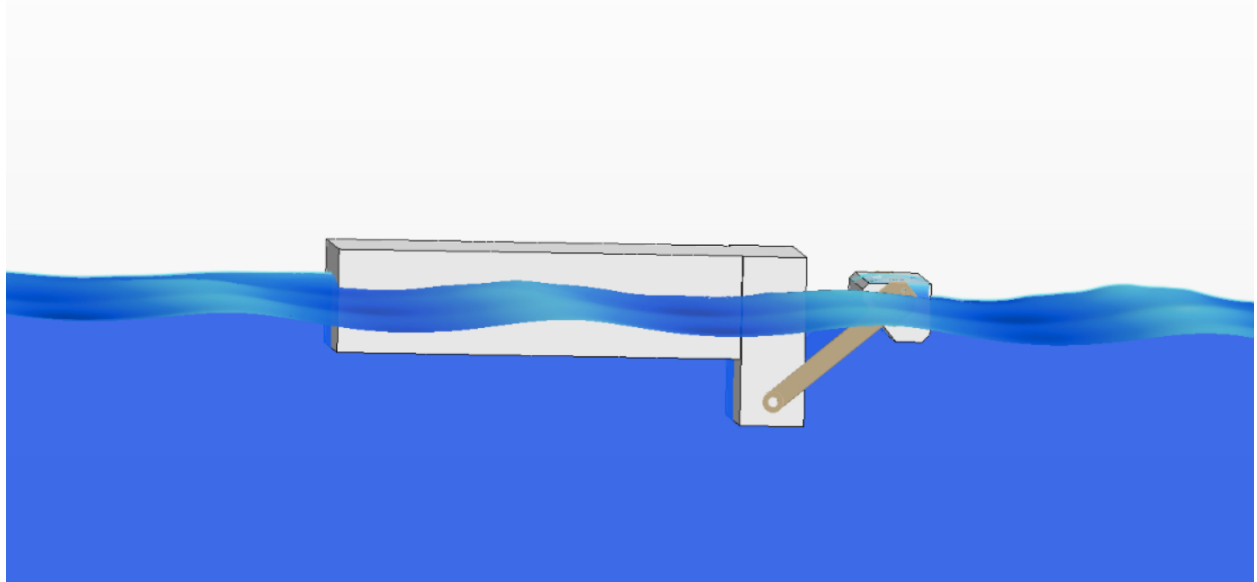




CHALMERS
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CFD Analysis of an Articulated-Body Wave Energy Converter

Comparison with Experimental Data and Frequency-Domain Modeling Predictions

Master's thesis in Applied Mechanics

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CHALMERS UNIVERSITY OF TECHNOLOGY
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MASTER'S THESIS 2026

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Comparison with Experimental Data and Frequency-Domain
Modeling Predictions

EMELIE STEWALL



CHALMERS
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EMELIE STEWALL

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Cover: Visualization of the WEC created in STAR-CCM+, showing the studied geometry within the wave domain.

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Abstract

Developing sustainable energy systems requires continued advancement in renewable energy technologies. Among these, wave energy is considered a particularly promising resource due to its substantial theoretical potential. This thesis investigates an articulated-body wave energy converter using a computational fluid dynamics (CFD) model developed in STAR-CCM+.

To evaluate the predictive capability of the numerical model, previously conducted model tests are replicated under regular wave conditions. Wave periods ranging from 0.7 to 2.2 seconds are considered, with the hinge-angle amplitude as the primary response quantity. This response is particularly important, as it is directly related to the relative motion available for energy extraction.

The simulation results are compared with experimental measurements and results from a previously developed frequency-domain model. The numerical setup includes wave generation, multibody dynamics, and key subsystems such as the mooring and Power Take-Off (PTO) systems. Two PTO formulations are implemented: a linear formulation and a fitted nonlinear formulation based on the measured torque-angular velocity relationship from the model tests.

Overall, the CFD models show close agreement with the experimental data across the complete range of analyzed cases. They capture the experimentally observed peak hinge-angle response and reduce overprediction near the peak compared with the previous frequency-domain model. All CFD models predict the peak response at the same wave period as the experiments and within 10% of the experimental peak amplitude, whereas the frequency-domain model overpredicts the peak by more than 30%. Comparisons between the two PTO representations further indicate that this subsystem significantly influences the predicted response across the investigated cases.

Among the investigated approaches, the CFD model with the fitted nonlinear PTO formulation provides the best agreement with experimental measurements, with a global error of approximately 10%. However, the comparison also shows that the improved agreement cannot be attributed solely to the hydrodynamic model, as the PTO formulation significantly influences the predicted hinge-angle response. These findings indicate that reliable prediction of the WEC response requires not only accurate hydrodynamic modeling but also representative subsystem input data, particularly for the PTO system.

Keywords: Articulated-Body Wave Energy Converter, Computational Fluid Dynamics, Hinge-Angle Response, Power Take-Off System, Wave Energy.

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Emelie Stewall, Gothenburg, June 2026

List of Acronyms

Below is the list of acronyms that have been used throughout this thesis listed in alphabetical order:

AMBWEC	Articulated Multi-Body Wave Energy Converter
BEM	Boundary Element Method
CAD	Computer-Aided Design
CFD	Computational Fluid Dynamics
COG	Center of Gravity
DFBI	Dynamic Fluid-Body Interaction
DOF	Degree of Freedom
HRIC	High-Resolution Interface Capturing
LCOE	Levelized Cost of Energy
NRMSE	Normalized Root Mean Square Error
PTO	Power Take-Off
RANS	Reynolds-Averaged Navier–Stokes
RAO	Response Amplitude Operator
RMSE	Root Mean Square Error
SST	Shear Stress Transport
VOF	Volume of Fluid
WEC	Wave Energy Converter

Nomenclature

Below is the nomenclature of indices, sets, parameters, and variables that have been used throughout this thesis.

Indices

i, j	spatial indices
t	temporal index

Parameters

g	Gravitational acceleration
A	Wave amplitude
H	Wave height
h	Water depth

Variables

T	Wave period
λ	Wavelength
ω	Angular wave frequency
k	Wave number
c	Wave celerity / phase speed
$\eta(x, t)$	Free-surface elevation
ψ	Wave phase
ϵ	Wave steepness
ρ	Fluid density
μ	Dynamic viscosity

ν	Kinematic viscosity
K	Power take-off coefficient
n	Power take-off exponent
ϕ	Velocity potential
u, v, w	Velocity components in the x -, y -, and z -directions
p	Pressure
ξ	Motion response amplitude or displacement
θ	Angular displacement
ω	Angular velocity vector
ω_h	Relative hinge angular velocity
α	Volume fraction
y^+	Dimensionless wall distance
u_τ	Friction velocity
τ_w	Wall shear stress
S_{ij}	Strain-rate tensor
ν_t	Turbulent viscosity
k	Turbulent kinetic energy
ε	Turbulent dissipation rate
ω	Specific dissipation rate

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1

Introduction

Meeting the rising global demand for electricity while reducing greenhouse gas emissions represents one of the major challenges in the coming decades. In 2024, global electricity demand increased by 4.3%, driven by industrial expansion, rapid electrification, the expansion of data centers, and increasing cooling demand associated with higher temperatures [1]. Despite the rapid growth in electricity demand, fossil fuels still accounted for more than half of global electricity generation, and CO₂ emissions continued to rise by 0.8% [2]. In addition, 2024 was the first full year in which the global mean temperature remained more than 1.5 °C above pre-industrial levels [3]. These developments emphasize the urgent need to accelerate the deployment of sustainable energy technologies to meet rising electricity demand while remaining consistent with international climate targets.

Although renewable energy capacity is expanding, it is not growing sufficiently fast to meet international climate goals. Current estimates indicate that global renewable electricity capacity must triple by 2030 to achieve these targets [4]. The current renewable energy mix is dominated by wind, solar, and hydropower, while the ocean is often regarded as one of the largest untapped sources of renewable energy [5]. Wave energy offers several potential advantages over wind and solar energy because it is relatively predictable and has a high energy density [6]. Despite these advantages, most wave energy converters (WECs) are still in the early conceptual or developmental stages. Further research and technological development are therefore required before wave energy can play a more significant role in future energy systems.

To advance the development of new energy technologies, numerical modeling is essential for predicting system performance and supporting design optimization. In a previous study, a frequency-domain model based on a multibody boundary element method was developed to simulate the dynamic behavior of an articulated wave energy converter [7]. However, comparisons with experimental measurements revealed discrepancies between the model predictions and the observed system behavior, likely attributable to limitations in the modeling approach. Therefore, this thesis investigates whether higher-fidelity CFD simulations can improve predictions of the WEC's dynamic response and provide a more reliable basis for its future design and development.

1.1 Background

To provide a foundation for the present study, this section introduces the fundamentals of wave energy and wave energy converters. Particular attention is given to the articulated-body wave energy converter investigated in this thesis. An overview of relevant previous research is also included, as the present study builds upon earlier work in this field.

1.1.1 Wave Energy

Ocean energy can be harvested from several sources, including waves, tidal currents, and ocean currents, as well as temperature and salinity-driven processes, all of which can be used for power generation [8]. Among these resources, wave energy is often considered one of the most promising due to its substantial theoretical potential [5]. The theoretical global wave energy potential has been estimated at approximately 32,000 terawatt-hours per year, which is sufficient to meet current annual global electricity demand [9]. The regions with the most favorable conditions for wave energy extraction are located between 40° and 60° latitude in both hemispheres, as illustrated in Figure 1.1 [10].

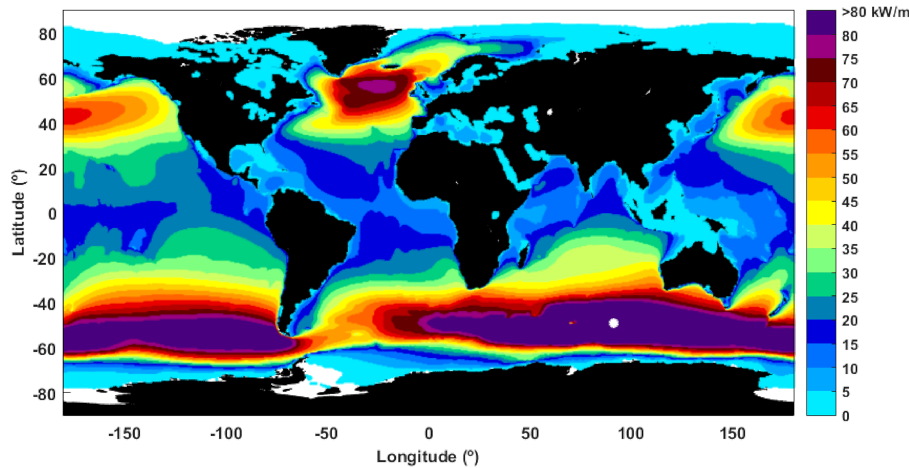


Figure 1.1: Global annual mean wave power production potential over 30 years [10].

Compared with several other renewable energy sources, wave energy is both energy-dense and capable of near-continuous operation. The power density of ocean waves typically ranges from $2\text{--}3\text{ kW/m}^2$, which is approximately ten times greater than that of wind power and nearly one hundred times greater than that of solar power [6]. In addition, wave conditions are relatively predictable, enabling power generation during periods when other renewable resources may be limited due to weather or seasonal variations.

Despite these advantages, wave energy technology remains in its early stages of development, yet it is widely regarded as a promising future energy source [8].

Moreover, the relatively high levelized cost of energy (LCOE), defined as the ratio of the total lifetime cost to total energy produced, is considered one of the main barriers to the large-scale deployment of WECs [11]. As of 2024, the total installed ocean energy capacity is approximately 513 MW [12]. Nevertheless, wave energy technology continues to develop, supported by increasing investment, technological progress, and supportive policy frameworks. For example, the European Union has set targets of achieving 1 GW of wave energy capacity by 2030 and 40 GW by 2050 [13].

1.1.2 Overview of Wave Energy Converters

Wave energy converters generate electricity by converting wave-induced motion into usable electrical power. A wide range of concepts has been proposed, and the industry has not yet converged on a single dominant design [8]. WECs are commonly classified according to their operating principles, deployment location, and directional response to incoming waves [14].

Classification by operating principle typically includes oscillating bodies, oscillating water columns, and overtopping devices [15]. Oscillating bodies extract energy from wave-induced structural motion. In general, an oscillating body WEC consists of several interconnected subsystems: a hydrodynamic subsystem that captures energy from waves, a Power Take-Off (PTO) system that converts mechanical motion into usable power, a mooring system that restrains motion, and a mechanical subsystem that guides the permitted motion [16]. In contrast, oscillating water columns use a chamber in which the oscillating water surface generates varying air pressure that drives a turbine. Overtopping devices operate on different principles, collecting water in an elevated reservoir and generating energy as the stored water is released back into the ocean [15]. These operating principles are illustrated in Figure 1.2.

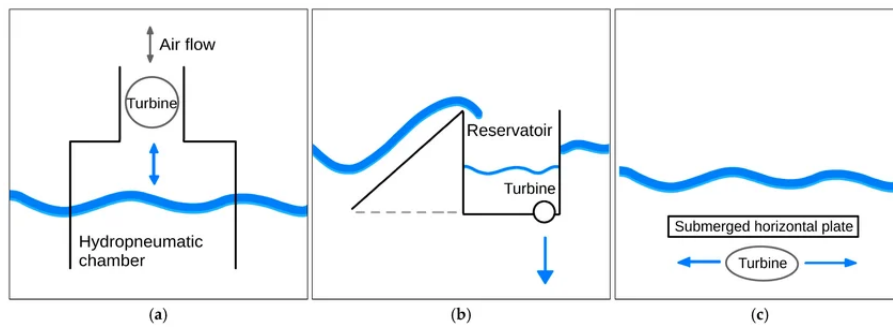


Figure 1.2: WEC configurations by operating principle: (a) oscillating water column, (b) overtopping device, (c) oscillating body [17].

WECs may also be classified by deployment location as onshore, nearshore, or offshore [18]. In addition, they may be categorized by their orientation relative to the incoming wave, typically grouped into point absorbers, attenuators, and terminators [19]. Point absorbers generally consist of a floating or submerged body that extracts energy from waves approaching from all directions. In contrast, terminators are ori-

ented perpendicular to the direction of wave propagation, whereas attenuators are aligned parallel to it. These configurations are illustrated in Figure 1.3.

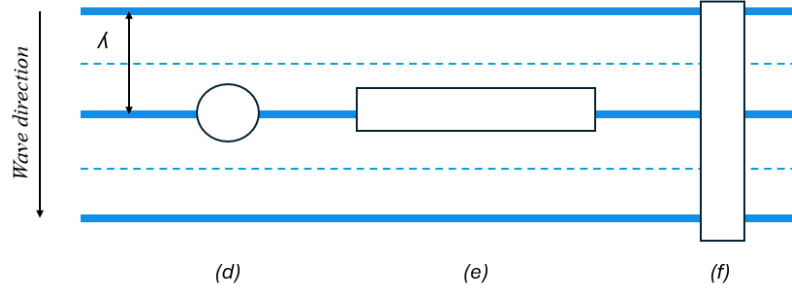


Figure 1.3: WEC configurations by orientation: (d) point absorber, (e) attenuator, (f) terminator.

1.1.3 Articulated-Body Wave Energy Converters

Among the various WEC concepts, articulated wave energy converters are a type of oscillating body device. They consist of two or more bodies connected by joints and extract energy from the relative rotational motion between them [20]. These devices are characterized by structural flexibility and a reconfigurable design.

1.1.4 Numerical Modeling Approaches of WECs

Numerical modeling is essential in WEC development, as it enables evaluation of wave energy converter performance before full-scale testing. Different modeling approaches are used depending on the required level of accuracy, computational cost, and stage of development [21]. These approaches differ primarily in how they account for nonlinear effects, ranging from linear to fully nonlinear models. In WEC analysis, potential flow based models and computational fluid dynamics (CFD) models are two widely used approaches.

Potential flow based frequency-domain models are commonly regarded as lower-fidelity approaches because they are typically based on linear assumptions and the assumption that the flow is incompressible, inviscid, and irrotational. Due to these simplifications, such models are commonly used during the early and intermediate stages of WEC development. However, these assumptions limit their ability to capture viscous effects and strongly nonlinear phenomena, such as slamming and wave breaking.

In contrast, CFD models are regarded as higher-fidelity approaches because they solve the governing flow equations, typically the Navier-Stokes equations, within a discretized fluid domain. These models are generally considered fully nonlinear and include viscous effects. However, the increased fidelity is accompanied by a significantly higher computational cost. Consequently, CFD is often applied in the later stages of WEC development, particularly for survivability analysis.

1.1.5 Previous Research

The WEC concept investigated in this thesis originates from a hinged five-body articulated WEC configuration. The original concept consisted of a central cylindrical body connected to four surrounding bodies that harvest energy from multiple directions, thereby forming a multibody floating system [22]. For modeling and experimental purposes, this configuration was simplified to a model consisting of a single barge and a single absorber body [7]. The simplified two-body configuration is shown in Figure 1.4.

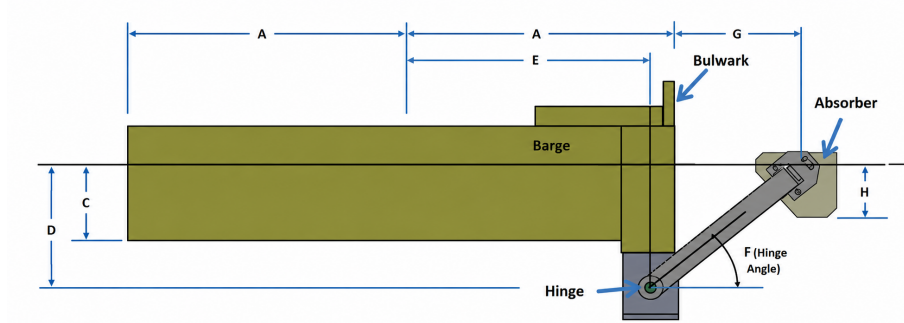


Figure 1.4: Original concept of articulated wave energy converter [7].

Ersdal and Moe (2013) investigated this articulated two-body wave energy converter consisting of a main barge and an absorber body connected by a submerged hinge [7]. Energy is extracted from the relative pitch motion between the absorber and the main body about the hinge. The hinge-angle response is therefore the primary quantity of interest in this study.

The experiments were conducted in a towing tank at the Kelvin Hydrodynamics Laboratory at the University of Strathclyde using a 1:12 scale model of the WEC, illustrated in Figure 1.5. The regular wave tests were performed using a wave amplitude of 0.02 m and wave periods ranging from 0.7 to 2.2 s. The Power Take-Off (PTO) system was represented by a programmable servo motor that applied a braking torque as a function of the hinge angular velocity, corresponding to the linear PTO case, ($n=1$). Although the intended control law was linear with a nominal PTO gain of $K=4$ Nms/rad, the measured torque-velocity relationship was not perfectly linear. This was partly attributed to non-ideal behaviors in the PTO system, including friction in the gearbox and motor. A least-squares fit to the measured data yielded an effective PTO gain of approximately $K=6.7$ Nms/rad. For this reason, both PTO gain factors were considered throughout their analysis.

