460	1429450		1256140	1255930	1254670
1.5	1318660	1297710	1751000	1602550	1511620	1301050	1276910	1323260
2	1295970	1345150	2224170	1862060	1798710	1302580	1307190	1312480
2.5	1488050	1397690	2588310	2493460	2239490	1333050	1341260	1388430
3	1448150	1442480	3960700	2689790		1376750	1404820	1465950
3.5	1463650	1495370	2616250	3431640		1389040	1454980	1582670
4	1479390	1628740	3207880	3223970	3130380	0	1562910	1663780
4.5	1684150		3670580	3919660	3393570	0	1658820	1702380
5	1743540		4326700	4819910	0	0	1758270	2057740
5.5	1919350	2164070	0	5351990	0	0	1821020	2435170
6	2092470	0	0	0	0	0	2213640	2500420
6.5	2380400	0	0	0	0	0	2277410	2709350
7	2794130	0	0	0	0	0	2526530	3158700
7.5	3754220	0	0	0	0	0	2649660	3456310
8	2850080	0	0	0	0	0	2724920	5140090
8.5	3223770	0	0	0	0	0	3416620	5898330
9	3481520	0	0	0	0	0	3394410	6640650
9.5	0	0	0	0	0	0	4256520	0
10	4281240	0	0	0	0	0	4454330	0
10.5	0	0	0	0	0	0	4638580	0

Figure 5.9: Mooring line 6 (left) and Mooring line 8 (right) for $\varepsilon=0.035$

In both cases, the mooring system is not influenced by the damping lid factor or the moonpool effects. It is observed that every cell in the tables above and below is green, indicating that the mooring lines successfully meet the ultimate limit state and can withstand all weather conditions.

Hm0 m	N 0	NE 45	E 90	SE 135	S 180	SW 225	W 270	NW 315
0.5	1291550	1297890	1323600	1278260	1227010	1233440	1245990	1235010
1	1340870	1444200	1442510	1284980		1257560	1258550	1239010
1.5	1460830	1625240	1770370	1315370	1268290	1420090	1279830	1257720
2	1549680	1990320	2225150	1349640	1376720	1472720	1309200	1260100
2.5	1929750	2190100	2586530	1428770	1631990	1571790	1346100	1283040
3	1821840	2427010	3975880	1453280		1828060	1408860	1301540
3.5	1972900	2769750	2628640	1659030		1933530	1459110	1338450
4	2181630	3257010	3086210	1598830	1726250	0	1573190	1370690
4.5	2335850		3735210	1743440	1864120	0	1672440	1353230
5	2676260		4288170	1936020	0	0	1767470	1401580
5.5	2842570	5809310	0	1954300	0	0	1830660	1424970
6	3286860	0	0	0	0	0	2221390	1525400
6.5	3983480	0	0	0	0	0	2289720	1514840
7	4048660	0	0	0	0	0	2531730	1637970
7.5	5340220	0	0	0	0	0	2643440	1811830
8	4912030	0	0	0	0	0	2739000	1858330
8.5	4957390	0	0	0	0	0	3385600	2141550
9	5643360	0	0	0	0	0	3410880	2438710
9.5	0	0	0	0	0	0	4264790	0
10	6351720	0	0	0	0	0	4471280	0
10.5	0	0	0	0	0	0	4635580	0

Hm0 m	N 0	NE 45	E 90	SE 135	S 180	SW 225	W 270	NW 315
0.5	1308900	1303800	1303330	1211720	1192560	1214780	1234790	1247100
1	1371720	1469270	1402580	1214450		1222350	1245890	1259750
1.5	1546510	1659770	1666400	1222400	1179020	1366250	1261800	1296510
2	1649460	2036120	2019580	1240670	1285390	1426520	1280440	1335320
2.5	2153400	2262720	2298480	1301700	1526150	1477350	1306310	1389150
3	2014440	2479240	3349250	1321890		1742160	1356040	1425000
3.5	2201940	2831800	2337190	1435610		1827120	1392490	1465280
4	2488750	3284660	2717080	1446430	1636010	0	1487340	1564410
4.5	2653830		3265060	1516800	1776740	0	1559550	1570410
5	3111540		3672010	1793370	0	0	1642860	1630760
5.5	3365140	5961170	0	1722920	0	0	1699900	1769990
6	3681540	0	0	0	0	0	2011240	1886590
6.5	4589110	0	0	0	0	0	2061070	1850940
7	4915720	0	0	0	0	0	2326980	1995470
7.5	6345370	0	0	0	0	0	2324450	2091810
8	5994450	0	0	0	0	0	2388360	2235880
8.5	6095990	0	0	0	0	0	2889540	2295350
9	7009590	0	0	0	0	0	2881390	2556400
9.5	0	0	0	0	0	0	3564540	0
10	7769710	0	0	0	0	0	3643910	0
10.5	0	0	0	0	0	0	3859190	0

Figure 5.10: Mooring line 1 (left) and Mooring line 2 (right) for $\varepsilon=0.1$

Hm0 m	N 0	NE 45	E 90	SE 135	S 180	SW 225	W 270	NW 315
0.5	1304290	1243490	1199400	1145340	1225480	1255080	1299130	1319130
1	1375480	1249650	1187170	1161670	0	1286010	1359330	1397340
1.5	1469210	1258800	1236840	1206940	1276010	1306700	1455140	1558930
2	1549480	1299170	1298940	1264050	1370890	1326050	1532230	1751150
2.5	1723850	1336670	1437850	1405400	1565120	1390930	1654670	1962520
3	1893850	1429150	1494140	1566090	0	1423980	1829450	2188440
3.5	1935920	1504400	1462560	1740350	0	1489930	1903550	2465120
4	2119290	1637110	1743170	1937560	2656170	0	1984250	2737780
4.5	2301040	0	1918510	2639760	2659040	0	2172370	3112930
5	2775540	0	2379780	2715560	0	0	2522950	3699260
5.5	3026040	2324260	0	3056500	0	0	2775830	4380090
6	3163790	0	0	0	0	0	3438660	5057110
6.5	3873640	0	0	0	0	0	3572740	4533370
7	4758930	0	0	0	0	0	3880630	5228910
7.5	5333170	0	0	0	0	0	4225580	5394440
8	4958090	0	0	0	0	0	4249330	8222220
8.5	5192360	0	0	0	0	0	4834930	9754850
9	6266190	0	0	0	0	0	4930510	10837300
9.5	0	0	0	0	0	0	5490370	0
10	7115310	0	0	0	0	0	6338250	0
10.5	0	0	0	0	0	0	7901450	0

Hm0 m	N 0	NE 45	E 90	SE 135	S 180	SW 225	W 270	NW 315
0.5	1302600	1241300	1185920	1106420	1200710	1247460	1298100	1323730
1	1366330	1252460	1172590	1119960	0	1278520	1367200	1405640
1.5	1450540	1267750	1232940	1141540	1192010	1307340	1482110	1577180
2	1534690	1314520	1307230	1173050	1265440	1359900	1576410	1787040
2.5	1689280	1322360	1473680	1284170	1359730	1419330	1723180	2007150
3	1847400	1425260	1546960	1416970	0	1457130	1945590	2246740
3.5	1891160	1511620	1511860	1537000	0	1518900	2031020	2544060
4	2038830	1706090	1831200	1728340	2347620	0	2152040	2823590
4.5	2137730	0	2058110	2422110	2215680	0	2401110	3223910
5	2523240	0	2659630	2534160	0	0	2846650	3840620
5.5	2863800	2636910	0	2803360	0	0	3192060	4480110
6	3082960	0	0	0	0	0	4024030	5149750
6.5	3256440	0	0	0	0	0	4186330	4689190
7	4155680	0	0	0	0	0	4586780	5292680
7.5	4668600	0	0	0	0	0	5013340	5956210
8	4287740	0	0	0	0	0	5027650	8251590
8.5	4570210	0	0	0	0	0	5729860	9831170
9	5514270	0	0	0	0	0	5902520	10900500
9.5	0	0	0	0	0	0	6614670	0
10	6273330	0	0	0	0	0	7677050	0
10.5	0	0	0	0	0	0	9599860	0

Figure 5.11: Mooring line 3 (left) and Mooring line 4 (right) for $\varepsilon=0.1$

Hm0 m	N 0	NE 45	E 90	SE 135	S 180	SW 225	W 270	NW 315
0.5	1227920	1193630	1184460	1247190	1309740	1339840	1302660	1245670
1	1237040	1170300	1169600	1250490	0	1617360	1371430	1259870
1.5	1264500	1173570	1227300	1270450	1561070	1747540	1487620	1279840
2	1257000	1231420	1303610	1299640	1786780	2010130	1581690	1299490
2.5	1306890	1281390	1434070	1373540	2173320	2854310	1727640	1323010
3	1310240	1472820	1500160	1462810	0	3135820	1951760	1355490
3.5	1323550	1614830	1677810	1495290	0	3722540	2038920	1398650
4	1346470	1830730	1867070	1640960	2884880	0	2172630	1401470
4.5	1499660	0	2211330	1825480	3068650	0	2426390	1448770
5	1562240	0	2571730	2326070	0	0	2867240	1478910
5.5	1780700	2355780	0	2647290	0	0	3217120	1481270
6	2008100	0	0	0	0	0	4053420	1510250
6.5	2244180	0	0	0	0	0	4207630	1615620
7	2269590	0	0	0	0	0	4593340	1711590
7.5	2483960	0	0	0	0	0	5019930	1910410
8	2328210	0	0	0	0	0	5039650	1990710
8.5	2759870	0	0	0	0	0	5696060	2185870
9	2663420	0	0	0	0	0	5897590	2437960
9.5	0	0	0	0	0	0	6606140	0
10	3456010	0	0	0	0	0	7657260	0
10.5	0	0	0	0	0	0	9612800	0