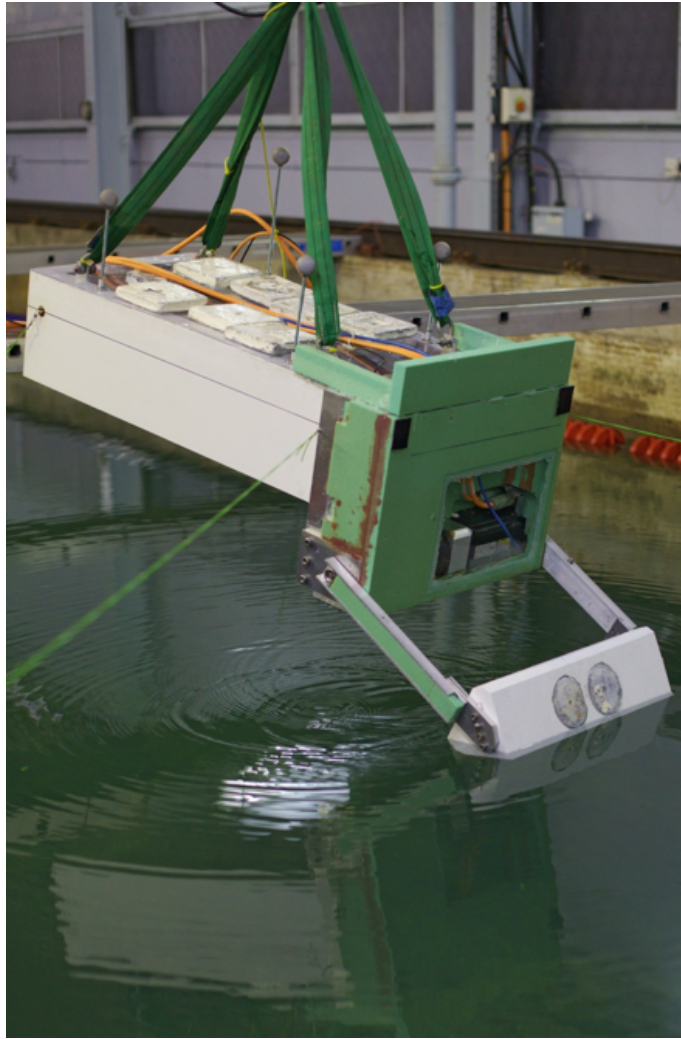


Figure 1.5: The wave energy converter (WEC) used in the experimental study [7].

A frequency-domain model based on a multibody boundary element method was developed to simulate the dynamic behavior of the articulated wave energy converter. The hydrodynamic coefficients were computed using HydroD, a DNV software tool for linear potential flow analysis. Hydrodynamic coupling between the bodies was included, and viscous effects were incorporated as additional damping terms. The equations of motion were solved in MATLAB for the six rigid-body degrees of freedom of the assembly and one additional rotational degree of freedom at the hinge.

The comparison showed that the numerical model captured some of the observed trends in the measured response, but significantly overpredicted the hinge response for certain wave periods, particularly for wave periods exceeding 1.2 s, as shown in Figure 1.6.

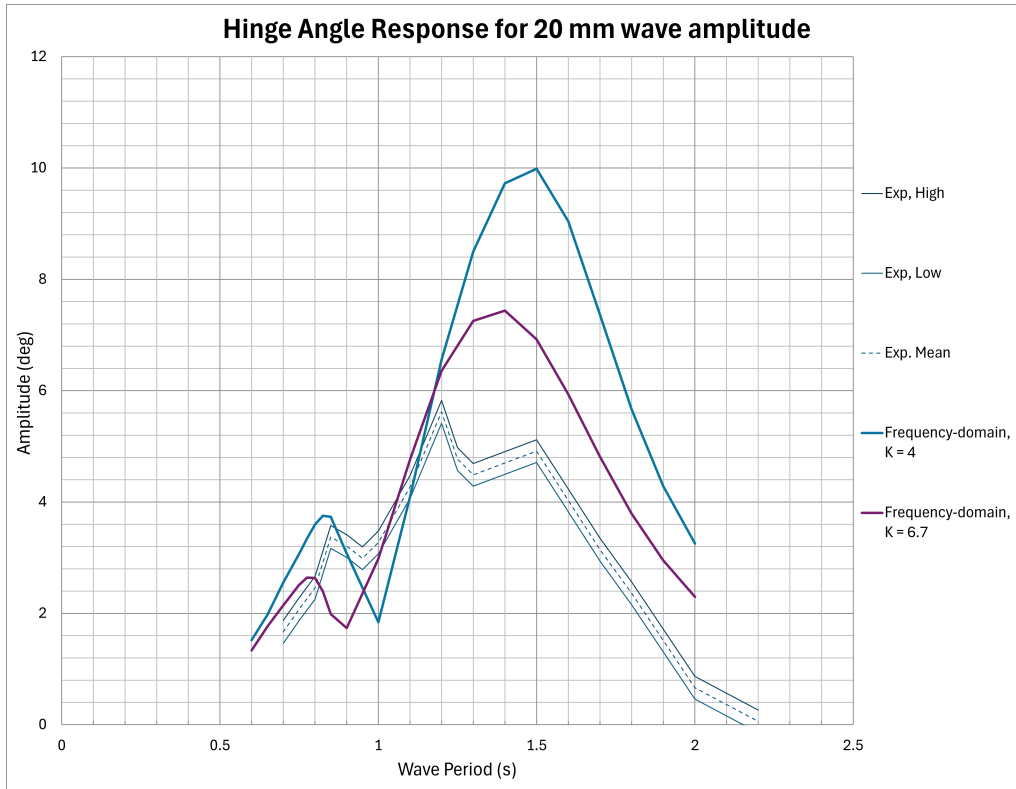


Figure 1.6: Comparison of hinge-angle amplitude responses from the frequency-domain model and experimental measurements [7].

In Figure 1.6, the high, mean, and low curves represent the experimental results, while the curves labeled $K = 4$ and $K = 6.7$ represent frequency-domain model predictions obtained using two different PTO gain factors.

The experimental results also indicated weaker interaction between the bodies than predicted numerically. The authors suggested that the discrepancies may be related to nonlinear viscous effects, green water phenomena on the absorber, and submergence effects. The results further suggest that the tank walls significantly affected the measurements, representing a potential source of error and contributing to the differences between the experimental data and the frequency-domain model predictions. To investigate the latter, mirror bodies were placed on both sides of the original position. This improved the agreement for shorter wave periods to some extent, but the discrepancies were not fully resolved.

In addition to the regular wave response analysis, the study investigated power performance under irregular waves and examined how different PTO characteristics affect energy capture and peak power levels.

1.2 Problem Statement

The discrepancies between the experimental measurements and the frequency-domain model motivate further investigation into whether a higher-fidelity numerical ap-

proach can provide a more accurate prediction of the dynamic response of the articulated-body wave energy converter. Since frequency-domain models rely on simplifying assumptions, they may be limited in their ability to capture nonlinear hydrodynamic effects, viscous damping, and changes in wetted area. These effects may contribute to the discrepancies observed in the previous study, particularly around the peak hinge-angle response. This thesis, conducted in collaboration with Entr, part of Aker Solutions, therefore examines whether a CFD approach can improve the predictive capability of the existing modeling framework.

1.3 Objective and Research Questions

The objective of this thesis is to develop and evaluate a computational fluid dynamics (CFD) model of the articulated-body wave energy converter operating in regular waves, with a primary focus on predicting the relative hinge-angle response between the connected bodies. Accordingly, this thesis addresses the following research questions:

- To what extent can the CFD model reproduce the experimentally measured hinge-angle response across the investigated wave periods?
- How does the performance of the CFD model compare with the existing frequency-domain model in predicting the hinge-angle response?

1.4 Limitations

This study is subject to the following limitations:

- Only regular wave conditions are considered.
- The analysis focuses primarily on reproducing the experimentally measured hinge-angle results. Absorbed power is outside the scope of this thesis, while the overall motion of the structure is considered only to a limited extent and is not analyzed in detail.
- The study is limited to a hydrodynamic analysis, and environmental effects beyond wave structure interaction are therefore neglected.
- No specific deployment site is considered. Instead, the input parameters are chosen to reproduce the controlled wave environment, the scaled prototype geometry, and the model test conditions used in the laboratory experiments.

2

Theoretical Background

This chapter presents the theoretical background for wave hydrodynamics and floating-body response. It also introduces the fundamental principles of frequency-domain modeling and computational fluid dynamics. Together, these topics provide the theoretical foundation for understanding the WEC response, as well as the numerical modeling approaches used in the previous study and in the present thesis.

2.1 Fundamentals of Wave Theory

Waves can be described using several theoretical frameworks depending on the underlying assumptions and the required level of accuracy. This section introduces the fundamental characteristics of waves and provides an overview of linear and nonlinear wave theory.

2.1.1 Wave Characteristics

Waves are commonly described by a set of characteristic parameters and may be classified as either regular or irregular waves [23]. A regular wave is an idealized wave that is periodic, unidirectional, and characterized by constant amplitude and period. In contrast, irregular waves are composed of multiple wave components with varying amplitudes, periods, and propagation directions.

A regular wave can be described using both temporal and spatial quantities [24]. In the temporal domain, the wave period, T , is defined as the time required for one complete wave cycle to pass a fixed point. The corresponding angular frequency, ω , represents the temporal rate of oscillation. In the spatial domain, the wavelength, λ , is the horizontal distance between two successive points with the same phase. The corresponding wave number, k , describes the spatial frequency of the wave and is defined in Equation 2.1.

$$k = \frac{2\pi}{\lambda} \quad (2.1)$$

Furthermore, the wave celerity, c , describes the propagation speed of a wave phase. For a regular wave, the celerity is given by Equation 2.2.

$$c = \frac{\omega}{k} = \frac{\lambda}{T} \quad (2.2)$$

The main geometric wave parameters are illustrated in Figure 2.1.

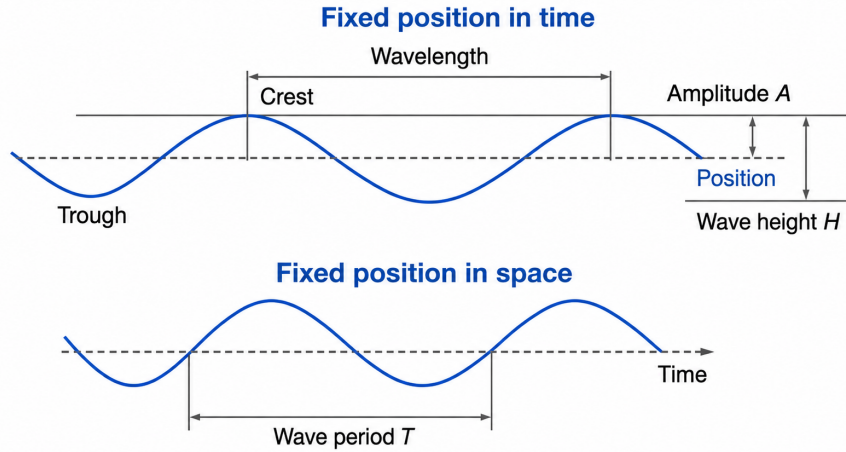


Figure 2.1: Wave characteristics [25].

These parameters include the crest, the highest point of the wave, and the trough, the lowest point. The wave height, H , is the vertical distance between the crest and trough, whereas the amplitude, A , is the vertical distance from the mean free surface to the crest. For a regular sinusoidal wave, the amplitude is therefore equal to half of the wave height.

2.1.2 Linear Wave Theory

Linear wave theory, also referred to as Airy wave theory, is a first-order approximation used to describe surface gravity waves and forms the basis for more advanced wave models [26]. The theory assumes that the wave amplitude is small relative to both the wavelength and the water depth. Under this small-amplitude assumption, nonlinear effects such as wave asymmetry, wave breaking, and higher-order harmonics can be neglected, thereby allowing the free surface boundary condition to be linearized [27].

The theory is based on the assumptions that the fluid is inviscid, incompressible, and irrotational [27]. As a result, the velocity field can be described using a velocity potential ϕ . For an incompressible fluid, conservation of mass gives Laplace's equation for the velocity potential shown in Equation 2.3.

$$\nabla^2 \phi = 0 \quad (2.3)$$

The free surface elevation of a regular wave propagating in the positive x -direction can be expressed as Equation 2.4.

$$\eta(x, t) = \frac{H}{2} \cos(kx - \omega t + \psi) \quad (2.4)$$

where η is the free surface elevation, H is the wave height, k is the wave number, ω is the angular frequency and ψ is the phase angle. By applying the linearized free surface boundary conditions, the dispersion relation is obtained, as shown in Equation 2.5.

$$\omega^2 = gk \tanh(kh) \quad (2.5)$$

Here, h is the water depth and g is the gravitational acceleration. This relation describes the dependence between wave frequency, wavelength, and water depth.

2.1.3 Nonlinear Wave Theory

Nonlinear wave theory becomes necessary as wave steepness increases and the assumptions of linear wave theory no longer hold [27]. In such cases, the wave profile may no longer be symmetric and sinusoidal. Higher-order wave theories can therefore be used to describe regular waves more accurately by accounting for nonlinear features, including sharper crests and flatter troughs, as well as the dependence of wave speed on wave amplitude and wavelength.

Stokes wave theory is a perturbation theory for periodic gravity waves, in which the solution is expanded in powers of wave steepness [27]. The wave steepness, ε , is a dimensionless parameter that describes the relative steepness of the wave and is defined as shown in Equation 2.6.

$$\varepsilon = kA \quad (2.6)$$

where k is the wave number and A is the wave amplitude. Moreover, the free surface elevation can be expressed as a perturbation series with higher harmonic contributions, as described in Equation 2.7.

$$\eta = \varepsilon\eta_1 + \varepsilon^2\eta_2 + \varepsilon^3\eta_3 + \dots \quad (2.7)$$

Here η_1 represents the first-order linear contribution, while the higher-order terms represent nonlinear corrections to the wave profile. In fifth-order Stokes theory, the expansion is retained up to the fifth-order term. Stokes' theory is commonly applied to regular waves in intermediate and deep water, where nonlinear effects are important.

2.2 Dynamics of Floating Structures

To describe the response of the wave energy converter considered in this thesis, it is necessary to introduce the fundamental dynamics of floating structures. These principles are important because the body's response determines the relative motion available for energy extraction. Therefore, this section introduces the main factors governing the dynamics of floating bodies, including hydrodynamic loading and external constraints such as the PTO and mooring systems.

2.2.1 Degrees of Freedom

In a three-dimensional space, a rigid floating structure has six degrees of freedom (6-DOF), which describe the full rigid-body motion of the structure [28]. These include three translational and three rotational components, as illustrated in Figure 2.2.

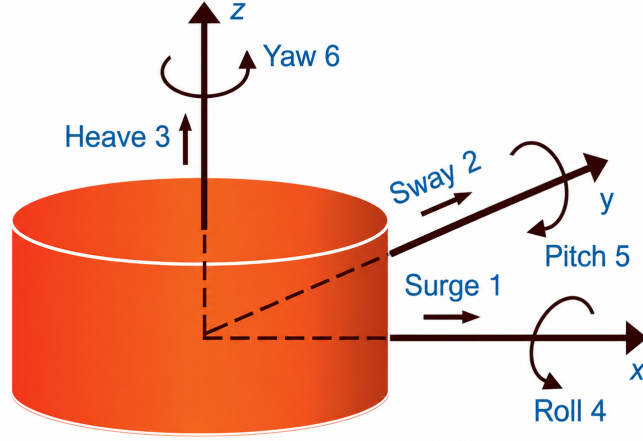


Figure 2.2: Six degrees of freedom of a floating structure [25].

The translational motions are surge along the longitudinal axis, sway along the lateral axis, and heave along the vertical axis. The rotational motions are roll, pitch, and yaw about these axes. A two-body WEC can therefore have up to 12 rigid-body degrees of freedom before constraints are applied.

2.2.2 Equations of Motion

The translational motion of a WEC can be described using rigid-body dynamics of a floating structure [29]. The motion is governed by Newton's second law and can be expressed as Equation 2.8.

$$\mathbf{M} \ddot{\mathbf{x}} = \mathbf{F}_{\text{hydrodynamic}} + \mathbf{F}_{\text{hydrostatic}} + \mathbf{F}_{\text{mooring}} + \mathbf{F}_{\text{PTO}} + \mathbf{F}_{\text{other}} \quad (2.8)$$

Here $\mathbf{M}$ is the mass matrix and $\ddot{\mathbf{x}}$ is the translational acceleration vector. The terms on the right-hand side represent the external force contributions acting on the device, including hydrodynamics, hydrostatics, mooring, PTO, and other external forces. Furthermore, the rotational motion is governed by Euler's rigid-body equation [30], which can be written as Equation 2.9.

$$\mathbf{I} \dot{\boldsymbol{\omega}} + \boldsymbol{\omega} \times (\mathbf{I} \boldsymbol{\omega}) = \sum \mathbf{M} \quad (2.9)$$

where $\mathbf{I}$ is the mass moment of inertia tensor, $\boldsymbol{\omega}$ is the angular velocity, $\dot{\boldsymbol{\omega}}$ is the angular acceleration vector and $\mathbf{M}$ is the resultant external moments acting on the body.

2.2.3 Hydrodynamic Forces

Hydrodynamic forces arise from the interaction between the surrounding fluid and the WEC. They result from pressure and viscous stresses acting on the body's wetted surface [31]. The total hydrodynamic force and moment can be expressed as Equation 2.10 and Equation 2.11.

$$\mathbf{F} = \iint_S (p \mathbf{n} + \boldsymbol{\tau}) dS \quad (2.10)$$

$$\mathbf{M} = \iint_S \mathbf{r} \times (p\mathbf{n} + \boldsymbol{\tau}) dS \quad (2.11)$$

where $\mathbf{F}$ is the hydrodynamic force and $\mathbf{M}$ is the hydrodynamic moment. The term $\boldsymbol{\tau}$ represents the viscous stress contribution, while p is the fluid pressure. The vectors $\mathbf{r}$ and $\mathbf{n}$ are the position vector and the surface normal vector.

2.2.4 External Constraints on Motion

The motion of the wave energy converter is governed not only by hydrodynamic loading but also by external subsystems acting on the structure. In this study, the most relevant external constraints are the Power Take-Off (PTO) and mooring systems.

Power Take-Off System

The Power Take-Off system is one of the main subsystems of a wave energy converter. Its primary function is to convert wave-induced mechanical motion into usable energy [32]. In addition to energy conversion, the PTO system influences the device's dynamic response by introducing a resisting force or moment that opposes relative motion. The PTO characteristics are therefore directly related to both power production and structural loading and consequently influence the overall efficiency of the WEC and the levelized cost of energy.

Several PTO concepts exist for wave energy applications, including hydraulic, pneumatic, mechanical, and electrical systems [6]. In numerical models, the PTO is often represented in simplified form using a spring-damper behavior [32]. For an articulated wave energy converter, where energy may be extracted from relative rotational motion, the PTO can be represented by a resisting moment that depends on the relative angular velocity [7]. This relation can be expressed as Equation 2.12.

$$T(t) = -K |\dot{\theta}|^n \text{sign}(\dot{\theta}) \quad (2.12)$$

where $T(t)$ is the resisting PTO moment, $\dot{\theta}$ is the relative angular velocity, K is the PTO gain and n determines the PTO characteristic. The cases $n = 0$, $n = 1$, and $n = 2$ correspond to constant-magnitude, linear, and quadratic damping, respectively. The negative sign indicates that the PTO moment acts in the opposite direction to the motion.

Mooring System

The mooring system is an essential subsystem of a floating wave energy converter. Its main function is to maintain the device within its intended operating region while allowing the motion required for energy extraction [33]. The mooring system connects the floating device to one or more fixed anchoring points, typically located on the seabed. It therefore provides station keeping but also influences the dynamic response, overall performance, and total cost of the WEC.