Hm0 m	N 0	NE 45	E 90	SE 135	S 180	SW 225	W 270	NW 315
0.5	1246560	1209360	1195470	1235110	1298390	1328850	1295760	1250000
1	1264990	1205820	1182790	1240270	0	1586060	1354280	1263060
1.5	1316580	1230030	1231870	1256190	1536870	1712750	1449310	1273860
2	1317720	1311160	1295500	1284540	1790490	1959870	1527660	1298460
2.5	1390350	1396210	1398280	1383510	2250960	2757760	1648670	1320420
3	1418650	1606050	1467840	1407960	0	3059670	1827950	1340480
3.5	1435160	1787250	1633540	1475150	0	3581450	1906840	1369090
4	1492470	2007750	1734210	1618130	3113340	0	1996130	1435950
4.5	1716350	0	2014530	1779260	3326260	0	2192000	1427580
5	1840440	0	2304010	2169580	0	0	2540900	1476950
5.5	2150290	2752030	0	2271690	0	0	2790520	1495560
6	2510790	0	0	0	0	0	3457490	1594290
6.5	2872570	0	0	0	0	0	3595820	1686880
7	2909460	0	0	0	0	0	3892070	1774400
7.5	3242400	0	0	0	0	0	4244050	1926040
8	3030330	0	0	0	0	0	4266640	2064920
8.5	3370670	0	0	0	0	0	4791940	2355560
9	3455780	0	0	0	0	0	4929530	2709070
9.5	0	0	0	0	0	0	5476920	0
10	4333830	0	0	0	0	0	6317050	0
10.5	0	0	0	0	0	0	7911750	0

Figure 5.12: Mooring line 5 (left) and Mooring line 6 (right) for $\varepsilon=0.1$

Hm0 m	N	NE	E	SE	S	SW	W	NW
	0	45	90	135	180	225	270	315
0.5	1221040	1223970	1303600	1411030	1317870	1257230	1227880	1205850
1	1229730	1219140	1393340	1467930	0	1328230	1255380	1231900
1.5	1269410	1222910	1539230	1667070	1661850	1345800	1258690	1249320
2	1257370	1255860	1781450	1936110	1992230	1382040	1282910	1270170
2.5	1298330	1287040	2100300	2349560	2555010	1503560	1317720	1310850
3	1356940	1394580	2271930	2736960	0	1549320	1359430	1406390
3.5	1374230	1440830	2444470	2935940	0	1691280	1397270	1455340
4	1403930	1594330	2675750	3294770	3713000	0	1503350	1521060
4.5	1572850	0	3067730	4381770	3962760	0	1540760	1566170
5	1633370	0	3755540	5112240	0	0	1701550	1851380
5.5	1734080	2261810	0	5496200	0	0	1767730	2126520
6	2016120	0	0	0	0	0	2022350	2282430
6.5	2365320	0	0	0	0	0	2050610	2431830
7	2734580	0	0	0	0	0	2157810	2954120
7.5	3040230	0	0	0	0	0	2236760	2920940
8	2768390	0	0	0	0	0	2279210	4458800
8.5	3117690	0	0	0	0	0	2765070	5316810
9	3375130	0	0	0	0	0	2981610	5932970
9.5	0	0	0	0	0	0	3361530	0
10	4038100	0	0	0	0	0	3491690	0
10.5	0	0	0	0	0	0	3849380	0

Hm0 m	N	NE	E	SE	S	SW	W	NW
	0	45	90	135	180	225	270	315
0.5	1241850	1249110	1320860	1373080	1285410	1236180	1228890	1221080
1	1257770	1274190	1430120	1427030	0	1264730	1263270	1254320
1.5	1305820	1301890	1609390	1610330	1522950	1277390	1271400	1280280
2	1296940	1359870	1913860	1881800	1761430	1303470	1305980	1311090
2.5	1349050	1413080	2318860	2292920	2213110	1358940	1348930	1375030
3	1417020	1546000	2492320	2676570	0	1397460	1411140	1462810
3.5	1423670	1686020	2686610	2886540	0	1501230	1462700	1548280
4	1452590	1872570	3075010	3229900	3305470	0	1593270	1598950
4.5	1657370	0	3617130	4183860	3473480	0	1648460	1688350
5	1712790	0	4391670	4875060	0	0	1860080	1990310
5.5	1798390	2962440	0	5277460	0	0	1913880	2272740
6	2059810	0	0	0	0	0	2252830	2431580
6.5	2379590	0	0	0	0	0	2307490	2673250
7	2730850	0	0	0	0	0	2439560	3144120
7.5	3017400	0	0	0	0	0	2576000	3163800
8	2741240	0	0	0	0	0	2631280	4775890
8.5	3024810	0	0	0	0	0	3243990	5476030
9	3188260	0	0	0	0	0	3562480	6128970
9.5	0	0	0	0	0	0	3978990	0
10	3844240	0	0	0	0	0	4298190	0
10.5	0	0	0	0	0	0	4741730	0

Figure 5.13: Mooring line 6 (left) and Mooring line 8 (right) for $\varepsilon=0.1$

The following tables present the tilt motion for both scenarios. Notably, differences in the results are observed, particularly in the North direction at a wave height of 4 m and in the Northwest direction at a wave height of 5 m. In the first scenario, which incorporates a small damping lid factor, the results fail to meet the limit state defined by DNV standards. However, in the second scenario, where a higher damping lid factor is applied, the limit state is successfully met.

Hm0 m	N	NE	E	SE	S	SW	W	NW
	0	45	90	135	180	225	270	315
0.5	0.941046	1.150239	0.97966	0.923671	0.116878	1.751785	1.756087	1.688317
1	1.999394	1.330931	0.933761	0.935228	0	3.456187	2.530262	2.333903
1.5	3.903264	1.450728	1.634333	0.966958	0.613913	4.966816	3.233917	3.478328
2	4.66214	2.381704	2.290639	1.156979	1.925648	5.426397	4.007043	4.303555
2.5	7.51064	3.600744	2.846323	1.446753	5.215709	8.162982	4.865609	5.165406
3	8.221148	4.333973	5.03006	1.904655	0	9.042589	5.645943	6.154201
3.5	8.392159	5.188425	3.728981	2.903945	0	9.872909	6.357024	7.114091
4	10.11747	6.110179	4.08174	2.635944	7.455344	0	6.942186	8.080286
4.5	11.38369	0	4.642883	2.603933	9.542841	0	7.584873	9.006963
5	12.89266	0	5.083127	3.656428	0	0	8.408098	10.08817
5.5	14.25897	9.805997	0	4.045162	0	0	8.957794	10.91913
6	15.21598	0	0	0	0	0	9.52359	11.99872
6.5	16.27206	0	0	0	0	0	10.09454	12.94507
7	17.2611	0	0	0	0	0	10.86345	14.04219
7.5	20.83406	0	0	0	0	0	11.41779	15.02794
8	18.93087	0	0	0	0	0	12.20114	15.96956
8.5	19.60948	0	0	0	0	0	12.94867	16.19362
9	21.0126	0	0	0	0	0	13.26747	17.73601
9.5	0	0	0	0	0	0	14.1532	0
10	23.31497	0	0	0	0	0	14.27146	0
10.5	0	0	0	0	0	0	14.38537	0

Hm0 m	N	NE	E	SE	S	SW	W	NW
	0	45	90	135	180	225	270	315
0.5	0.845727	1.142786	0.978119	0.920989	0.101351	1.763617	1.643277	1.644068
1	1.892669	1.188289	0.90815	0.929892	0	3.453961	2.436537	2.282522
1.5	3.21725	1.207511	1.007094	0.94752	0.415598	4.477893	3.217651	3.147091
2	4.624623	1.457874	1.179989	1.177014	0.978874	5.379608	4.007055	4.129592
2.5	5.935562	2.625279	1.90874	1.495437	3.09949	8.146749	4.768871	5.178069
3	7.212651	3.86806	2.58962	2.433089	0	9.090486	5.526109	6.166144
3.5	8.479598	4.761044	3.101105	3.646344	0	9.804044	6.249709	7.116255
4	9.822405	5.961384	3.63968	4.874599	6.223525	0	6.937488	8.089307
4.5	11.10239	0	4.620933	5.835783	8.339802	0	7.560465	9.051205
5	12.39163	0	5.045512	7.348005	0	0	8.311498	9.952379
5.5	13.76749	9.702396	0	8.7054	0	0	8.946507	10.895
6	14.92584	0	0	0	0	0	9.559285	11.85738
6.5	16.03039	0	0	0	0	0	10.051	12.94431
7	16.61944	0	0	0	0	0	10.76507	13.97427
7.5	17.85216	0	0	0	0	0	11.49628	14.89851
8	18.47949	0	0	0	0	0	12.2181	15.583
8.5	19.71943	0	0	0	0	0	12.84434	16.11743
9	20.65731	0	0	0	0	0	13.30064	17.10166
9.5	0	0	0	0	0	0	14.21222	0
10	23.43197	0	0	0	0	0	14.45091	0
10.5	0	0	0	0	0	0	14.65368	0