Several types of mooring systems can be used for WEC applications. For floating WECs, mooring systems may be classified as passive, active, or reactive [33]. Passive mooring systems are mainly used for station keeping, whereas active mooring systems play a significant role in controlling the device's dynamic behavior. Reactive mooring systems use restraining forces as part of the power generation process. In numerical models, the mooring response is often simplified using a linear spring [34].

Mooring systems can also be classified according to the configuration of the lines, commonly as taut or catenary systems [33]. Taut mooring systems use stiff tensioned lines, whereas catenary mooring systems use slack lines.

2.3 Frequency-Domain Modeling

Frequency-domain modeling is widely used in ocean engineering. In this approach, the motion of a floating structure is assumed to exhibit harmonic oscillations that vary sinusoidally in time at the same frequency as the incident wave [34]. This assumption enables the governing equations to be linearized and solved in the frequency domain.

2.3.1 Potential Flow Theory

Frequency-domain models are commonly based on potential flow theory, which assumes that the fluid is incompressible, inviscid, and irrotational [34]. Under these assumptions, viscous effects and vorticity are therefore neglected. As a result, the velocity field can be expressed as the gradient of a scalar velocity potential ϕ , as shown in Equation 2.13.

$$\nabla\phi = \left(\frac{\partial\phi}{\partial x}, \frac{\partial\phi}{\partial y}, \frac{\partial\phi}{\partial z} \right) = (u, v, w) = \mathbf{u} \quad (2.13)$$

For an incompressible fluid, conservation of mass then leads to Laplace's equation for the velocity potential, as shown in Equation 2.14.

$$\nabla^2\phi = 0 \quad (2.14)$$

The velocity potential is obtained by solving Laplace's equation together with the appropriate boundary conditions. Once the velocity field has been obtained, the pressure field can be evaluated using Bernoulli's equation.

2.3.2 Boundary Element Method

A commonly used numerical approach for solving potential-flow problems is the boundary element method (BEM) [35]. The method is based on reformulating Laplace's equation as a boundary-integral equation, for example, using Green's functions. This reduces the dimensionality of the problem, since only the boundaries of the fluid domain need to be discretized rather than the entire volume.

2.3.3 Frequency-Domain Response

In linear frequency-domain wave-structure interaction modeling, the hydrodynamic problem is commonly divided into radiation and diffraction problems [35]. The diffraction problem describes how the incident wave is affected by the presence of the body, while the radiation problem describes the waves generated by the body's motion. In a linear formulation, the total velocity potential is decomposed into incident, diffraction, and radiation components.

The hydrodynamic quantities obtained from the radiation and diffraction problems can be assembled into the equation of motion [36]. For a linear single degree of freedom system, the motion can be described as Equation 2.15.

$$[M + A(\omega)] \ddot{x} + B(\omega)\dot{x} + Cx = F_{\text{exc}}(\omega) \quad (2.15)$$

where x is the displacement, M is the mass, $A(\omega)$ is the frequency dependent added mass, $B(\omega)$ is the radiation damping coefficient, C is the hydrostatic restoring coefficient and $F_{\text{exc}}(\omega)$ is the wave-excitation force. Assuming harmonic motion, $x = ae^{i\omega t}$, the response amplitude operator can be expressed as shown in Equation 2.16.

$$\text{RAO}(\omega) = \frac{a}{\zeta_a} = \frac{F_{\text{exc}}(\omega)}{C - (M + A(\omega))\omega^2 + iB(\omega)\omega} \quad (2.16)$$

where a is the motion amplitude of the body, ζ_a is the incident wave amplitude and i is the imaginary unit. The response amplitude operators (RAOs) are used in linear frequency-domain modeling to describe the relationship between the incoming wave and the resulting motion of the floating body at a given frequency.

2.3.4 Limitations of the Approach

Frequency-domain modeling is computationally efficient but relies on simplified assumptions when solving hydrodynamic problems. However, these assumptions can introduce errors when nonlinear or viscous effects are important. For example, Ersdal and Moe identified several limitations of the previous frequency-domain model, including green-water effects, viscous effects, and variations in submergence and wetted surface area [7].

Viscous Effects

Viscous flow effects are nonlinear hydrodynamic effects associated with shear stresses and energy dissipation in the fluid. These effects include, for instance, viscous drag, flow separation, resonance, and vortex shedding [37]. They can become particularly important when body motion and relative fluid velocities are significant. Since linear frequency-domain modeling assumes inviscid flow, viscous effects are neglected [35].

Green Water Effects

Green water is a strongly nonlinear free surface phenomenon in which water flows onto and floods the deck or upper surface of a floating structure due to large relative

wave motions [38]. It is commonly studied in ship and offshore hydrodynamics. Green water events may occur when the structure is submerged by a wave or when a large wave impacts it [39]. These events can affect the motion response and may induce significant hydrodynamic loads [40].

Submergence Effects

Submergence effects are nonlinear effects related to variations in the wetted surface of a floating body as it moves in waves [41]. In linear frequency-domain theory, body motions are assumed to be small and hydrodynamic pressures and forces are evaluated on the mean wetted surface [42]. If the submerged geometry changes significantly during the motion, this assumption becomes less accurate. As a result, linear models may not fully capture the influence of changing submergence during larger body motions.

2.4 Computational Fluid Dynamics

Computational fluid dynamics (CFD) is a numerical approach for analyzing flow by solving the governing equations of fluid motion on a discretized computational domain. Since this thesis is based on CFD simulations, the following sections introduce the governing equations, turbulence modeling, free surface modeling, and near wall treatment.

2.4.1 Governing Flow Equations

The mathematical foundation of fluid dynamics is given by the governing equations, which express the conservation of mass, momentum, and energy [43]. The conservation of mass, also known as the continuity equation, is expressed as Equation 2.17.

$$\frac{d\rho}{dt} + \rho \frac{\partial v_i}{\partial x_i} = \frac{\partial \rho}{\partial t} + \frac{\partial(\rho v_i)}{\partial x_i} = 0 \quad (2.17)$$

For incompressible flow Equation 2.17 reduces to Equation 2.18:

$$\frac{\partial v_i}{\partial x_i} = 0 \quad (2.18)$$

The conservation of momentum is described by the Navier-Stokes equations, as shown in Equation 2.19.

$$\rho \frac{dv_i}{dt} = -\frac{\partial P}{\partial x_i} + \frac{\partial \tau_{ji}}{\partial x_j} + \rho f_i = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j} \left(2\mu S_{ij} - \frac{2}{3}\mu \frac{\partial v_k}{\partial x_k} \delta_{ij} \right) + \rho f_i \quad (2.19)$$

Here P is the pressure, τ_{ji} is the stress tensor, f_i represents body forces per unit mass, μ is the dynamic viscosity and S_{ij} is the strain rate tensor, defined as Equation 2.20.

$$S_{ij} = \frac{1}{2} \left(\frac{\partial v_i}{\partial x_j} + \frac{\partial v_j}{\partial x_i} \right) \quad (2.20)$$

For constant viscosity and incompressible flow, the momentum equation reduces to Equation 2.21.

$$\rho \frac{dv_i}{dt} = -\frac{\partial P}{\partial x_i} + \mu \frac{\partial^2 v_i}{\partial x_j \partial x_j} + \rho f_i \quad (2.21)$$

2.4.2 Reynolds-Averaged Navier–Stokes Equations

When turbulence is present, the Reynolds-averaged Navier–Stokes (RANS) equations are commonly used to describe the mean flow field [43]. In the RANS approach, the velocity and pressure are decomposed into mean and fluctuating components, as in Equation 2.22.

$$\phi = \bar{\phi} + \phi' \quad (2.22)$$

Applying Reynolds averaging to the continuity and momentum equations gives the incompressible RANS equation, which can be written as Equation 2.23 and Equation 2.24.

$$\frac{\partial \bar{v}_i}{\partial x_i} = 0 \quad (2.23)$$

$$\rho \frac{\partial \bar{v}_i \bar{v}_j}{\partial x_j} = -\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j} \left(\mu \frac{\partial \bar{v}_i}{\partial x_j} - \rho \overline{v'_i v'_j} \right) \quad (2.24)$$

where $-\rho \overline{v'_i v'_j}$ is the Reynolds stress tensor, which represents the effect of turbulent fluctuations on the mean flow.

2.4.3 Turbulence Modeling

The Reynolds stress tensor is commonly closed using the Boussinesq approximation [43]. Using this approximation, the Reynolds stress tensor can be modeled as shown in Equation 2.25.

$$\overline{v'_i v'_j} = -\nu_t \left(\frac{\partial \bar{v}_i}{\partial x_j} + \frac{\partial \bar{v}_j}{\partial x_i} \right) + \frac{1}{3} \delta_{ij} \overline{v'_k v'_k} = -2\nu_t \bar{S}_{ij} + \frac{2}{3} \delta_{ij} k \quad (2.25)$$

where ν_t is the turbulent viscosity, k is the turbulent kinetic energy, δ_{ij} is the Kronecker delta, and $\bar{S}_{ij}$ is the mean strain-rate tensor. This approximation provides a closure for the RANS equations by modeling the Reynolds stresses.

The k- ϵ Turbulence Model

The k- ϵ model is a two-equation turbulence model that solves transport equations describing the turbulent kinetic energy, k and its dissipation rate ϵ , as shown in Equation 2.26 and Equation 2.27 [44]. The model is commonly used for fully turbulent flows and is therefore suitable for high Reynolds numbers. It is particularly suitable for flows with low pressure gradients.

$$\frac{\partial k}{\partial t} + \bar{v}_j \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\nu + \frac{\nu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + P_k - \varepsilon \quad (2.26)$$

$$\frac{\partial \varepsilon}{\partial t} + \bar{v}_j \frac{\partial \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\nu + \frac{\nu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + \frac{\varepsilon}{k} (C_{\varepsilon 1} P_k - C_{\varepsilon 2} \varepsilon) \quad (2.27)$$

The production term is defined in Equation 2.28 and the turbulent viscosity is defined in Equation 2.29.

$$P_k = -\overline{v'_i v'_j} \frac{\partial \bar{v}_i}{\partial x_j} \quad (2.28)$$

$$\nu_t = C_\mu \frac{k^2}{\varepsilon} \quad (2.29)$$

where C_μ , $C_{\varepsilon 1}$, $C_{\varepsilon 2}$, σ_k , and σ_ε are empirical constants.

The k- ω Turbulence Model

The k- ω model is also a two-equation turbulence model. It solves transport equations for the turbulent kinetic energy, k , see Equation 2.30 and the specific dissipation rate, ω , see Equation 2.31 [45]. Compared with the k- ϵ model, this model is better suited for predicting flows in the near-wall region and flows with adverse pressure gradients.

$$\frac{\partial k}{\partial t} + \bar{v}_j \frac{\partial k}{\partial x_j} = P_k - \beta^* k \omega + \frac{\partial}{\partial x_j} \left[(\nu + \sigma_k \nu_t) \frac{\partial k}{\partial x_j} \right] \quad (2.30)$$

$$\frac{\partial \omega}{\partial t} + \bar{v}_j \frac{\partial \omega}{\partial x_j} = \alpha \frac{\omega}{k} P_k - \beta \omega^2 + \frac{\partial}{\partial x_j} \left[(\nu + \sigma_\omega \nu_t) \frac{\partial \omega}{\partial x_j} \right] \quad (2.31)$$

where α , β^* , β , σ_k , and σ_ω are empirical constants. The turbulent viscosity is defined as Equation 2.32.

$$\nu_t = \frac{k}{\omega} \quad (2.32)$$

The SST k- ω Turbulence Model

The shear stress transport (SST) model, also known as the SST k- ω model, combines the advantages of the k- ω and k- ϵ formulations. It applies the k- ω formulation in the near-wall region and transitions towards a k- ϵ formulation in the free stream [43]. The SST k- ω model is commonly applied in CFD simulations of wave energy converters [46].

2.4.4 Free Surface Modeling

In multiphase flow, the interface between the different phases must be resolved [27]. Several approaches can be used for this purpose. A commonly used approach is the Volume of Fluid (VOF) method, which tracks the interface between phases within an Eulerian approach. In this method, α , referred to as an indicator function, represents the volume fraction of one phase. It takes values between 0 and 1, representing the

different phases, while intermediate values indicate the transition region between them. The transport equation for α can be written as Equation 2.33.

$$\frac{\partial \alpha}{\partial t} + \mathbf{u} \cdot \nabla \alpha = 0 \quad (2.33)$$

The local density and viscosity are then calculated from the volume fraction, as shown in Equation 2.34 and Equation 2.35.

$$\rho = \alpha \rho_w + (1 - \alpha) \rho_a \quad (2.34)$$

$$\mu = \alpha \mu_w + (1 - \alpha) \mu_a \quad (2.35)$$

where ρ_w and ρ_a are the densities of the two phases, while μ_w and μ_a are the corresponding dynamic viscosities. In the VOF method, the interface is represented numerically as a finite transition region in which the phases are treated as a mixture rather than as a physically sharp interface.

2.4.5 Near Wall Treatment

Near walls, the velocity gradient becomes large [43], and the associated wall shear stress is introduced as τ_w , and the friction velocity is defined as shown in Equation 2.36.

$$u_\tau = \sqrt{\frac{\tau_w}{\rho}} \quad (2.36)$$

The non-dimensional wall distance, y^+ , is defined as Equation 2.37:

$$y^+ = \frac{y u_\tau}{\nu} \quad (2.37)$$

where y is the distance from the wall and ν is the kinematic viscosity. The y^+ value can be used to evaluate whether the near-wall mesh resolution is sufficient for the chosen wall treatment. Two commonly used approaches are wall functions and low Reynolds number models.

The near-wall region of a turbulent boundary layer can be divided into different regions. Closest to the wall is the viscous region, which typically corresponds to $y^+ \leq 5$, where viscous effects dominate. Farther away from the wall is the logarithmic region, typically corresponding to $y^+ \geq 30$, where the velocity profile approximately follows the logarithmic law of the wall and turbulent diffusion dominates. Between these regions is the transitional buffer region.

3

Methodology

This chapter presents the methodology used to develop, verify, and evaluate the CFD model. The model was based on an experimental model test setup used in the WEC tests, with the aim of reproducing the experimental conditions numerically. The primary objective was to assess the agreement between the CFD results and the experimental data, as well as to compare the predictions with those obtained from the previous frequency-domain model. The chapter describes the complete modeling workflow, including the development of the WEC geometry, the construction of the computational domain, and the integration of the model with its associated subsystems.

3.1 Research Design

The overall research design used in this thesis is illustrated in Figure 3.1.

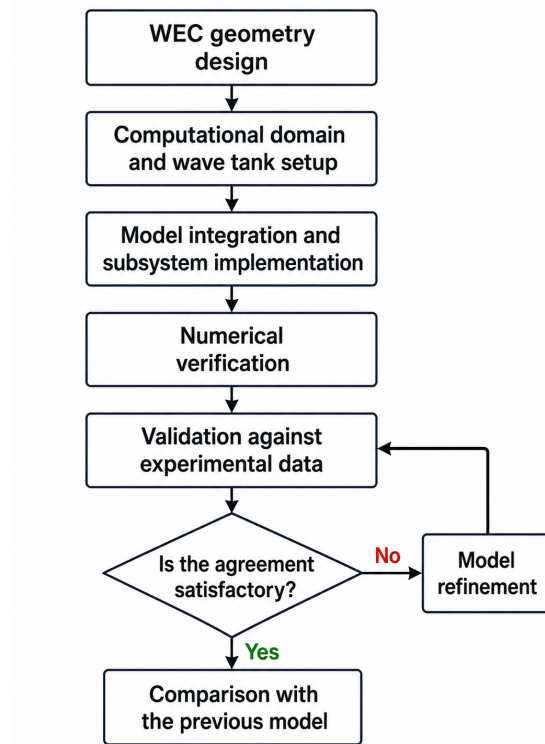


Figure 3.1: Workflow for the development and validation of the CFD model.

As shown in Figure 3.1, the workflow was structured as a stepwise modeling process. It began with the development of the WEC geometry and the construction of the numerical wave tank. These two components of the model were initially developed separately to ensure that each part was correctly defined before being combined into a complete CFD model. This stepwise approach was used to isolate potential sources of error before evaluating the full model. The complete model incorporated the relevant subsystems, including the PTO system, the mooring system, and the multibody motion of the two-body WEC.

After assembly, numerical verification was performed to ensure that the simulations were physically reasonable and numerically stable. The model was then evaluated through comparison with experimental data from the model tests. Throughout the model development process, the numerical setup, assumptions, boundary conditions, mesh resolution, and subsystem implementation were reviewed and refined.

Once the model was finalized, the simulation results were compared with the experimental data and the results from the previous frequency-domain model. This comparison was used to evaluate the performance of the developed CFD model and assess its ability to reproduce the experimental response compared with the previous modeling approach.

3.2 Model Development

The numerical model was developed in Simcenter STAR-CCM+, and the simulations were performed on a high-performance computing cluster. This section describes the development of the WEC model, the computational domain, and the complete numerical model used to replicate the experimental model test setup.

3.2.1 WEC Model

The WEC model was created using the 3D-CAD tools in STAR-CCM+ to represent the 1:12-scale prototype used in the experimental model tests. The geometry was primarily based on the available CAD model provided by Aker Solutions, which was considered the most representative source for the physical prototype's geometry. This model comprised two interconnected rigid bodies, the barge and the absorber. The barge represented the main floating body, while the absorber was the smaller body connected to the barge through the hinge.

The available geometric information was not fully consistent. Differences were identified between the reference drawings and the CAD model, and detailed dimensions were not available for all components. Furthermore, the supplied CAD model included detailed representations of the gearbox, motor, and associated components, whereas in the final experimental prototype, these components were enclosed in a protective box to prevent water ingress. Since this box represented the external geometry exposed to the surrounding fluid during the model tests, the detailed internal components were simplified and replaced by the corresponding box structure

in the numerical model.

Consequently, certain assumptions and simplifications were required during the development of the geometry. The final WEC model was developed to represent the experimental prototype as accurately as possible, using available CAD data, reference drawings, and identifiable features of the physical prototype. The front and top views of the finalized model are shown in Figure 3.2.

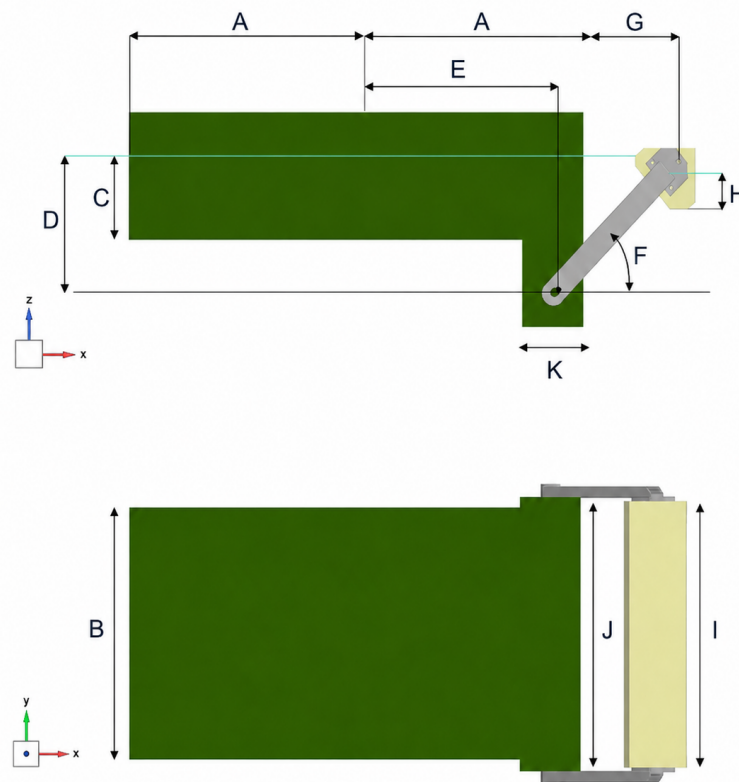


Figure 3.2: Front and top views of the WEC.