Figure 5.14: Tilt motion at $\varepsilon=0.035$ (left) Tilt motion at $\varepsilon=0.1$ (right)

The following figures illustrate the acceleration at the top of the tower for each wave height, peak period, wind speed, and direction at this location for both damping lid scenarios. A noticeable difference is observed in the East and South directions at a wave height of 2.5 m, as well as in the West and Northwest directions at a wave height of 4.5 m. In these cases, the results fail to meet the limit state with a lower damping lid factor but successfully pass with a higher damping lid factor.

Hm0 m	N	NE	E	SE	S	SW	W	NW		Hm0 m	N	NE	E	SE	S	SW	W	NW	
	0	45	90	135	180	225	270	315			0	45	90	135	180	225	270	315	
0.5	0.508632	0.064571	0.116757	0.014003	0.225468	0.367735	0.599271	0.367721		0.5	0.471774	0.056164	0.115916	0.013623	0.240455	0.347918	0.524219	0.358682	
1	1.00924	0.361835	0.437608	0.098418	0	1.02337	1.01249	0.743785		1	0.972152	0.315135	0.438852	0.102175	0	0.99301	0.917732	0.690103	
1.5	1.61703	0.553233	1.23257	0.308689	0.79246	1.550457	1.3377	1.221194		1.5	1.30637	0.471685	0.838579	0.250362	0.792016	1.261714	1.29808	1.028255	
2	1.78537	1.135214	2.1188	0.791368	1.48461	1.356875	1.59928	1.391082		2	1.72342	0.825764	1.34239	0.618185	1.32302	1.400776	1.60743	1.327852	
2.5	2.83822	1.9857	3.1896	1.315838	3.13231	2.521635	1.80982	1.711542		2.5	2.03831	1.490705	2.32117	0.999125	2.57165	2.416173	1.80814	1.686753	
3	2.61462	2.241475	5.5676	2.249975	0	2.641256	2.06451	1.96815		3	2.33544	2.043527	3.08894	1.617899	0	2.697043	1.99129	1.969253	
3.5	2.52884	2.624336	4.03585	3.427329	0	3.242078	2.27497	2.304596		3.5	2.56189	2.413836	3.24031	2.220994	0	3.076975	2.27144	2.282495	
4	2.88684	2.738203	4.59718	3.115113	4.70997	0	2.7086	2.6191		4	2.83603	2.778325	3.68844	2.907265	4.22147	0	2.54985	2.583381	
4.5	3.06885	0	4.52529	3.302427	4.76686	0	2.76602	2.914649		4.5	3.06395	0	4.35992	3.297725	5.00144	0	2.71953	2.776587	
5	3.60114	0	5.06894	3.981702	0	0	3.13414	2.955723		5	3.41329	0	4.88598	3.975196	0	0	2.89661	2.903981	
5.5	3.98896	4.176713	0	4.573491	0	0	3.22797	3.298935		5.5	3.85283	3.965616	0	4.582937	0	0	3.22679	3.15981	
6	3.89318	0	0	0	0	0	3.61319	3.489559		6	3.79765	0	0	0	0	0	3.37824	3.487658	
6.5	4.21444	0	0	0	0	0	3.64975	3.756113		6.5	4.06128	0	0	0	0	0	3.4055	3.818744	
7	4.14136	0	0	0	0	0	3.82241	4.181872		7	4.03506	0	0	0	0	0	3.59494	4.062422	
7.5	5.68507	0	0	0	0	0	3.93121	4.288324		7.5	4.43957	0	0	0	0	0	3.79208	4.238775	
8	4.36395	0	0	0	0	0	4.06369	4.312573		8	4.5949	0	0	0	0	0	4.0124	4.236017	
8.5	4.26761	0	0	0	0	0	4.25355	4.303976		8.5	4.25513	0	0	0	0	0	4.08833	4.378162	
9	5.31963	0	0	0	0	0	4.24988	4.939356		9	5.56955	0	0	0	0	0	4.38845	4.854074	
9.5	0	0	0	0	0	0	4.58613	0		9.5	0	0	0	0	0	0	4.78391	0	
10	5.42783	0	0	0	0	0	5.29675	0		10	5.45296	0	0	0	0	0	5.09701	0	
10.5	0	0	0	0	0	0	4.55618	0		10.5	0	0	0	0	0	0	4.67746	0	

Figure 5.15: Tower top acceleration at $\varepsilon=0.035$ (left) Tower top acceleration at $\varepsilon=0.1$ (right)

Last but not least, comparing the tilt motion with accelerations on tower top of wind turbine is noted that the dominant factor for serviceable limit state for this FOWT is the acceleration and not the tilt of the structure.

By translating the probability of occurrences into operational days throughout the year, the maximum number of days in which the wind turbine can generate electricity is calculated and presented in the following figure for both two scenarios.

Hm0 m	N	NE	E	SE	S	SW	W	NW	All	Hm0 m	N	NE	E	SE	S	SW	W	NW	All
	0	45	90	135	180	225	270	315	classes		0	45	90	135	180	225	270	315	classes
0.5	1.447461	0.06617	0.099254	0.016542	0.008271	0.041356	10.46307	1.323393	13.46552	0.5	1.447461	0.06617	0.099254	0.016542	0.008271	0.041356	10.46307	1.323393	13.46552
1	10.30592	0.363933	1.033901	0.636883	0	0.256407	43.6885	4.656688	60.94224	1	10.30592	0.363933	1.033901	0.636883	0	0.256407	43.6885	4.656688	60.94224
1.5	13.50688	0.355662	0.95946	0.802307	0.057898	0.231594	45.72322	6.062793	67.69981	1.5	13.50688	0.355662	0.95946	0.802307	0.057898	0.231594	45.72322	6.062793	67.69981
2	12.92789	0.330848	0.595527	0.578984	0.041356	0.173695	39.07317	4.66496	58.38644	2	12.92789	0.330848	0.595527	0.578984	0.041356	0.173695	39.07317	4.66496	58.38644
2.5	9.726937	0.281221	0.595527	0.529357	0.016542	0.06617	31.08319	4.176958	46.4759	2.5	9.726937	0.281221	0.595527	0.529357	0.016542	0.06617	31.08319	4.176958	46.4759
3	6.65832	0.215051	0.248136	0.587256	0	0.041356	25.24372	3.391194	36.38503	3	6.65832	0.215051	0.248136	0.587256	0	0.041356	25.24372	3.391194	36.38503
3.5	4.623604	0.132339	0.173695	0.190238	0	0.008271	19.29672	1.910648	26.33552	3.5	4.623604	0.132339	0.173695	0.190238	0	0.008271	19.29672	1.910648	26.33552
4	3.184414	0.041356	0.049627	0.041356	0.008271	0	12.98579	1.257223	17.58804	4	3.184414	0.041356	0.049627	0.041356	0.008271	0	12.98579	1.257223	17.58804
4.5	2.042988	0	0.024814	0.033085	0.008271	0	9.412631	0.868477	12.39026	4.5	2.042988	0	0.024814	0.033085	0.008271	0	9.412631	0.868477	12.39026
5	1.293038	0	0.008271	0.016542	0	0	7.113236	0.612069	9.040427	5	1.293038	0	0.008271	0.016542	0	0	7.113236	0.612069	9.040427
5.5	0.967731	0.008271	0	0.008271	0	0	4.706316	0.421831	6.11242	5.5	0.967731	0.008271	0	0.008271	0	0	4.706316	0.421831	6.11242
6	0.645154	0	0	0	0	0	3.085159	0.190238	3.920551	6	0.645154	0	0	0	0	0	3.085159	0.190238	3.920551
6.5	0.405289	0	0	0	0	0	1.836207	0.090983	2.33248	6.5	0.405289	0	0	0	0	0	1.836207	0.090983	2.33248
7	0.27295	0	0	0	0	0	1.265494	0.181967	1.720411	7	0.27295	0	0	0	0	0	1.265494	0.181967	1.720411
7.5	0.181967	0	0	0	0	0	0.835392	0.049627	1.066985	7.5	0.181967	0	0	0	0	0	0.835392	0.049627	1.066985
8	0.099254	0	0	0	0	0	0.570713	0.041356	0.711324	8	0.099254	0	0	0	0	0	0.570713	0.041356	0.711324
8.5	0.008271	0	0	0	0	0	0.132339	0.016542	0.157153	8.5	0.008271	0	0	0	0	0	0.132339	0.016542	0.157153
9	0.049627	0	0	0	0	0	0.107526	0.008271	0.165424	9	0.049627	0	0	0	0	0	0.107526	0.008271	0.165424
9.5	0	0	0	0	0	0	0.074441	0	0.074441	9.5	0	0	0	0	0	0	0.074441	0	0.074441
10	0.008271	0	0	0	0	0	0.033085	0	0.041356	10	0.008271	0	0	0	0	0	0.033085	0	0.041356
10.5	0	0	0	0	0	0	0.008271	0	0.008271	10.5	0	0	0	0	0	0	0.008271	0	0.008271
Total	68.35324	1.794851	3.788212	3.440821	0.14061	0.818849	256.7362	29.92522	365	Total	68.35324	1.794851	3.788212	3.440821	0.14061	0.818849	256.7362	29.92522	365