The reference letters shown in the geometry views correspond to the dimensions listed in Table 3.1. These include the principal dimensions of the barge, absorber, and hinge location, as well as the protective box structure.

Table 3.1: Main geometric dimensions of the WEC model used in the CFD simulation.

Dimension	Reference	Value	Unit
Half length of barge	A	567.5	mm
Width of barge	B	500	mm
Draft of barge	C	166	mm
Depth of hinge	D	266	mm
Longitudinal position of hinge	E	489.5	mm
Nominal hinge angle	F	31.7	degrees
Barge front to absorber	G	372.5	mm
Draft of absorber body	H	115.6	mm
Width of absorber body	I	540	mm
Width of barge box	J	560	mm
Length of barge box	K	135	mm
Arm length	—	459.4	mm

In addition to the external geometry, the rigid body mass properties of the WEC model are summarized in Table 3.2.

Table 3.2: Mass and inertial properties of the WEC model.

Property	Barge	Absorber	Unit
Mass	109.584	8.039	kg
CoG, x	0.636	1.368	m
CoG, y	0.000	0.000	m
CoG, z	0.010	-0.034	m
Moment of inertia, I_{xx}	4.296	0.523	kg m ²
Moment of inertia, I_{yy}	14.760	0.635	kg m ²
Moment of inertia, I_{zz}	13.195	0.312	kg m ²

The radii of gyration were prescribed from the available model data, while the mass and center of gravity were determined from the constructed geometry. The coordinate system was defined with the still water level as the vertical reference and the origin located at the left edge of the barge. The moments of inertia were evaluated about the center of mass. To ensure consistency with the floating condition of the experimental prototype, the mass and center of gravity were iteratively adjusted until the numerical model reproduced the intended draft and equilibrium position in water. Compared with the published data, the barge mass differs by approximately 3%, while the absorber mass differs by less than 1%.

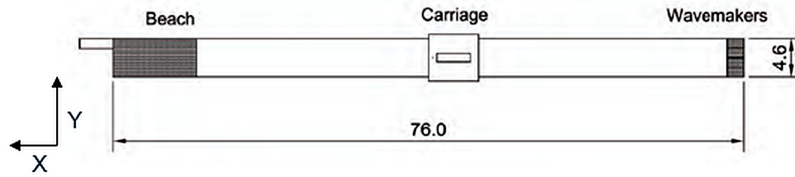
3.2.2 Numerical Wave Tank Setup

The computational domain was designed to replicate the towing tank used in the experimental study. The main dimensions of the experimental towing tank are summarized in Table 3.3. The experiments were conducted using regular waves with a wave height of 0.04 m and wave periods ranging from 0.7 to 2.2 s.

Table 3.3: Properties of the towing tank used in the experimental study.

Property	Value
Tank length	76.0 m
Tank width	4.6 m
Tank depth	2.5 m
Water depth	2.15 m

The experimental towing tank is shown in Figure 3.3.

**Figure 3.3:** Experimental towing tank setup.

In the experiments, the waves propagated along the length of the tank. This arrangement was used to define the numerical domain in STAR-CCM+, with the wave propagation direction prescribed in the negative x -direction. In the experimental setup, the waves were generated by a computer-controlled absorbing wave maker.

In the CFD model, the incident waves were generated numerically using the Volume of Fluid (VOF) wave model in STAR-CCM+. This model provides predefined wave functions that can be used to initialize the wave field and prescribe transient wave boundary conditions. Several wave formulations were available within this framework, and a fifth-order wave model was selected in this study to represent the regular waves, as it provided a more realistic wave profile than linear wave theory. Regular waves were used in all simulations to reproduce the experimental conditions and enable comparison with the previous frequency-domain model. Prior to each simulation, the wave field was initialized throughout the computational domain to improve numerical stability and reduce the required simulation time. The wave propagation direction was defined to match the experimental setup.

For the CFD domain, the tank width and water depth were kept consistent with the experimental setup. A still water level was defined at $z = 0$, with the water phase extending below this level and an air region included above it. To limit computational cost, the total domain length was reduced relative to the full experimental towing tank. Therefore, instead of modeling the full tank length, the computational domain length was scaled according to the wavelength of each simulated wave case. Consequently, the total domain length varied between the different wave periods. A schematic representation of the CFD domain is shown in Figure 3.4.

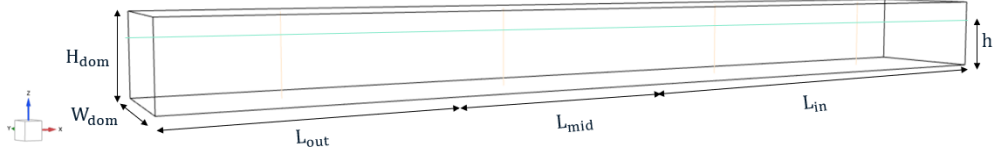


Figure 3.4: Schematic representation of the CFD domain.

The symbols shown in Figure 3.4 correspond to the domain dimensions listed in Table 3.4.

Table 3.4: Definition of the main dimensions used in the CFD domain.

Symbol	Description	Value	Unit
W_{dom}	Domain width	4.6	m
H_{dom}	Total domain height	2.65	m
h	Still water depth	2.15	m
L_{in}	Distance from inlet to central region	2λ	m
L_{mid}	Length of central region containing the WEC	2.0	m
L_{out}	Distance from central region to outlet	2λ	m
L_{tot}	Total domain length	$4\lambda + 2.0$	m

In addition, wave probes were used to monitor the free surface elevation along the domain. These probes were used to evaluate wave propagation and verify that the desired wave height was maintained in the region of interest. Two probes were positioned close to the inlet and in the central region where the WEC was placed in the integrated model.

The boundary conditions used in the numerical model are summarized in Table 3.5.

Table 3.5: Boundary conditions and applied forcing.

Boundary	Type	Forcing
Inlet	Velocity inlet	1λ
Outlet	Pressure outlet	1λ
Bottom	No-slip wall	—
Front	No-slip wall	—
Back	No-slip wall	—
Top	Slip wall	—

A velocity inlet was applied at the inlet, while a pressure outlet was applied at the outlet. At the inlet, the volume fraction and velocity field corresponding to the incident wave were prescribed. At the outlet, the hydrostatic pressure and volume fraction associated with the fifth-order VOF waves were applied. The bottom, front, and back boundaries were defined as no-slip walls, whereas the top boundary was defined as a slip wall. Wave forcing was applied at both the inlet and outlet. In the present study, a forcing length of one wavelength was applied at both the inlet and

outlet to reduce the required domain length and thereby limit the computational cost.

The flow was modeled as a two-phase system, with the free surface captured using the Volume of Fluid (VOF) method. To be consistent with the model tests conducted in freshwater, the numerical model also used freshwater properties for both phases. The fluid properties employed for water and air are summarized in Table 3.6.

Table 3.6: Input properties for water and air used in the study.

Property	Water	Air	Unit
Density, ρ	998.2	1.204	kg/m ³
Dynamic viscosity, μ	1.002×10^{-3}	1.825×10^{-5}	Pa·s
Kinematic viscosity, ν	1.004×10^{-6}	1.516×10^{-5}	m ² /s
Temperature, T	20	20	°C
Reference pressure, p_{ref}	101325	101325	Pa

The simulations were performed as three-dimensional, implicit, unsteady calculations in STAR-CCM+. The flow field was solved using a segregated flow solver, and the governing equations were modeled using the RANS equations. Turbulence was modeled using the SST k - ω model, and an all y^+ wall treatment was applied at the wall boundaries. The second-order, high-resolution interface capturing (HRIC) scheme was used to capture the free surface, while the SIMPLE algorithm was used for pressure-velocity coupling.

Each wave simulation ran for at least ten wave periods. The wave height was evaluated over the final five wave periods to ensure that the wave field had reached a stable periodic state, defined as the point at which the wave height and shape remained consistent between successive wave periods. Each time step was solved using ten inner iterations.

Wave Tank Mesh Strategy

The mesh strategy was developed to accurately resolve free surface motion and wave propagation while maintaining a reasonable computational cost. Since the simulations were performed for several wave periods, both the wavelength and the domain length varied between cases. Therefore, trimmed hexahedral cells were used, and the mesh resolution was defined relative to the wave height and wavelength rather than using fixed cell sizes for all cases. Depending on the simulated wave period and corresponding wave domain length, the final meshes contained approximately 3.0-4.3 million cells.

The vertical resolution in the free surface region was defined using a prescribed number of cells per wave height, N_H . The corresponding vertical cell size is given by Equation 3.1.

$$\Delta z_{\text{wave}} = \frac{H}{N_H} \quad (3.1)$$

Similarly, the horizontal resolution in the wave propagation direction was based on a prescribed number of cells per wavelength N_λ , as shown in Equation 3.2.

$$\Delta x_{\text{wave}} = \frac{\lambda}{N_\lambda} \quad (3.2)$$

Mesh adaptation accounted for the different wave periods considered in the study. With increasing wavelength, the domain length increased, and the streamwise cell size was adjusted accordingly. In STAR-CCM+, an automated expression selected the coarsest streamwise refinement level that still satisfied the prescribed number of cells per wavelength. Consequently, the total number of cells remained within the same order of magnitude across all wave cases while maintaining the prescribed resolution for each wavelength.

A refinement zone was applied around the still water level to accurately capture the free surface. This region was refined because large volume fraction gradients occur at the air-water interface. Vertical extent was defined based on wave height to ensure that full wave motion remained within the refined region. The refinement zone extended $\pm 1.5H$ around the still water level in the vertical direction and covered the full domain in the streamwise and transverse directions. This setup reduced numerical smearing of the volume-fraction field.

Further away from the free surface, the mesh was gradually coarsened in regions where smaller gradients were expected. The background mesh size was defined as a multiple of the vertical cell size. This reduced the total cell count without compromising the resolution in the main region of interest. Figure 3.5 shows a close-up of the free surface refinement zone, while Figure 3.6 shows a close-up of the prism layers at the tank wall.

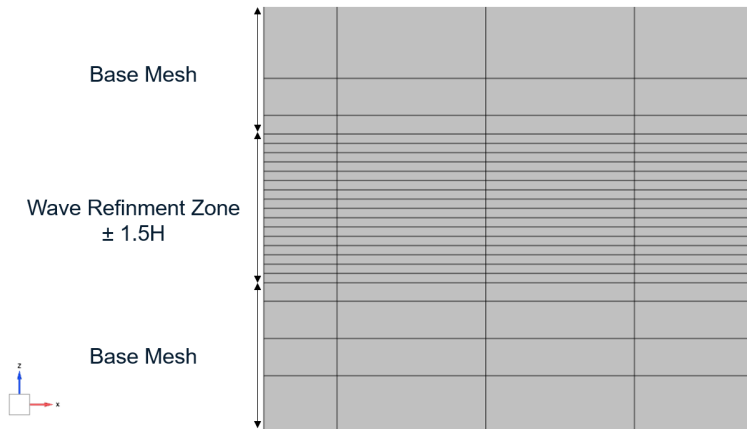


Figure 3.5: Illustration of the free surface refinement zone.

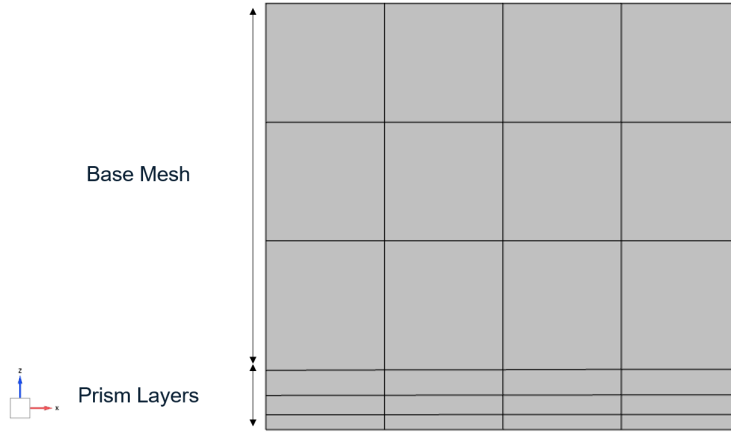


Figure 3.6: Illustration of the prism-layer mesh.

Three prism layers were applied to the no-slip tank walls and later to the WEC surface within the full-system model. The total prism-layer thickness was set equal to the base mesh size, with a prism-layer stretching factor of 1.3. Since the wall boundaries were located away from the main region of interest, the prism-layer resolution was kept limited.

Wave Tank Mesh Study

A mesh sensitivity study was performed to assess the influence of the spatial discretization on the generated waves. The study considered variations in the number of cells per wave height and wavelength, as well as in the background mesh size. The objective was to identify a mesh strategy applicable to all investigated wave periods while keeping the difference between the inlet wave height and the wave height at the WEC position within approximately 5%. According to the STAR-CCM+ VOF wave modeling guidelines, the free surface refinement region should enclose the full wave motion, with a recommended resolution of approximately 20 cells per wave height and 80-100 cells per wavelength within the refinement zone [47]. These recommendations were used as guidance when defining the mesh study.

The mesh study was carried out for three representative wave cases, corresponding to the shortest, intermediate, and longest wave periods considered in the simulations. These cases are summarized in Table 3.7.

Table 3.7: Representative wave cases considered in the mesh study.

Case	Period (s)	Wavelength (m)	Wave height (m)
Short wave	0.7	0.77	0.04
Medium wave	1.5	2.25	0.04
Long wave	2.2	7.56	0.04

The mesh performance was evaluated by comparing the generated wave height at

selected wave probes located near the inlet and at the center of the domain. The relative change in wave height between these probe locations was calculated according to Equation 3.3.

$$\varepsilon_H = \frac{H_{\text{middle}} - H_{\text{inlet}}}{H_{\text{inlet}}} \times 100 \quad (3.3)$$

where H_{inlet} is the wave height measured near the inlet and H_{middle} is the wave height measured at the center of the domain. The mesh sensitivity cases are summarized in Table 3.8.

Table 3.8: Summary of mesh sensitivity studies showing vertical, horizontal, and background mesh refinements.

Case	Cells per wave height (N_H)	Cells per wavelength (N_λ)	Base size
Z1	5	120	$8 \Delta z$
Z2	10	120	$8 \Delta z$
Z3	15	120	$8 \Delta z$
Z4	20	120	$8 \Delta z$
X1	N_H^*	60	$8 \Delta z$
X2	N_H^*	80	$8 \Delta z$
X3	N_H^*	120	$8 \Delta z$
X4	N_H^*	160	$8 \Delta z$
B1	Final	Final	$4 \Delta z$
B2	Final	Final	$8 \Delta z$
B3	Final	Final	$16 \Delta z$
B4	Final	Final	$32 \Delta z$

Here, N_H^* denotes the selected vertical resolution from the Z-cases, while "Final" denotes the selected values of N_H and N_λ from Z- and X-cases. The Z-cases were used to study the influence of the vertical free surface resolution. The X-cases were used to evaluate the effect of horizontal resolution on the direction of wave propagation. The B-cases were then performed after selecting the final values of N_H and N_λ from the Z- and X-case studies to study the influence of the background mesh size.

In addition, a time step sensitivity study was conducted to assess the influence of temporal discretization on the generated waves. The recommended time step from the STAR-CCM+ VOF wave modeling guidelines is given by Equation 3.4 [47].

$$\Delta t = \frac{P}{2.4 n} \quad (3.4)$$

where P denotes the wave period and n is the number of divisions of the wavelength [47]. This recommendation was used as a guideline for selecting the investigated time step cases, which are summarized in Table 3.9.

Table 3.9: Time step refinement cases used in the wave tank simulations.

Case	Time step Δt
T1	0.01
T2	0.005
T3	0.0025
T4	0.001

In addition to wave height error and the stability of the generated wave field, computational cost was considered when selecting the final time step sizes. The CPU time required to simulate one wave period was estimated using Equation 3.5.

$$t_{\text{CPU},T} = \frac{T}{\Delta t} t_{\text{step}} \quad (3.5)$$

where T is the wave period, Δt is the time step size and t_{step} is the average CPU time per time step. This estimate was used to compare the computational cost of different time step cases.

Overall, the purpose of the mesh and time step sensitivity studies was to identify a numerical setup that could accurately maintain the prescribed wave height and reduce numerical diffusion at the air-water interface. At the same time, the mesh resolution was reduced in regions with lower expected gradients to limit the computational cost. The selected setup was required to maintain the generated wave height with only a small variation between the inlet and the central region of the domain. The final mesh and time step settings were selected based on the relative wave height error, the stability of the generated wave field, and the associated computational cost.

3.2.3 Integration of the WEC into the Fluid Domain

After developing the fluid domain, the 3D-CAD model of the WEC was integrated into the final CFD model. In accordance with the experimental model test setup, the WEC was positioned at the center of the computational domain, corresponding to the intended measurement location in the numerical wave tank. Its motion was then modeled together with the PTO and mooring systems. In addition, full-system mesh and time step sensitivity studies were performed to assess the numerical sensitivity of the integrated model.

Body Integration

An important step in the integration process was defining the WEC's initial position and verifying its hydrostatic equilibrium before applying wave loads. The initial position of the WEC in the fluid domain is shown in Figure 3.7.

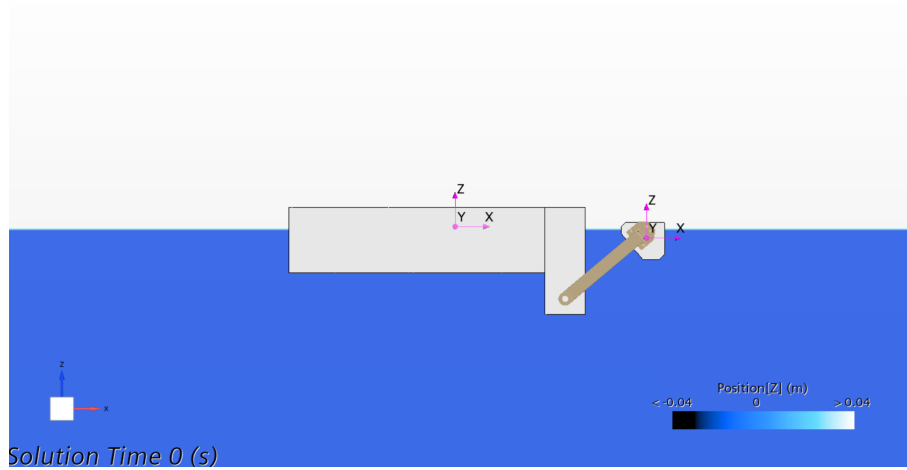


Figure 3.7: Initial position of the WEC relative to the still water level in the computational domain.

To avoid an initial moment, the center of gravity and the center of buoyancy were aligned in the x - and y -directions, since any horizontal offset between these points would generate an initial moment and could cause the structure to rotate artificially at the beginning of the simulation. The equilibrium condition was verified by simulating the structure in still water and confirming that no initial motion occurred. Consequently, any subsequent motion of the WEC was caused by incoming waves rather than by initial imbalance. The WEC surfaces were defined as no-slip wall boundaries.

DFBI Motion and Hinge Coupling

Body motion was modeled using two DFBI bodies: one representing the absorber and one representing the barge. Multibody motion was applied to both bodies. Each body was allowed to move in three degrees of freedom: surge, heave, and pitch. This resulted in a total of six degrees of freedom for the WEC system, rather than the full twelve degrees of freedom present for two unconstrained rigid bodies. A ramp time equal to one wave period was used to gradually introduce the incident wave and reduce transient numerical effects at the beginning of the simulation.

The connecting arm was not included as a fluid-interacting surface in the simulation. Instead, it was represented as a body that followed the WEC's motion. This simplification was introduced to reduce mesh complexity and computational cost. However, the arm was included in the model's visual representation to improve the interpretation of the WEC motion.

To model the hinge connection, a body coupling was introduced as a revolute joint between the absorber and the barge. This joint represented the physical hinge connection and allowed relative rotation about the hinge axis, while constraining all other relative translations and rotations. It was defined relative to the barge's center of gravity and aligned with the hinge axis.

Power Take-Off System Implementation

The PTO system was represented by an internal resisting moment acting opposite to the relative hinge motion between the absorber and the barge. In the previous frequency-domain model, a linear PTO formulation was used. This formulation was based on model tests, in which the controller input for the PTO system was set to $K = 4.0$. However, a least-squares fit to the measured torque-angular velocity data resulted in an effective linear gain factor of $K = 6.7$ [7], as shown in Figure 3.8.

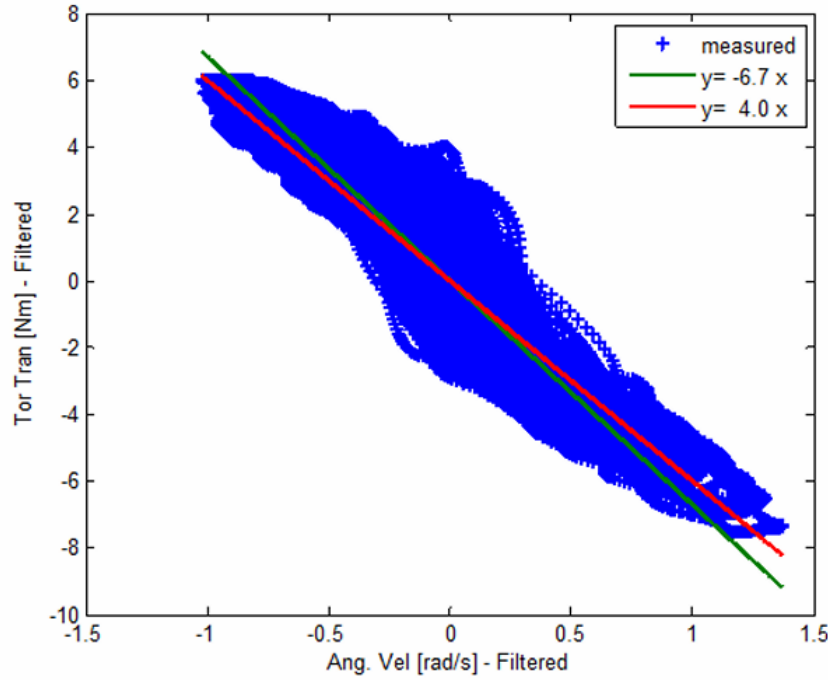


Figure 3.8: Linear PTO model showing the measured torque-angular velocity relationship and the least-squares fit used to determine the effective linear PTO gain coefficient [7].

In the present study, this linear formulation was implemented as the baseline PTO model to enable comparison with previous frequency-domain results. The relative hinge angular velocity was calculated according to Equation 3.6.

$$\omega = \omega_{y,\text{Absorber}} - \omega_{y,\text{Barge}} \quad (3.6)$$

The baseline PTO system was therefore implemented in accordance with Equation 3.7.

$$M_{\text{PTO}} = -6.7\omega \quad (3.7)$$

During the present work, the PTO representation was re-evaluated to verify the formulation used in the CFD model and to investigate whether a purely linear damping model was sufficient to describe the measured torque-angular velocity relation. The replotted data indicated that the measured PTO response exhibited a steep transition near zero angular velocity, while the outer branches were approximately linear. This behavior is illustrated in Figure 3.9.

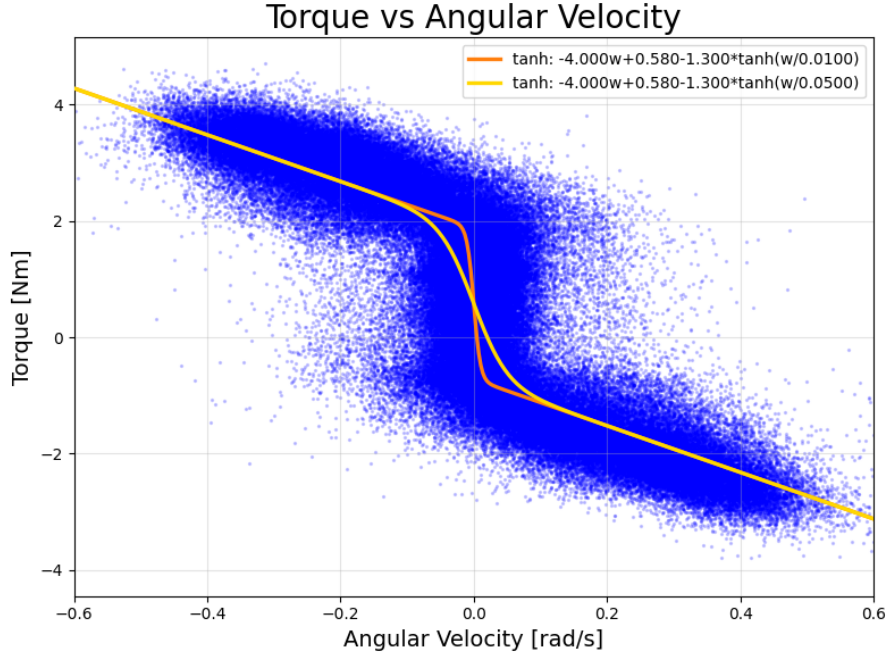


Figure 3.9: Re-evaluated PTO model using the measured model test torque and angular-velocity data.

Figure 3.9 shows the full data set in the converged response region. The corresponding fitted data for each wave period are provided in Appendix B. Based on the observed nonlinear behavior around zero angular velocity, an alternative PTO formulation was introduced using a smooth hyperbolic tangent function, as shown in Equation 3.8. This formulation was not intended to replace the linear baseline model, but was included to assess the sensitivity of the CFD response to the representation of the measured PTO behavior.

$$M_{\text{PTO}} = -4.000\omega + 0.580 - 1.300 \tanh\left(\frac{\omega}{a}\right) \quad (3.8)$$

where a controls the width of the transition around zero velocity. Two values, $a = 0.01$ and $a = 0.05$, were investigated to assess the sensitivity of the predicted hinge-angle response to the sharpness of this transition. The smaller value represents a steeper transition, while the larger bandwidth gives a smoother transition.

The hyperbolic tangent form was chosen because it captured the main shape of the measured torque-angular velocity data. The linear term represented the approximately linear outer branches of the measured relation and corresponded to the nominal controller input of $K = 4.0$, while the hyperbolic tangent term captured the steep transition around $\omega = 0$, which was likely associated with friction in the gearbox. The coefficients were selected by comparing different parameter combinations based on their agreement with the measured data, using the error measures defined in Section 3.3.3. Although some outliers may have influenced the fitting result, the selected curve provided a representative description of the dominant trend in the data.

The linear baseline model and both nonlinear PTO models were defined as field functions in STAR-CCM+. The resulting PTO moment was applied to the DFBI bodies of the absorber and the barge with equal magnitude but opposite signs, ensuring that the moment resisted the relative hinge motion.

Mooring System Implementation

The mooring system used in the model tests was not fully specified. Based on discussions with a previous researcher involved in the experiments, the system was assumed to be a soft mooring system with unknown global stiffness. The mooring lines were therefore assumed to function primarily as a station keeping system, without having a dominant influence on the dynamic response of the WEC.

Based on images of the experimental setup, the mooring arrangement appeared to consist of four lines oriented approximately 45° relative to the tank direction. This configuration was implemented in STAR-CCM+ using DFBI body coupling, where the mooring system was represented by four spring-damper elements. The lines were attached to the WEC and connected to fixed points in the computational domain.

Since the global mooring stiffness was unknown, the stiffness parameters were determined iteratively. The elastic coefficient was adjusted until the simulated surge oscillation matched the dominant surge behavior observed in the experimental data. Here, the global stiffness refers to the effective stiffness of the complete mooring arrangement, while the elastic coefficient refers to the stiffness assigned to each spring element. The final properties assigned to the mooring system are summarized in Table 3.10.

Table 3.10: Mooring system properties.

Property	Description / Value
Global stiffness	4.8 N/m
Elastic constant	2.0 N/m
Relaxation length	8.0 m
Total mooring length	10.0 m

The resulting mooring setup was assessed to ensure that it did not excessively influence the WEC response. To support this assumption, a limited sensitivity study was conducted in which the global mooring stiffness was both doubled and halved. The corresponding variations in hinge-angle amplitude were found to be negligible, indicating that the predicted response is not sensitive to the chosen mooring stiffness. The final mooring configuration is shown in Figure 3.10.

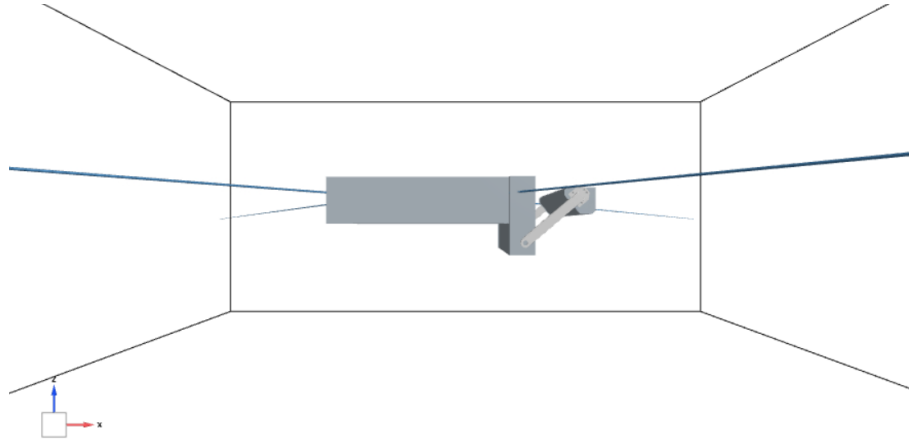


Figure 3.10: Implemented mooring configuration in the computational domain.

This configuration provided a restoring force while allowing the WEC to respond freely to the incident waves.

Full System Mesh Strategy

The mesh used for the full system simulation was a morphing mesh, which allowed the computational grid to deform with boundary motion without requiring re-meshing of the domain. This approach was chosen instead of an overset mesh method to avoid interpolation between separate meshes. This was considered suitable because the body motions were expected to be relatively smooth and moderate.

An initial mesh study was performed for the numerical wave tank without the WEC, as described in Section 3.2.2. The purpose of this study was to ensure that the incident wave was accurately resolved before introducing the floating structure. The resulting mesh resolution was then used as a lower limit for the full-system simulations to preserve the wave quality after the WEC was included.

For the complete CFD model, the mesh strategy focused on resolving the primary quantity of interest, the hinge-angle response, while keeping the computational cost as low as possible. Since a large number of simulations were required for the different wave periods, the aim was to use the coarsest mesh that still provided sufficient accuracy in the predicted hinge-angle amplitude.

Two additional refinement regions were therefore introduced: one local refinement region surrounding the WEC hull and one surface refinement applied to the WEC geometry. The final full-system mesh included the free surface refinement region from the wave tank mesh study, a local refinement zone around the WEC hull, and a body-fitted surface refinement on the WEC geometry. The free surface refinement was retained to preserve wave quality, while local hull and surface refinements were introduced to improve resolution near the moving bodies. The corresponding refinement zones are illustrated in Figure 3.11.

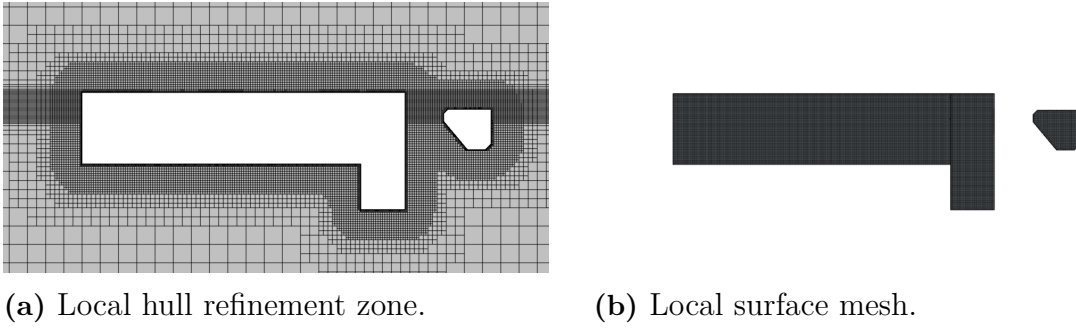


Figure 3.11: Illustration of the full-system refinement zones and the WEC surface mesh.

Figure 3.11a shows the local hull refinement, while Figure 3.11b shows the surface mesh applied to the WEC geometry. Moreover, the characteristic mesh size used in these regions was based on the smallest resolution obtained from the wave tank mesh study, as defined in Equation 3.9.

$$\Delta_{\text{wave}} = \min\left(\frac{H}{N_H}, \frac{\lambda}{N_\lambda}\right) \quad (3.9)$$

The value of Δ_{wave} therefore represented the smaller of the vertical free surface cell size and the horizontal wave propagation cell size.

Full System Mesh Study

A mesh sensitivity study was performed to investigate how local hull and surface refinements affected the hinge-angle response. The hinge-angle amplitude was used as the evaluation metric, as it was the primary response parameter in this study. To limit the computational cost, the study was performed for a representative intermediate wave case corresponding to $T = 1.5$ s. This case was selected because it provided a representative, full-system response.

First, the local hull refinement around the WEC was varied according to the cases listed in Table 3.11. The local hull refinement was defined as a refinement region extending approximately 0.1 m around the WEC geometry. This study was used to determine the influence of the surrounding mesh resolution on the predicted motion response.

Table 3.11: Local hull refinement study for the full geometry with waves.

Case	Local hull refinement size
H1	$4.0 \Delta_{\text{wave}}$
H2	$2.0 \Delta_{\text{wave}}$
H3	$1.0 \Delta_{\text{wave}}$

After selecting a suitable local hull refinement, the surface mesh on the WEC geometry was varied while keeping the hull refinement fixed. The surface refinement cases are summarized in Table 3.12.

Table 3.12: Surface mesh sensitivity study for hinge response with fixed hull refinement.

Case	Surface target size
S1	$4.0 \Delta_{\text{wave}}$
S2	$2.0 \Delta_{\text{wave}}$
S3	$1.0 \Delta_{\text{wave}}$

A time step sensitivity study was also performed for the full geometry with waves. The reference time step, Δt_{wave} , was based on the time step resolution selected from the initial wave tank mesh study described in Section 3.2.2. The investigated time step cases are shown in Table 3.13.

Table 3.13: Time step sensitivity study for the full geometry with waves.

Case	Time step
TG1	Δt_{wave}
TG2	$0.75 \Delta t_{\text{wave}}$
TG3	$0.5 \Delta t_{\text{wave}}$

Based on the mesh and time step sensitivity studies, the final refinement zones were selected to balance numerical accuracy and computational cost. The selected settings were applied to the full set of wave periods used in the comparison with experimental data.

3.3 Model Evaluation and Comparison

The main objective of this study was to compare the simulated hinge-angle amplitude with the experimental data and the results from the previously developed frequency-domain model. This section describes the selected comparison cases, the procedure for extracting hinge-angle amplitudes, and the error metrics used to evaluate and quantify these differences.

3.3.1 Selected Comparison Cases

To ensure consistency, simulated wave periods were selected to match those in the experimental test data. Consequently, the comparison was limited to cases with corresponding experimental measurements, enabling direct evaluation of each CFD simulation against the measured hinge-angle response, wave height, and angular velocity at the hinge. The corresponding wavelengths and wave heights are summarized in Table 3.14.

Table 3.14: Wave periods, calculated wavelengths, and wave heights used in the comparison with experimental data.

Period (s)	Wavelength (m)	Wave height (m)
0.70	0.77	0.04
0.75	0.88	0.04
0.80	1.00	0.04
0.85	1.13	0.04
0.90	1.26	0.04
0.95	1.41	0.04
1.00	1.56	0.04
1.10	1.89	0.04
1.20	2.25	0.04
1.25	2.44	0.04
1.30	2.64	0.04
1.50	3.51	0.04
1.70	4.51	0.04
1.80	5.06	0.04
2.00	6.25	0.04
2.20	7.56	0.04

A total of 48 final simulation cases were conducted for the selected wave periods included in the comparison. This number excludes preliminary simulations conducted during model development, including mesh sensitivity studies, time step refinement, and intermediate build-up simulations.

To perform all cases efficiently, a design manager study was used in STAR-CCM+. The same reference simulation file was used for all cases, while the wave period was varied between simulations. Since the numerical setup was defined as a function of the wave period, changing this input also modified the corresponding wavelength, domain length, mesh distribution, and time step.

3.3.2 Hinge-Angle Amplitude Extraction

In the CFD simulations, the hinge angle was calculated as the relative angular displacement between the absorber and the barge, as shown in Equation 3.10.

$$\theta_{\text{hinge}}(t) = \theta_A(t) - \theta_B(t) \quad (3.10)$$

To obtain a representative amplitude, the final five wave periods of the stabilized response were used. For each wave period, the hinge-angle amplitude was calculated as half of the difference between the maximum and the minimum hinge angle. The mean amplitude over the final five wave periods was computed according to Equation 3.11.

$$\overline{A}_\theta^{(5)} = \frac{1}{5} \sum_{k=N-4}^N \frac{\theta_{\text{hinge},k}^{\text{max}} - \theta_{\text{hinge},k}^{\text{min}}}{2} \quad (3.11)$$

3.3.3 Error Metrics

The hinge-angle amplitudes predicted by the CFD simulations were compared with the mean experimental values and with the results from the previous frequency-domain model. The comparison was performed at the wave periods for which experimental data were available. Therefore, no interpolation was applied, and all methods were evaluated only at the available experimental wave periods.

To quantify the agreement between each model and the experimental data, the signed error was first calculated for each wave period, as shown in Equation 3.12.

$$e_i = \theta_{\text{model},i} - \theta_{\text{exp},i} \quad (3.12)$$

The overall deviation between the model predictions and the experimental data was evaluated using root mean square error (RMSE), defined in Equation 3.13.

$$\text{RMSE} = \sqrt{\frac{1}{N} \sum_{i=1}^N (\theta_{\text{model},i} - \theta_{\text{exp},i})^2} \quad (3.13)$$

where N is the number of wave periods included in the comparison. To enable comparison relative to the magnitude of the measured experimental response, the RMSE was normalized by the range of the experimental hinge-angle amplitude, as shown in Equation 3.14.

$$\text{NRMSE} = \frac{\text{RMSE}}{\max(\theta_{\text{exp}}) - \min(\theta_{\text{exp}})} \times 100 \quad (3.14)$$

where the maximum and minimum values are taken over the experimental hinge-angle amplitudes included in the comparison.

4

Results and Discussion

This chapter presents and discusses the main results from the CFD simulations of the articulated-body wave energy converter. The predicted hinge-angle response is evaluated by comparing it with experimental measurements and the previous frequency-domain model. Moreover, selected flow visualizations are included to support the interpretation of the response. Finally, the wave elevation validation, mesh sensitivity study, and time step sensitivity study are presented to assess the reliability of the numerical setup.