Figure 5.16: Operational days per year at $\varepsilon=0.035$ (left) Operational days per year at $\varepsilon=0.1$ (right)

Summarizing the operational days for each scenario, it was calculated that with a damping lid factor of 0.035, the wind turbine operates for 336 days per year, corresponding to 92.11% annual activity. On the contrary, with a damping lid factor of 0.1, the operational days increase to 345 per year, resulting in a 94.45% annual activity. This indicates a difference of 2.34% in the estimation of operational days when selecting a damping lid factor without conducting CFD simulations or experimental tests.

Chapter 6

Discussion, Conclusions, and Future Directions

6.1 Reflection on Objectives

What is the sensitivity of moonpool characteristics to variations in damping lid factor?

The numerical results presented in Chapter 4.1 clearly demonstrate the high sensitivity of the moonpool's hydrodynamic behavior to variations in the damping lid factor ε . The added mass and radiation damping coefficients exhibit noticeable resonance peaks at frequencies that closely align with the piston and sloshing modes predicted by Molin's theoretical model. These resonance frequencies, particularly around 0.67 rad/s (piston mode) and 0.93 to 2.15 rad/s (sloshing modes), become extremely high when $\varepsilon = 0$, confirming the overestimation typically observed in potential flow simulations that neglect viscous effects. It is obvious in the motion RAOs, especially for the heave mode (Figure 4.17), that the peak shifts from the theoretical piston frequency to approximately 0.825 rad/s, due to the interaction between moonpool water motion and the freely floating barge.

As ε increases, these resonance peaks are progressively damped. The damping lid method effectively smooths out the unrealistic peaks in both hydrodynamic coefficients and motion responses, demonstrating its utility in approximating viscous damping. Figures 4.1 to 4.6 show that added mass values decrease in magnitude with increasing ε , particularly in surge, heave, and pitch. Radiation damping coefficients (Figures 4.9 to 4.14) follow a similar trend, with reduced amplitudes at the resonance frequencies. This damping behavior translates into more stable RAO curves, where excessive motion amplitudes are significantly diminished with higher ε values.

Looking at the curves of wave elevation and hydrodynamic coefficients, is observed that small increases in ε , particularly up to values around 0.030 to 0.035, lead to a large reduction of the resonance effects inside the moonpool. After this range, further increases in the damping lid factor result in smaller changes in the response, indicating that the system becomes less sensitive to ε . This means that applying a damping lid factor larger than 0.030 is enough to significantly decrease the peaks caused by the limitations of potential flow theory, but the realistic value of ε is still unknown.

Overall, the results confirm that the moonpool's dynamic characteristics are strongly

influenced by the damping lid factor, and selecting an appropriate ε value is essential to obtain realistic and physically consistent predictions in potential flow-based simulations.

How do different damping lid factors affect the loads and motions on mooring systems?

Different values of the damping lid factor ε have a clear and measurable impact on the dynamic loads experienced by the mooring system. The influence is stronger at wave frequencies near the internal resonance modes of the moonpool, particularly around the piston mode 0.82 rad/s and the first sloshing mode 1.05 rad/s. At these critical frequencies, low or zero damping leads to exaggerated motion responses in the platform, which in turn amplify the tension forces in the mooring lines.

As the damping lid factor increases, these resonance-driven tensions are significantly reduced. For ε values above 0.030, the peaks in the frequency responses of the mooring line are noticeably suppressed, indicating that the damping lid helps to stabilize the hydrodynamic excitation transferred from the moonpool to the mooring system. This effect was consistently observed in both regular wave conditions and irregular sea states. In irregular waves, despite the broader frequency content, the same resonance modes dominated the response, and the damping lid remained effective at reducing dynamic loading when ε exceeded 0.03.

A focused comparison between three similar damping lid values (0.03, 0.032, and 0.035) showed the ε which damp well the system. When ε was set to 0.03, the platform's heave motion became too large, up to four times the wave height, meaning the damping was too low. At $\varepsilon = 0.032$, the motion became smaller but was still slightly unstable. At $\varepsilon = 0.035$, the platform's motion became stable, and the mooring loads appeared realistic. This shows that a damping value of around 0.035 is the minimum needed to reduce resonance effects and keep the system stable.

What are the responses of the system under operational conditions in real sea states when different damping lid factors are applied?