4.1 Predicted Hinge-Angle Response

The hinge-angle represents the relative rotational motion between the barge and the absorber body and is therefore the primary response quantity used for model evaluation. In this study, the comparison is based on the extracted steady-state hinge-angle amplitude for each investigated wave period, as presented in Section 3.3.1.

As described in Section 3.2.3, the PTO representation was identified as a potential source of uncertainty, since the measured torque-angular velocity relationship deviates from the linear PTO assumption. Here K denotes the PTO gain factor, while the parameter a controls the transition width of the nonlinear PTO relation around zero angular velocity. The hinge-angle response is evaluated using both the linear PTO representation and the nonlinear representations. Table 4.1 summarizes the models included in the comparison, while Figure 4.1 shows the PTO formulations alongside the measured torque-angular velocity data.

Table 4.1: Overview of models included in the hinge-angle comparison.

Model	Description
Experiment	Experimental hinge-angle response from the towing-tank measurements.
CFD linear PTO	CFD model using a linear PTO representation with the effective PTO gain $K = 6.7$.
CFD nonlinear PTO	CFD model using the selected fitted nonlinear PTO relation based on the measured torque-angular velocity behavior.
Frequency-domain model	Previous frequency-domain model based on the boundary element method.

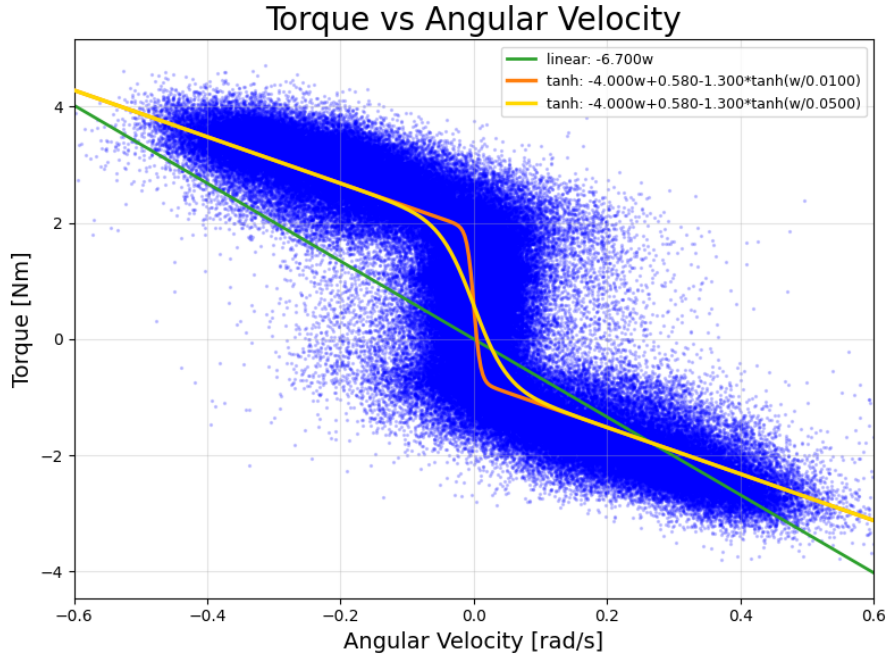


Figure 4.1: Comparison of the linear PTO model, nonlinear PTO model, and measured torque-angular velocity data.

The hinge-angle results are presented in four steps. First, the CFD results obtained using the linear PTO model are compared with those from the previous frequency-domain model to assess the influence of the hydrodynamic modeling approach. The influence of the PTO representation is then evaluated by comparing CFD results with experimental measurements, including both linear and nonlinear formulations and the effect of the nonlinear bandwidth on the predicted response. Next, the results are discussed through an overall comparison of the different models. Finally, an error analysis is performed to quantify the model's performance.

4.1.1 Hinge-Angle Response with Linear PTO

The CFD results obtained with the linear PTO model, using $K = 6.7$, are compared with the previous frequency-domain model in Figure 4.2. The experimental measurements are included as a reference. This comparison assesses whether the higher-fidelity CFD approach improves the prediction of the hinge-angle response when using the same effective PTO gain representation.

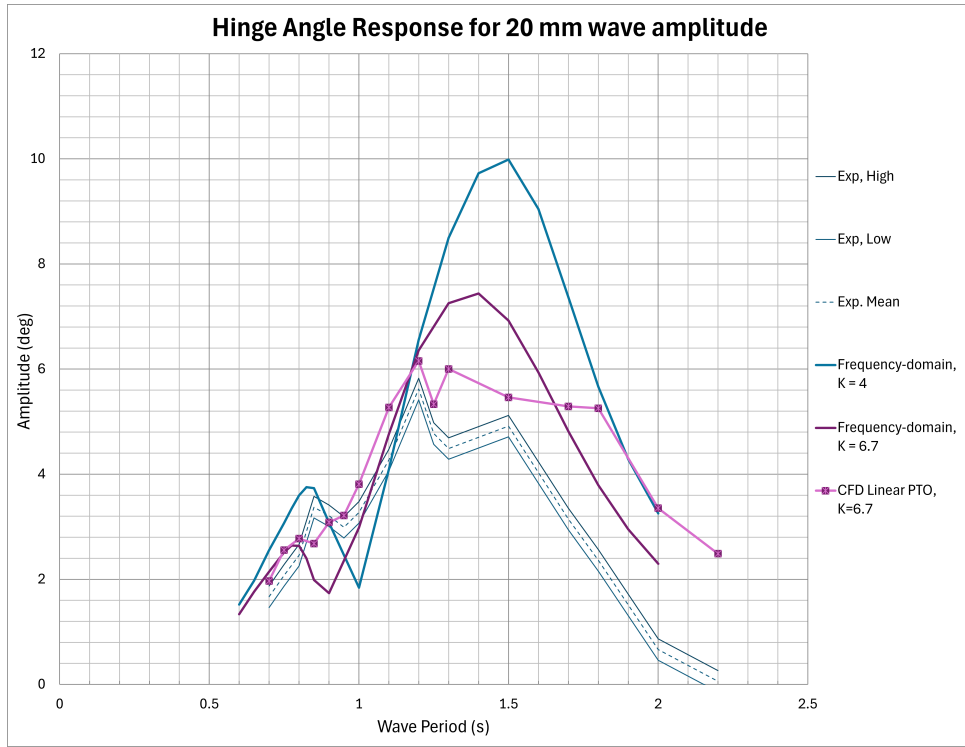


Figure 4.2: Comparison of the hinge-angle amplitudes from the experimental measurements, the CFD simulations with the linear PTO model $K = 6.7$ and the previous frequency-domain model.

Figure 4.2 shows that the CFD model captures the main features of the experimental hinge-angle response, including the increase toward the peak response region. In this region, the frequency-domain model predicts significantly higher hinge-angle amplitudes than those measured experimentally. Compared with the frequency-domain model, the CFD results predict a lower peak amplitude and reduce the overprediction observed around $T = 1.3$ s. The CFD model therefore shows closer agreement with the experimental measurements in this region, although it still overpredicts the response for several wave periods after the peak.

At longer wave periods, the discrepancy between the CFD results and experimental measurements increases. In this region, the frequency-domain model appears to give better agreement with the experimental measurements than the CFD model with the linear PTO representation. This indicates that the CFD model improves prediction over much of the response range, particularly around the peak, but the improvement is not uniform across all wave periods.

4.1.2 Influence of PTO Modeling

The influence of the PTO representation was evaluated by comparing CFD results from the linear formulation with those from the two fitted nonlinear PTO formulations. The nonlinear PTO formulation differs in the bandwidth parameter introduced in Equation 3.8. The parameter a controls the sharpness of the transition

around zero angular velocity. The case with $a = 0.01$ represents a steeper transition, whereas $a = 0.05$ represents a smoother transition. The resulting hinge-angle amplitudes are compared with the experimental hinge-angle measurements in Figure 4.3.

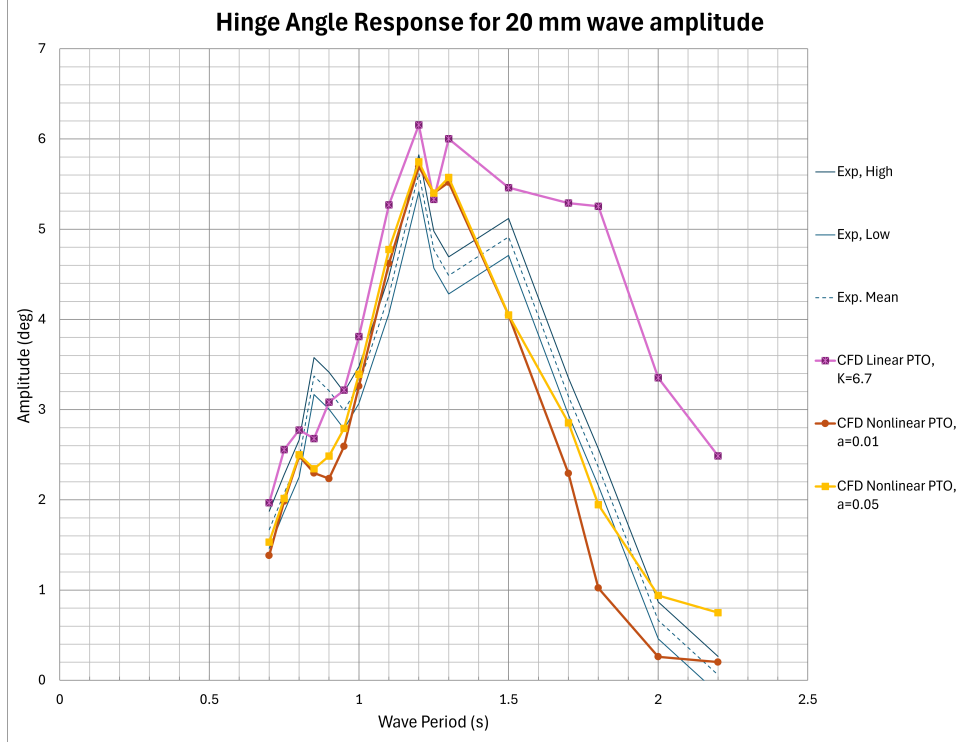


Figure 4.3: Comparison of hinge-angle amplitudes between the experimental measurements and the CFD models.

Figure 4.3 shows that all CFD models capture the overall trend of the experimental response, including the global peak around $T = 1.2$ s. In this region, the predicted amplitudes are comparable in magnitude to the experimental measurements, and the main peak response is well reproduced. However, the CFD models predict smaller local response peaks at slightly shorter wave periods than observed experimentally. This shift is observed in both the local peaks around $T = 0.8$ s and $T = 1.5$ s in the experimental response. Overall, this indicates that the models reproduce the main response level and peak well, but shift certain local response features toward shorter wave periods.

The linear PTO model generally predicts slightly higher amplitudes than the nonlinear formulations. This difference becomes more pronounced at longer wave periods, where the linear PTO model overpredicts the response, particularly beyond $T = 1.5$ s. Here, the nonlinear PTO model more closely follows the decreasing experimental trend in this region. This indicates that the nonlinear PTO representation improves agreement with experimental measurements at longer wave periods, where lower angular velocities make the transition behavior in the nonlinear PTO relation more influential.

The sensitivity to the nonlinear PTO bandwidth is relatively limited compared with the difference between the linear and nonlinear PTO representations. Before and around the peak, both nonlinear formulations produce nearly identical results, indicating that the response is not highly sensitive to the bandwidth parameter in this range. Beyond the peak, the bandwidth's influence becomes more noticeable, particularly for longer wave periods. For wave periods above approximately 1.5 s, the smoother transition with $a = 0.05$ yields slightly higher amplitudes than the steeper transition with $a = 0.01$. However, the difference is not monotonic and remains relatively small in absolute terms, with the largest offset being approximately 0.92° . This suggests that the bandwidth parameter primarily affects the low-amplitude region after the peak, while its influence on the peak response remains limited.

4.1.3 Overall Response Comparison

An overall comparison of the hinge-angle response is presented in Figure 4.4. The figure includes the experimental measurements, the CFD predictions obtained with the linear PTO formulation and the two nonlinear PTO formulations, as well as the previous frequency-domain model. This combined comparison provides a basis for assessing the relative influence of the hydrodynamic model and the PTO representation on the predicted WEC response.

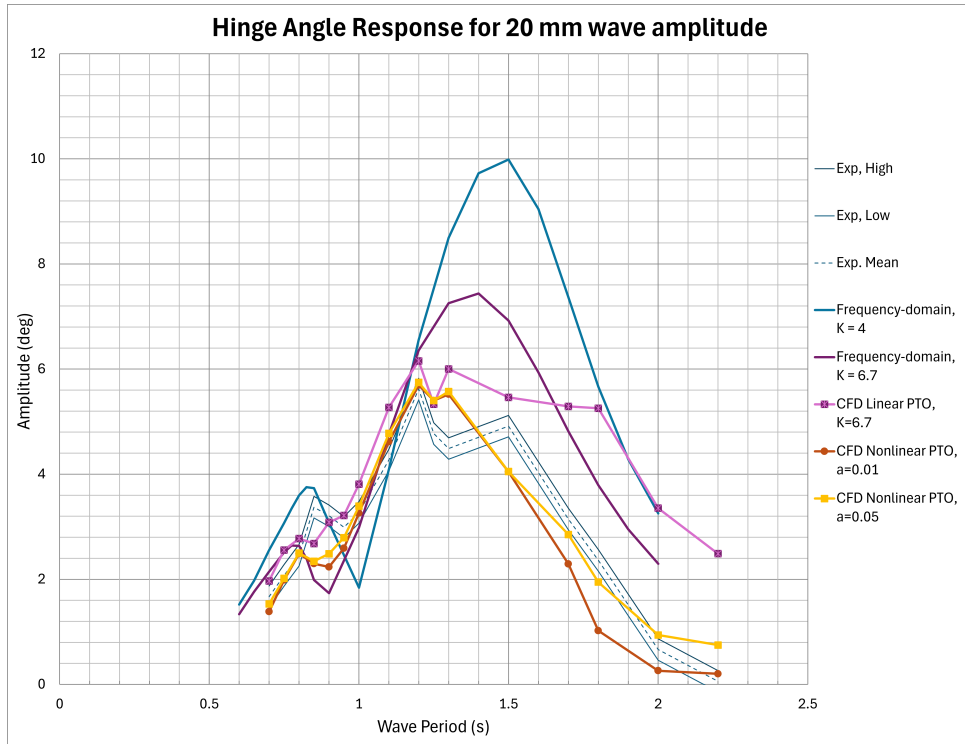


Figure 4.4: Overall comparison of hinge-angle amplitudes from the experimental measurements, CFD simulations, and the previous frequency-domain model.

The comparison shows that the predicted hinge-angle response is influenced by both the hydrodynamic modeling approach and the PTO representation. When the same effective linear PTO gain factor, $K = 6.7$, is used, the CFD models generally reduce

the overprediction of the hinge-angle amplitude around the peak response region. Since the PTO representation is equivalent in this comparison, the difference between the two predictions is primarily attributed to the hydrodynamic formulation. The CFD model accounts for effects such as viscous damping, nonlinear free surface behavior, and variations in wetted surface. These effects are either neglected or only approximately represented in the linear frequency-domain model, which helps explain why the CFD prediction is closer to the experimental response near the peak.

However, the improvement achieved with CFD is not uniform across all investigated wave periods. The CFD model with a linear PTO still overpredicts the response at longer wave periods, indicating that higher-fidelity hydrodynamic modeling alone is insufficient to reproduce the experimental measurements over the full response range. The nonlinear PTO formulations provide closer agreement in this region, indicating that the measured nonlinear behavior of the PTO system directly influences the global WEC response.

Overall, the best agreement is therefore obtained when the CFD model is combined with a PTO formulation based on the measured torque-angular velocity relationship. This demonstrates that reliable prediction of the hinge-angle response requires both a hydrodynamic model capable of capturing nonlinear and viscous effects and a PTO model that accurately represents the experimentally observed subsystem behavior.

4.1.4 Error Analysis

Error analysis was performed using the metrics defined in Section 3.3.3. Model predictions were compared with the mean experimental hinge-angle amplitudes at the wave periods for which data were available. For the nonlinear PTO model, error analysis was based on the best-fit formulation with bandwidth parameter $a = 0.05$. NRMSE was normalized by the range of the measured experimental hinge-angle amplitudes, and the resulting global error measures are summarized in Table 4.2.

Table 4.2: Global error measures for the hinge-angle response.

Model	Number of points	RMSE (°)	NRMSE (%)
CFD nonlinear PTO	16	0.6	10.1
CFD linear PTO	16	1.4	25.2
Frequency-domain model	15	1.5	26.7

The error measures show that the CFD model with the nonlinear PTO formulation gives the lowest overall error. This model yields an RMSE of 0.6° and an NRMSE of 10.1%, representing a clear improvement over both the CFD model using the linear PTO formulation and the previous frequency-domain model. In comparison, the CFD model with the linear PTO yields an NRMSE of 25.2%, whereas the frequency-domain model yields 26.7%.

It should be noted that the global error measures do not capture the full period-by-period variation in model accuracy. They provide a compact measure of the overall

agreement with the experimental data, but the local differences between the models may vary across the investigated wave periods. Nevertheless, the results clearly demonstrate the improved predictive capability achieved with the nonlinear PTO formulation. The complete hinge-angle amplitudes and absolute errors for each evaluated wave period are provided in Appendix C.

The peak hinge-angle response was also compared separately, since the maximum response is of particular interest when evaluating the predicted motion amplitude. The peak amplitude, the corresponding wave period, and the deviation relative to the experimental peak are summarized in Table 4.3.

Table 4.3: Comparison of peak hinge-angle amplitudes and percentage difference relative to the experimental peak.

Model	θ_{peak} ($^{\circ}$)	T_{peak} (s)	$\Delta\theta_{\text{peak}}$ ($^{\circ}$)	$\Delta\theta_{\text{peak}}$ (%)
Experiment	5.620	1.20	—	—
CFD nonlinear PTO	5.746	1.20	+0.126	+2.24
CFD linear PTO	6.156	1.20	+0.536	+9.54
Frequency-domain model	7.437	1.40	+1.817	+32.33

The results show that both CFD models predict the peak hinge-angle within 10% of the experimental value. The nonlinear PTO model gives the closest agreement, with an overprediction of $+0.126^{\circ}$, corresponding to 2.24%. The CFD model with the linear PTO coefficient $K = 6.7$ also captures the peak reasonably well, with an overprediction of 0.536° , corresponding to 9.54%.

In contrast, the frequency-domain model overpredicts the peak by 1.817° , corresponding to 32.33%. Both CFD models predict the maximum response at $T = 1.20$ s, which matches the wave period observed in the experimental results. The frequency-domain model instead predicts the peak at $T = 1.40$ s, indicating a shift of the maximum response towards longer wave periods. In summary, the CFD models provide more accurate predictions of the peak response magnitude and location.

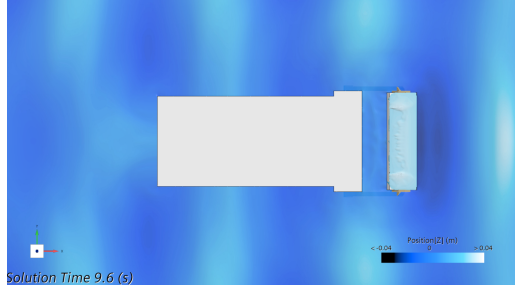
4.2 Visualization of the Simulated Response

This section presents visualizations of the simulated interaction between the waves and the structure for selected wave periods. The analysis first considers top-view visualizations of the free surface elevation and velocity field together with close-up views of the absorber motion. Side-view visualizations are then used to illustrate the device's motion.

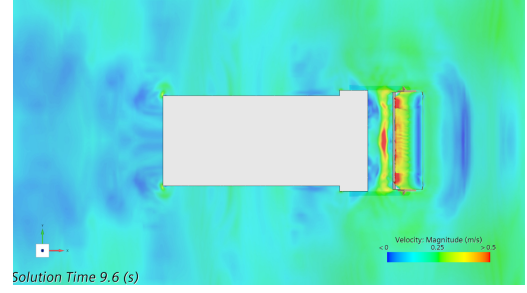
4.2.1 Top-View Comparison of Wave-Structure Interaction

For the best-fit nonlinear PTO model, additional visualizations of the free surface elevation and velocity field were generated for three representative wave periods. The shortest wave period was $T = 0.7$ s, an intermediate wave period, $T = 1.3$

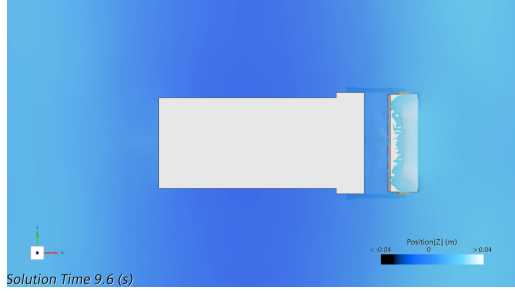
s, where the frequency-domain model showed one of its largest deviations, and the longest wave period was $T = 2.2$ s. For each wave period, snapshots of the free surface elevation and velocity field are taken at the same simulation time. The same physical quantities are shown for all cases, with consistent color bar limits to facilitate direct comparison across different wave periods. These cases illustrate how the wave field and the local flow around the WEC change as the wave period increases, as shown in Figure 4.5.



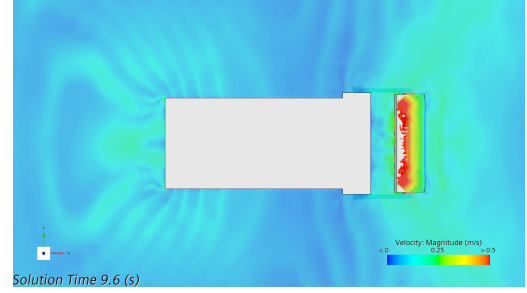
(a) Free surface elevation, $T = 0.70$ s.



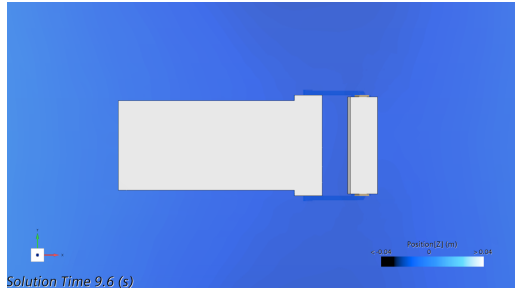
(b) Velocity field, $T = 0.70$ s.



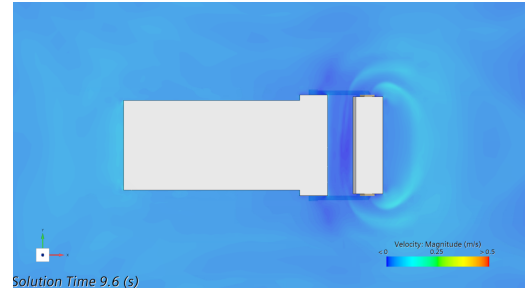
(c) Free surface elevation, $T = 1.3$ s.



(d) Velocity field, $T = 1.3$ s.



(e) Free surface elevation, $T = 2.2$ s.



(f) Velocity field, $T = 2.2$ s.

Figure 4.5: Top-view visualizations of the free surface elevation and velocity field for three representative wave periods.

The top-view visualizations show that the wave field is more compact for shorter wave periods than for longer ones. The velocity field is also stronger for the shorter and intermediate wave periods and decreases as the wave period increases. Furthermore, the results indicate that the absorber becomes partially submerged during short and intermediate wave periods, whereas it remains above the water surface during the longest wave period.

The comparison with the frequency-domain model suggests that some of the discrepancies observed in the previous model may be due to physical effects not fully captured by a linear potential-flow formulation. These effects include viscous damping, variations in wetted surface area, nonlinear free surface behavior, and possible submergence or green water effects. This may explain why the CFD model predicts the peak response more accurately than the frequency-domain model. During $T = 1.3$ s, water is observed on the absorber surface, violating some assumptions of the frequency-domain model. This behavior is further illustrated in Figure 4.6, which shows the temporal evolution of water overtopping the absorber.

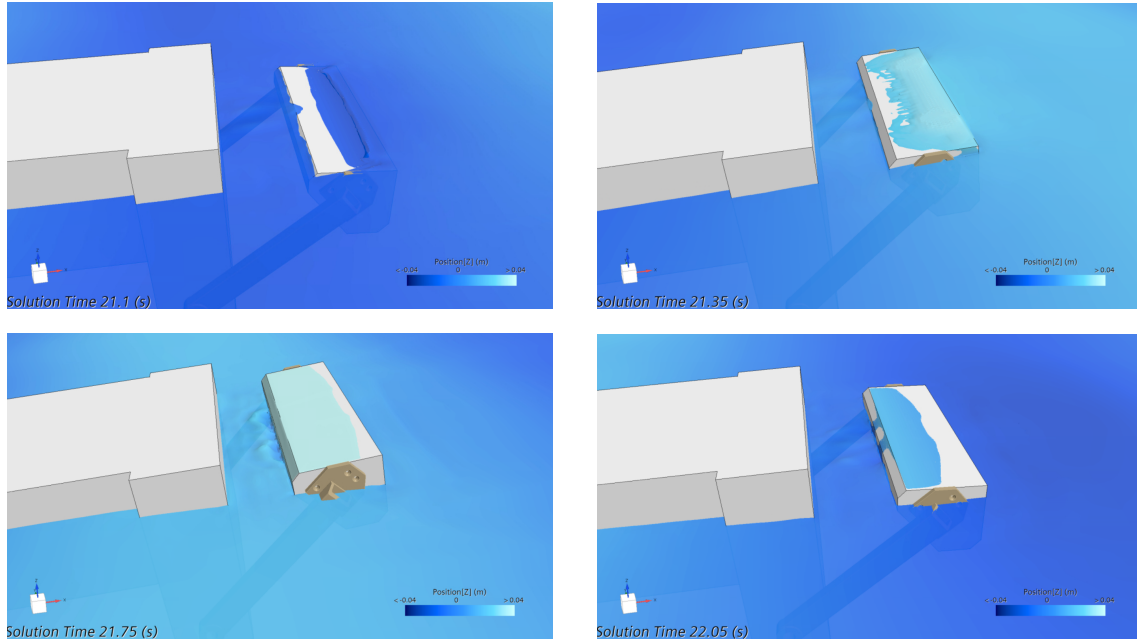


Figure 4.6: Sequence illustrating water overtopping the absorber for $T = 1.3$ s, highlighting the occurrence of green water effects during the wave cycle.

In contrast, the closer agreement of the frequency-domain model at longer wave periods may be partly explained by the reduced influence of submergence and green water effects in this range.

4.2.2 Side-View Motion Sequence

Figure 4.7 shows a side-view sequence of the articulated WEC for the representative intermediate wave period, $T = 1.3$ s. The sequence illustrates the device's motion between $t = 14$ and $t = 20$ s.

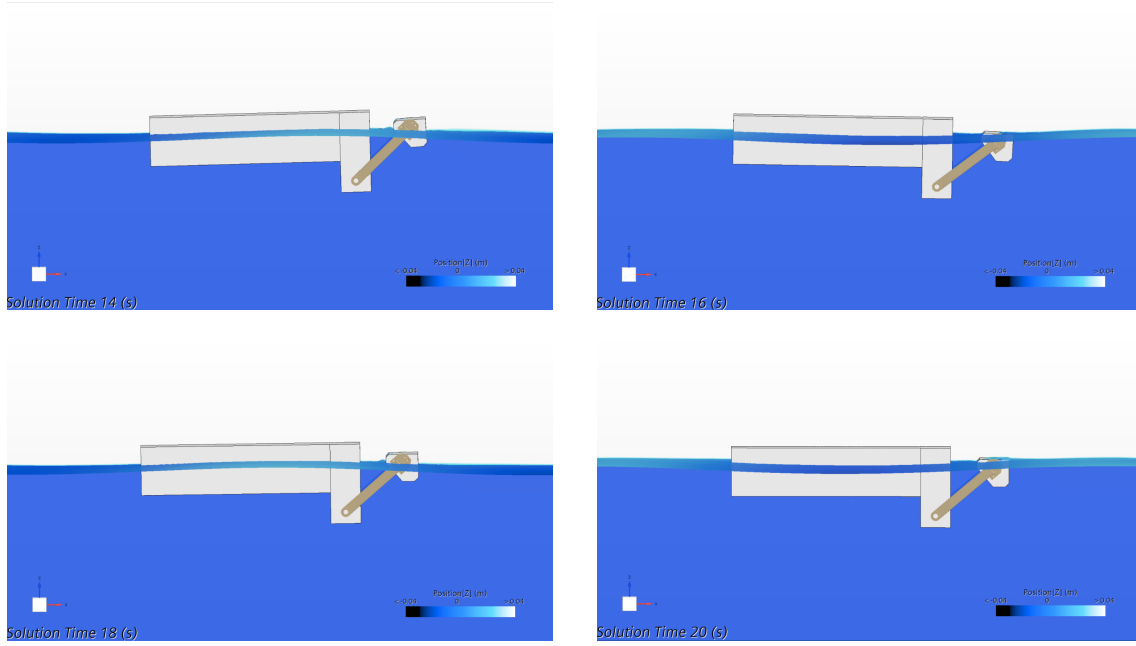


Figure 4.7: Visualization of the floater’s movement in waves for the selected simulation case.

The sequence shows the articulated WEC’s periodic motion as the incoming waves interact with the absorber. During parts of the wave cycle, water is observed on the absorber, indicating partial submergence and possible green water effects. This is consistent with the top-view visualizations, in which water was also observed on top of the absorber during intermediate wave periods.

4.3 Numerical Verification of the CFD Setup

Although the main objective of this study is to compare the predicted hinge-angle response with experimental measurements, numerical verification was performed to ensure that the selected CFD setup provided a reliable basis for this comparison. The verification focused on two aspects: the numerical wave tank’s ability to preserve the prescribed incident wave and the sensitivity of the predicted hinge-angle response to mesh and time step refinement.

4.3.1 Wave Elevation Validation

The wave elevation validation was performed without the WEC in the computational domain. The purpose was to evaluate whether the numerical wave tank could maintain the prescribed regular wave height as the wave propagated through the domain. The validation followed the mesh- and time step sensitivity procedure described in Section 3.2.2. Three representative wave periods were considered: the shortest wave period $T = 0.7$ s, an intermediate wave period $T = 1.5$ s, and the longest wave period $T = 2.2$ s. These cases were selected to cover the full range of

wavelengths used in the final simulations.

The generated wave height was evaluated by comparing the wave height near the inlet with the wave height at the intended WEC position. An example of this comparison for $T = 1.5$ s is shown in Figure 4.8.

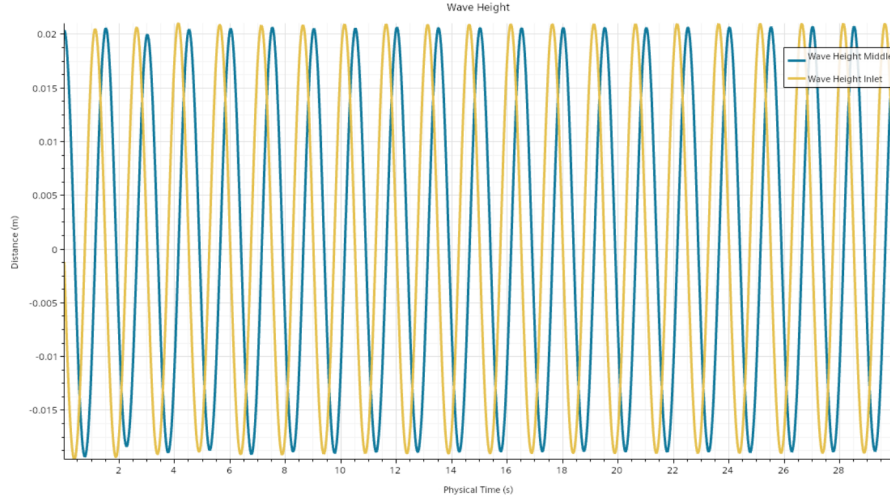


Figure 4.8: Comparison of the generated wave height near the inlet and at the intended WEC position for $T = 1.5$ s.

The wave tank mesh was selected through iterative refinement by varying the vertical free surface resolution, the horizontal resolution in the wave propagation direction, and the background mesh size. The selected values are summarized in Table 4.5.

After the wave tank mesh was selected, the influence of the time step size was evaluated. The time step size was found to have a stronger influence on the predicted wave height than further mesh refinement. The final time step values were therefore selected individually for the representative wave periods to keep the wave height error within approximately 5%, while maintaining a reasonable computational cost. The resulting wave height errors are summarized in Table 4.4.

Table 4.4: Wave height validation for the representative wave periods using the selected time step sizes.

Wave period (s)	Time step Δt (s)	H_{inlet}	H_{num}	Relative error (%)
0.7	0.001	19.833	18.919	-5
1.5	0.005	19.978	19.773	-1
2.2	0.010	20.001	19.369	-3

The numerical wave height remained within approximately 5% of the inlet wave height for all three representative wave periods. The largest deviation occurred for the shortest wave period, with wave height underpredicted by approximately 5%. The intermediate and long wave cases showed smaller deviations, with errors of approximately 1% and 3%, respectively. These results indicate that the selected

wave tank mesh and time step settings reproduced the incoming wave field with sufficient accuracy for subsequent WEC simulations. The detailed computational cost comparison used when selecting the wave tank time step is provided in Appendix A.

4.3.2 Summary of Full System Sensitivity

A full system sensitivity study was performed for the representative wave period $T = 1.5\text{s}$ using the hinge-angle amplitude as the primary response quantity. The study examined the effects of local hull refinement, surface refinement, and time step size. The detailed full system sensitivity studies are provided in Appendix A.

The final simulations used meshes containing approximately 3-4.3 million cells, depending on the simulated wave period. The increase in cell count with wave period was expected, since longer wavelengths required a larger computational domain and a larger free surface refinement region. The selected numerical settings for the final simulations are summarized in Table 4.5.

Table 4.5: Selected numerical settings for the final full-system simulations.

Parameter	Selected value
Cells per wave height (N_H)	10
Cells per wavelength (N_λ)	80
Free surface refinement	$\pm 1.5H$
Base mesh size	$16\Delta z$
Local hull refinement size	$2.0\Delta_{\text{wave}}$
Surface target size	$1.0\Delta_{\text{wave}}$
Final mesh size	3.0–4.3 million cells
Time step for full-system simulations	$0.75\Delta t_{\text{wave}}$

The full system sensitivity studies showed that the selected mesh and time step settings provided a suitable balance between numerical accuracy and computational cost. Further refinement had a limited influence on the predicted hinge-angle response compared with the differences observed between the PTO formulation and modeling approaches. The selected setup was therefore considered suitable for the comparison with the experimental measurements and the previous frequency-domain model.

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Conclusion

The CFD simulations were carried out using different representations of the Power Take-Off system. The linear model used a simplified linear formulation, while the nonlinear model was based on a fitted relation derived from the measured torque-angular velocity behavior. Both approaches reproduced the main characteristics of the experimental hinge-angle response, particularly the peak location and magnitude. Both the linear and nonlinear CFD models predicted the maximum hinge-angle response at the same wave period as the experiments and within 10% of the experimental peak amplitude. In contrast, the frequency-domain model overpredicted the peak by more than 30% and shifted the maximum response to a longer wave period than observed experimentally.

The CFD models provide a clear improvement over the frequency-domain model in the peak-response region. However, this improvement is not uniform across the full investigated wave period range. At longer wave periods, the linear model tends to overpredict the hinge-angle response. This indicates that the predicted response is strongly influenced by the PTO formulation, particularly away from the peak response region. The CFD results should therefore not be interpreted as uniformly more accurate for all wave periods. Instead, the main improvement of the higher-fidelity approach is its ability to capture the location and magnitude of the experimentally observed peak response more accurately than the frequency-domain model.

A comparison between the linear and nonlinear CFD models demonstrates that the PTO representation significantly influences the predicted response. The nonlinear CFD model gives the best overall agreement with the experimental measurements, with a global error of approximately 10%. This suggests that the representation based on the measured torque-angular velocity relationship provides a more representative prediction of the WEC response than the simplified linear formulation. The linear CFD model still captures peak response behavior reasonably well, but its overprediction at longer wave periods results in weaker overall agreement.

Overall, the findings indicate that CFD improves the prediction of the hinge-angle response compared with the previous frequency-domain approach, especially in the critical peak response region. At the same time, the results demonstrate that the predicted response is strongly influenced by the subsystem representation. These differences cannot be attributed to hydrodynamic modeling alone. With a more representative PTO model, even lower-fidelity modeling approaches may potentially produce results closer to the experimental measurements. The main conclusion is

5. Conclusion

therefore that reliable prediction of the WEC response requires both accurate hydrodynamic modeling and representative subsystem input data, particularly for the PTO system.

6

Future Work

Further work should prioritize a more detailed characterization of the PTO model, as the present results show that this formulation strongly influences the predicted hinge-angle response. Therefore, the measured torque and angular-velocity scatter data should be analyzed in greater detail to develop an improved PTO model. This model should then be implemented in both the CFD and frequency-domain models to enable a more consistent comparison between the two numerical approaches. Instead of using a single global PTO representation across all wave periods, future work could also investigate period-dependent formulations in which the PTO relation is fitted separately for each wave period. This may reduce the sensitivity to the selected fitting bandwidth and provide a more representative description of the system behavior for each operating condition.

The validation procedure should also include a more comprehensive quantitative assessment of additional response quantities, rather than focusing solely on the hinge-angle amplitude. Although other parameters were considered in this thesis, the validation focused mainly on the hinge-angle amplitude. Since subsystem modeling strongly influences the predicted response, relying primarily on a single response quantity provides a limited assessment of the model's accuracy. Future work could therefore include a more systematic comparison of body motions, hinge angular velocity, and absorbed power. This would provide a more complete evaluation of the system dynamics and help identify whether remaining discrepancies originate from the hydrodynamic model, the PTO system, the mooring system, or the geometry representation.

Moreover, further work should investigate the parameters and assumptions that were uncertain in the present model. Examples include the geometric simplifications, the representation of the protective box structure, and the representation of the mooring system. A dedicated sensitivity study of these parameters would determine the extent to which they influence the predicted hinge response. This would also support further refinement of both the CFD and frequency-domain models.

The influence of the absorber geometry on green water and submergence effects should also be examined in future work. In particular, it would be relevant to investigate whether increasing the absorber height could reduce water overtopping, thereby improving the device's response efficiency and predictability. A modified absorber design that limits these effects may also improve the applicability of lower-fidelity models, such as frequency-domain approaches. However, this should be

assessed through a dedicated geometry sensitivity study, since the remaining discrepancies may also be influenced by the PTO formulation, mooring representation, viscous damping, and other modeling approaches.