Once the damping lid factor (ε) reaches the critical threshold that stabilizes the system, which identified as approximately $\varepsilon = 0.035$, the sensitivity of the platform's response to further increases in damping becomes low. The comparison between $\varepsilon = 0.035$ and $\varepsilon = 0.1$ under realistic sea conditions shows that both values result in similar overall system performance, particularly in terms of mooring line tensions, tilt motion, and tower-top accelerations.

Although $\varepsilon = 0.1$ slightly improves motion performance and keeps more pitch and acceleration values within design limits, the improvements are small. For example, the annual operational availability increases from 336 to 345 days, a difference of only 2.34 %. In practical terms, this means that once the damping lid is sufficient to suppress moonpool-induced resonance, further increases in ε do not substantially change the behavior of the floating platform.

Additionally, the mooring system is not significantly affected by the damping lid in either case, as it remains well within the ultimate limit state for all conditions. The dominant limiting factor for operability is tower-top acceleration rather than tilt.

In conclusion, the system’s response becomes relatively insensitive to the damping lid once ε exceeds 0.035. While using a much higher damping value such as 0.1 can provide additional safety margin, $\varepsilon = 0.035$ already offers a stable solution that satisfies most operational and design criteria. However, still the realistic value is missing and to find the exact value that represents reality, CFD analysis or experimental models are required.

6.2 Conclusions

This thesis investigated the role of the damping lid factor (ε) simulating the behavior of a barge-type Floating Offshore Wind Turbine (FOWT) with a central moonpool, focusing on its impact across hydrodynamic responses, mooring loads, and overall system performance under realistic sea conditions. The study confirmed that moonpool-induced resonances, especially the piston and sloshing modes, are significantly overestimated when viscous effects are not accounted for in potential flow simulations. To overcome this, the damping lid method was applied as a simplified numerical correction. Its effectiveness was validated across multiple wave conditions and system parameters.

In the hydrodynamic analysis, increasing ε led to strong damping of unrealistic resonance peaks in added mass, radiation damping, and motion RAOs. These effects were especially evident near resonance frequencies around 0.825 rad/s and 1.05 rad/s. A damping lid factor of approximately $\varepsilon = 0.035$ was identified as the minimum value needed to stabilize the system. Below this threshold, exaggerated responses persisted, while above it, further improvements were limited.

The response of the mooring system in the regular and irregular sea states showed a similar sensitivity. For $\varepsilon < 0.03$, tension fluctuations were unrealistically high, while values at or above 0.035 brought the system into a stable and physically meaningful range. In the case study, comparing $\varepsilon = 0.035$ and $\varepsilon = 0.1$ under real sea conditions demonstrated only minor differences in tilt, tower-top acceleration, and operability (a 2.34 % increase in annual operability for the higher damping). Importantly, both values satisfied the ultimate limit state requirements for mooring.

In conclusion, the damping lid method provides an efficient and practical approach to incorporate viscous effects in potential flow-based simulations for this class of FOWT. When the goal is to evaluate ultimate limit state behavior or operational criteria, such as mooring loads, tilt limits, or power availability, potential flow models enhanced with a properly tuned damping lid can offer reliable insight without the need for CFD or experimental data.

However, if the objective is to determine the true value of the damping lid factor for a specific barge-moonpool configuration, then CFD simulations or physical experiments are unavoidable. The damping lid, while practical, remains a modeling assumption that must be calibrated externally to reflect real-world fluid behavior.

6.3 Future Work

While this thesis successfully explored the impact of the damping lid factor ε on the performance of a barge-type Floating Offshore Wind Turbine FOWT with a moonpool, several aspects remain open for future investigation.

One direction for further research is to expand the study to alternative floater geometries and moonpool configurations. This work focused on a square-shaped barge with a single central moonpool. Future studies could consider asymmetric hulls, multiple moonpools, or variations in moonpool shape and depth, as these factors may significantly influence internal resonance behavior and damping effectiveness.

Additionally, the environmental loading in this study included regular and irregular waves, as well as steady wind loads. However, current forces were not included, and wind loading was not represented using full wind spectrum or rotor control systems. Incorporating realistic wind spectra, dynamic wind gusts, and turbine control effects could improve the accuracy of the response predictions, particularly for tower-top motions and operational limits under complex metocean conditions.

Another valuable extension would be a deeper investigation into fatigue behavior. This thesis primarily evaluated responses in terms of ultimate limit states and serviceability thresholds. However, the long-term motion patterns and residual excitations, even when partially damped, may contribute to fatigue damage in mooring lines and structural components, which are critical for life-cycle design and maintenance planning.

In summary, future work should aim to generalize the findings across a broader range of structural and environmental conditions and expand the simulation scope to include fatigue, current effects, realistic wind-rotor interaction and a plenty of geometrical configurations in barge and moonpool. This would enhance the applicability and reliability of the damping lid method in practical floating wind design.

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