A dedicated scale effect study would also be valuable for assessing whether the model scale results remain representative of full-scale operation. Although Froude scaling may approximately preserve the dominant wave structure interaction, scale effects may still influence the full-scale response. This is particularly relevant to the PTO system, as the present results show that its formulation significantly influences the predicted hinge-angle response. However, it remains uncertain whether the sub-system would have the same relative influence at full scale, or whether the model scale PTO behavior can be scaled linearly. Future work should investigate how the hydrodynamic response, damping, and frictional losses change when the device is scaled up.

Finally, the model should be extended to more realistic operating conditions. The present study was limited to regular waves and focused on the hinge-angle response. Future work should therefore include irregular wave conditions, power production estimates, and additional performance metrics. This would make the model more representative of realistic sea states and provide a stronger basis for evaluating the wave energy converter's potential performance.

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A

Detailed Numerical Validation and Sensitivity Studies

This appendix presents the detailed numerical verification studies used to select the final wave tank and full system simulation settings. The studies include a computational cost comparison of wave tank time-step simulations, as well as full system mesh and time step sensitivity studies performed using the hinge-angle amplitude as the primary response quantity.

A.1 Wave-Tank Time Step Cost

The computational cost was considered when selecting the final time step sizes for the wave tank simulations. The estimated CPU time per simulated wave period is summarized in Table A.1.

Table A.1: Estimated computational cost per simulated wave period.

Wave period (s)	Time step Δt (s)	CPU time per wave period (min)
0.7	0.001	18.7
1.5	0.005	10.2
2.2	0.01	10.3
2.2	0.005	19.8

Table A.1 shows that reducing the time step size significantly increased the computational cost. For the $T = 2.2$ s case, reducing the time step from $\Delta t = 0.010$ s to $\Delta t = 0.005$ s nearly doubled the CPU time per simulated wave period. Since the selected time step of $\Delta t = 0.010$ still produced a wave height error within the target range, it was considered sufficient for the long wave case.

A.2 Local Hull Refinement

The full system mesh sensitivity study was performed for the representative wave period $T = 1.5$ s. The first study investigated the influence of the local hull refinement around the WEC. The percentage difference was calculated based on steady-state hinge-angle amplitudes relative to the finest case. The results are presented in Table A.2.

Table A.2: Effect of local hull refinement on the hinge-angle amplitude and computational cost for $T = 1.5$ s.

Case	Local hull refinement size	Difference from H3 (%)	$t_{\text{CPU},T}$ (min)
H1	$4.0 \Delta_{\text{wave}}$	-1.07	6.22
H2	$2.0 \Delta_{\text{wave}}$	-0.14	20.85
H3	$1.0 \Delta_{\text{wave}}$	0.00	38.82

The finest hull refinement case, H3, was used as the reference solution. The coarsest case, H1, gave a difference of -1.07% relative to H3, while the intermediate case, H2, reduced the difference to -0.14% . However, the computational cost increased substantially with increasing refinement, from 6.22 min per simulated wave height for H1 to 38.82 min for H3. The improvement from H2 to H3 was therefore small compared with the additional computational cost. Based on this result, the local hull refinement size $2.0 \Delta_{\text{wave}}$ was selected for the final simulations.

A.3 Surface Refinement

The second study investigated the influence of surface refinement on the WEC geometry. The results are shown in Table A.3.

Table A.3: Effect of surface refinement on the hinge-angle amplitude for $T = 1.5$ s.

Case	Local hull refinement size	Difference from S3 (%)	$t_{\text{CPU},T}$ (min)
S1	$4.0 \Delta_{\text{wave}}$	-8.85	7.23
S2	$2.0 \Delta_{\text{wave}}$	-0.19	21.08
S3	$1.0 \Delta_{\text{wave}}$	0.00	22.57

The coarsest surface mesh, S1, resulted in a relatively large difference of -8.85% , indicating that this resolution was too coarse to accurately predict the hinge-angle response. Refining the surface mesh to S2 reduced the difference to -0.19% . A further refinement to S3 only gave a small change in the response, while the computational time increased only slightly compared with S2. Since the surface mesh had a clear influence on the hinge-angle amplitude and since the additional cost of using S3 was limited, the finest surface target size was selected for the final mesh.

A.4 Cell Count

The final number of cells varied across different wave periods because several parameters, such as the domain size and the water refinement zone, were wavelength dependent. The resulting cell counts for the final simulations are shown in A.4.

Table A.4: Number of cells in the final mesh for the investigated wave periods.

Wave period T (s)	Number of cells
0.7	3 049 341
1.5	3 504 538
2.2	4 267 362

The number of cells increased with increasing wave period, from approximately 3.0 million cells for $T = 0.7$ s to approximately 4.3 million cells for $T = 2.2$ s. This increase was expected, since the longer wavelengths required a larger computational domain and larger free-surface refinement region.

A.5 Full-System Time Step Sensitivity

Moreover, a time step sensitivity study was performed on the complete WEC model. The results are presented in Table A.5.

Table A.5: Time step sensitivity of the hinge-angle response for $T = 1.5$ s.

Case	Time step	Difference from finest case (%)	$t_{\text{CPU},T}$ (min)
TG1	Δt_{wave}	6.68	7.916
TG2	$0.75 \Delta t_{\text{wave}}$	-0.84	7.234
TG3	$0.50 \Delta t_{\text{wave}}$	0.00	22.566

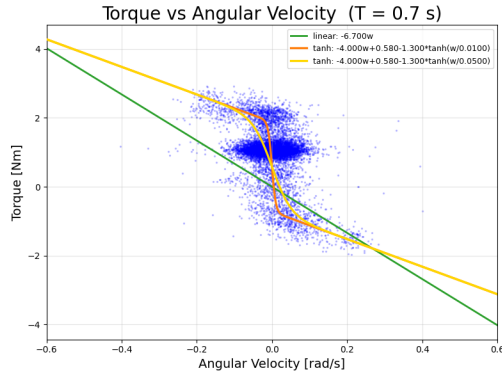
The coarsest time step case, TG1, showed a difference of 6.68% relative to TG3, indicating that the original time step from the wave tank validation was too coarse to accurately resolve the hinge-angle response of the full WEC system. Reducing the time step to $0.75 \Delta t_{\text{wave}}$ reduced the difference to -0.84% . Although the convergence was not strictly monotonic, the difference between TG2 and TG3 was less than 1 %, while TG3 required substantially higher computational cost. Based on this, $0.75 \Delta t_{\text{wave}}$ was selected for the final simulations as a compromise between accuracy and computational efficiency.

B

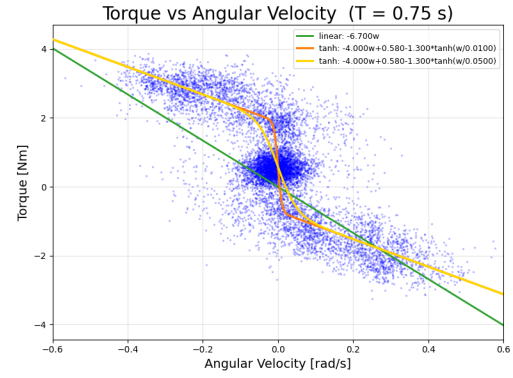
Experimental PTO

Torque–Angular-Velocity Data

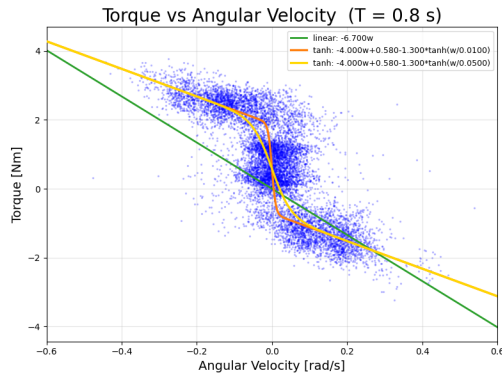
Figure B.1 shows the angular velocity and torque for each wave period, based on the experimental towing-test data. The complete scatter plot is presented in the main report, while the individual plots shown here are derived from the corresponding input data for each wave period. The green line represents the fitted linear PTO model with $K=6.7$, whereas the orange and yellow lines represent the nonlinear fits with bandwidths of 0.01 and 0.05.



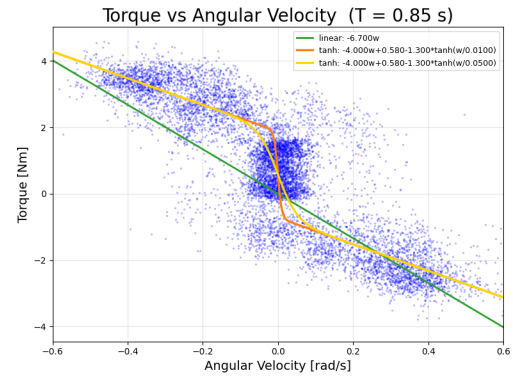
(a) $T = 0.70$ s.



(b) $T = 0.75$ s.

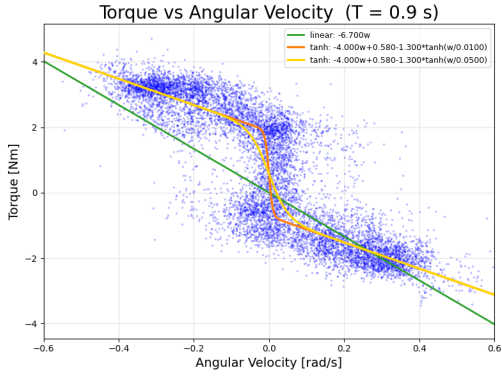


(c) $T = 0.80$ s.

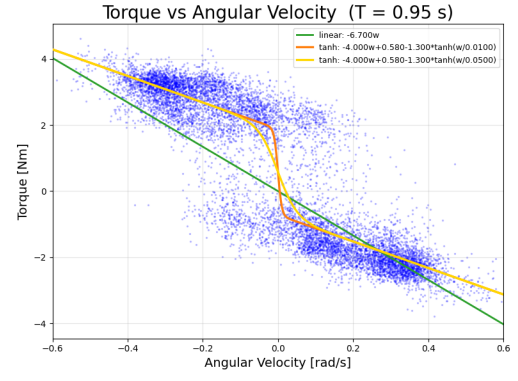


(d) $T = 0.85$ s.

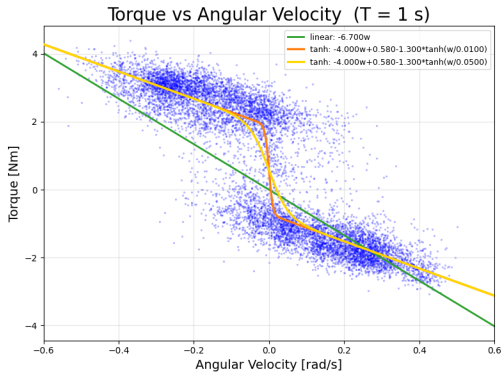
B. Experimental PTO Torque–Angular-Velocity Data



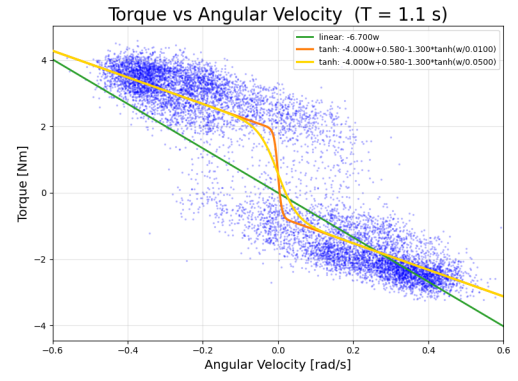
(e) $T = 0.90$ s.



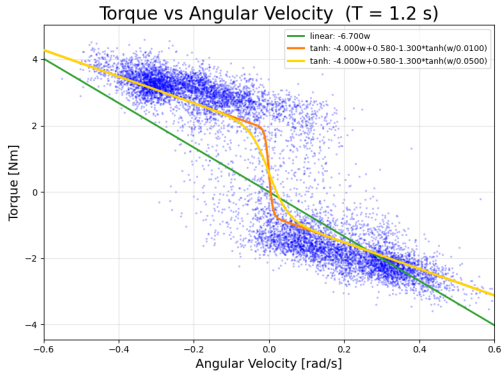
(f) $T = 0.95$ s.



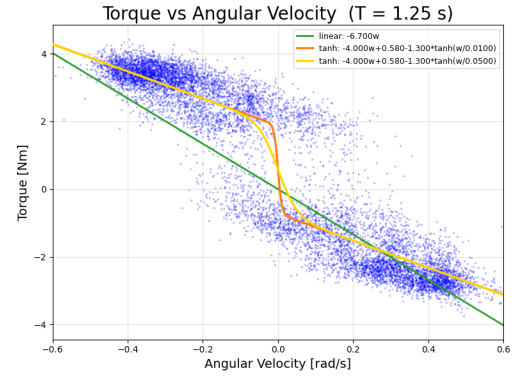
(g) $T = 1.00$ s.



(h) $T = 1.10$ s.

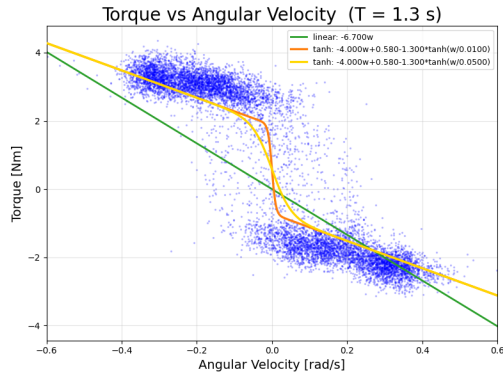


(i) $T = 1.20$ s.

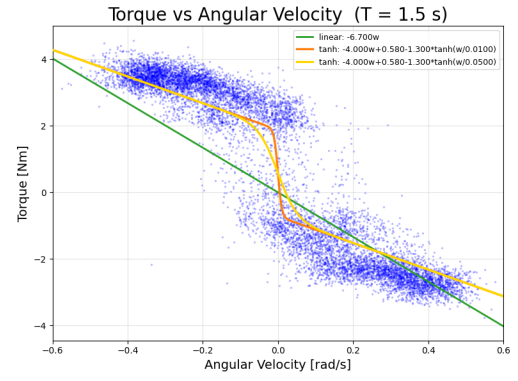


(j) $T = 1.25$ s.

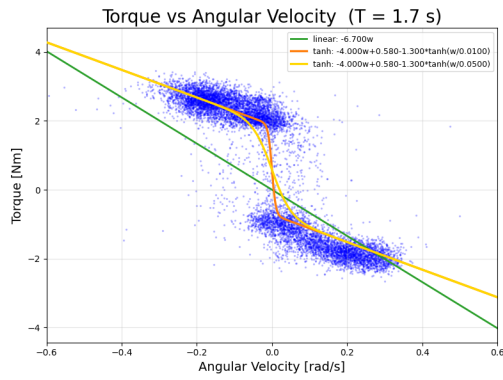
B. Experimental PTO Torque–Angular-Velocity Data



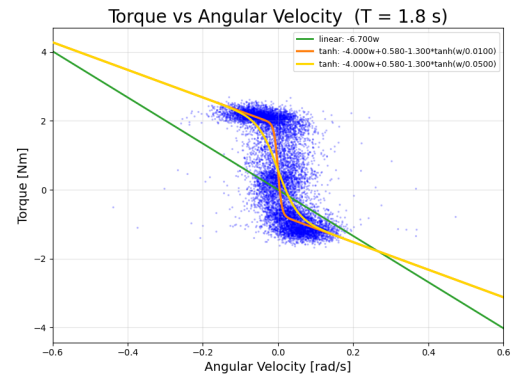
(k) $T = 1.30$ s.



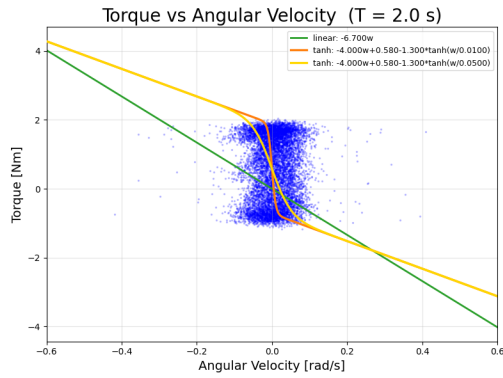
(l) $T = 1.50$ s.



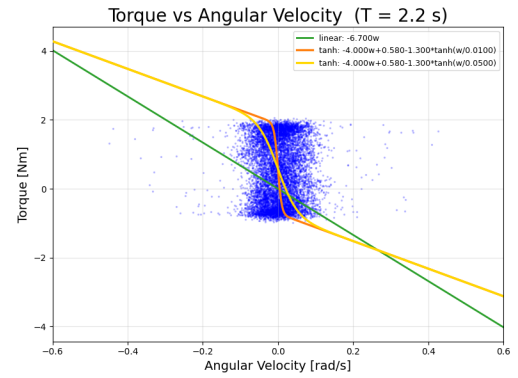
(m) $T = 1.70$ s.



(n) $T = 1.80$ s.



(o) $T = 2.00$ s.



(p) $T = 2.20$ s.

Figure B.1: Angular velocity and torque for different wave periods based on experimental towing-test data.

C

Comparison of Hinge-Angle Amplitudes and Absolute Errors

The hinge-angle amplitudes used for the overall comparison are summarized in Table C.1. The fitted PTO model is the nonlinear PTO formulation with bandwidth parameter $a = 0.05$, selected based on the bandwidth sensitivity analysis in Section 4.1.2. The absolute errors with respect to the experimental measurements are presented in Table C.2. Missing entries correspond to wave periods for which no simulation or model results were available.

Table C.1: Comparison of hinge-angle amplitudes from experiments, CFD models, and the previous frequency-domain model.

T (s)	Experiment (°)	CFD, $K = 6.7$ (°)	CFD, fitted PTO (°)	Frequency-domain model (°)
0.70	1.668	1.968	1.532	2.147
0.75	2.075	2.557	2.017	2.512
0.80	2.459	2.774	2.500	2.635
0.85	3.372	2.680	2.344	1.986
0.90	3.209	3.083	2.485	1.735
0.95	2.989	3.215	2.792	—
1.00	3.273	3.810	3.392	2.982
1.10	4.271	5.271	4.775	4.765
1.20	5.620	6.156	5.746	6.355
1.25	4.774	5.331	5.400	—
1.30	4.488	6.003	5.573	7.252
1.40	5.276	—	—	7.437
1.50	4.914	5.460	4.049	6.923
1.60	3.968	—	—	5.936
1.70	3.146	5.280	2.852	4.815
1.80	2.362	5.254	1.947	3.790
2.00	0.664	3.353	0.941	2.296
2.20	0.060	2.488	0.750	—

Table C.2: Absolute error in hinge-angle amplitude with respect to the experimental measurements.

T (s)	CFD, $K = 6.7$ error ($^{\circ}$)	CFD, fitted PTO error ($^{\circ}$)	Frequency-domain model error ($^{\circ}$)
0.70	0.300	0.136	0.479
0.75	0.482	0.058	0.437
0.80	0.315	0.041	0.176
0.85	0.692	1.028	1.386
0.90	0.126	0.724	1.474
0.95	0.226	0.197	—
1.00	0.537	0.119	0.291
1.10	1.000	0.504	0.494
1.20	0.536	0.126	0.735
1.25	0.557	0.626	—
1.30	1.515	1.085	2.764
1.40	—	—	2.161
1.50	0.546	0.865	2.009
1.60	—	—	1.968
1.70	2.134	0.294	1.669
1.80	2.892	0.415	1.428
2.00	2.689	0.277	1.632
2.20	2.428	0.690	—

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