



Xing Jingru

Enhancing Stationkeeping and Motion Reduction of Floating
Offshore Wind Turbines using Wave Devouring Propulsion
Technology

School of Water, Energy and Environment
Energy and Sustainability

PhD

Academic Year: 2020–2024

Supervisor: Dr Yang Liang

Associate Supervisor: Dr Lao Liyun

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Abstract

This PhD thesis investigates the enhancement of stationkeeping and motion reduction in Floating Offshore Wind Turbine (FOWT) platforms through the application of biomimetic-inspired Wave Devouring Propulsion (WDP) technology. Through a multidisciplinary approach that encompasses a comprehensive literature review, numerical simulations, and scaled-down water tank experiments, this research confirms the potential of WDP technology to significantly improve the stationkeeping capabilities of FOWT platforms. The thesis is structured around four key objectives, each addressing a critical aspect of WDP application in FOWT platforms—from theoretical underpinnings and simulation tool development for Fluid-Structure Interaction (FSI) analyses, to the investigation of sub-structure dynamics and practical feasibility studies for the integration of innovative sub-structures aimed at stationkeeping. This thesis contributes to the field by providing a comprehensive overview of WDP technology, introducing a novel numerical model for FSI simulations, insights into the hydrodynamic performance of submerged hydrofoils, and the experimental verification of WDP technology's effectiveness in enhancing platform stationkeeping. Notably, the study proposes optimized design guidelines for foil implementation, demonstrating a significant reduction in mooring tension and thereby advancing the practical applicability of WDP in the marine industry. By advancing the understanding and application of WDP technology, this thesis lays the groundwork for significant improvements in the sustainability and efficiency of maritime operations, aligning with global efforts towards achieving net-zero emissions in the maritime sector.

Keywords:

Flapping foil; Marine environment; Computational Fluid Dynamics (CFD); Fluid-Structure Interaction (FSI); Platform stabilisation.

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List of Abbreviations

AoA	Angle of attack
ASV	Autonomous surface vehicle
AUV	Autonomous underwater vehicles
BDIM	Boundary data immersion method
BEM	Boundary element method
BIM	Boundary integral method
CAD	Computer-aided design
CFD	Computational fluid dynamics
CFL	Courant-Friedrichs-Lewy
DOF	Degree of freedom
DP	Dynamic positioning
EEDI	Energy efficiency design index
FAWPD	Floater-adjusted wave propulsion device
FFT	Fast Fourier Transform
FLEUR	Flapping energy utilisation and recovery
FOWT	Floating Offshore Wind Turbine
FSI	Fluid-structure interaction
FVM	Finite volume method
FW-H	Ffowcs Williams-Hawkings
GPU	Graphics processing unit
ITOPF	International Tanker Owners Pollution Federation
LCCA	Life cycle cost analysis
LCOE	Levelized Cost of Energy
LE	Leading edge
LSM	Level Set Method
NACA	National advisory committee for aeronautics
NSV	Natural-energy-driven surface vehicle
NWT	Numerical wave tank
O&M	Operation and maintenance
PV	Photovoltaics
RPM	Revolutions per minute
RvK	reversed von Kármán vortex street
SALL	Self-adjusting lower limit
SAWG	Single-body architecture wave glider
SHM	Simple harmonic motion
SST	Shear stress transport
SWL	Still water level

TAPM	Thruster-assisted position mooring
TE	Trailing edge
TIWTE	Tianjin Research Institute for Water Transport Engineering
TOMOT	Team of ocean mobile observation technology
TVD	Total Variation Diminishing
UDF	User defined function
USV	Unmanned surface vehicle
vK	von Kármán vortex street
VOF	Volume of fluid
WAFT	Wave augmented foil technology
WDP	Wave devouring propulsion
WDPS	Wave devouring propulsion system
WEC	Wave energy converter

(x_{TE}, y_{TE})	Location of the TE
$\alpha_{eff}(t)$	Instantaneous effective AoA at time t
β	Ratio of the pitching frequency to the heaving frequency
$\Delta\theta$	Rotational deviation
ΔL	Extended length of the tension spring
Δy	Vertical deviation
λ	Wavelength
μ	Dynamic viscosity of the fluid
ω	Angular frequency
ϕ	Phase difference between the pitch and heave motion
ψ	Phase difference between wave and foil motion
ρ	Density of the surrounding flow
θ_0	Pitch amplitude
$\theta(t)$	Instantaneous angle between the incoming flow and the chord at time t
A	Peak-to-peak oscillating amplitude of the foil's TE
c	Foil chord length
C_l	Lift coefficient
C_t	Thrust coefficient
C_P	Power coefficient
$\bar{C}$	Time-averaged coefficient
D	Foil thickness
$\partial D_B(t)$	Moving boundary to represent the flapping foil
∂D_F	Moving boundary to represent the free surface
∂D_W	Moving boundary to represent the vortex wake
f	Oscillating frequency
$f_{encounter}$	Encounter wave frequency
f_n	Foil natural frequency
$F_h(t)$	Heave force
F_{ts}	Frequency ratio for torsional stiffness in pitching motion
F_L	Lift force
F_T	Thrust force
$\bar{F}$	Time-averaged force
h_0	Heave amplitude
$h(t)$	Instantaneous vertical displacement of the foil at time t
$\dot{h}(t)$	Instantaneous heave velocity at time t
I	Foil inertia moment
k	Reduced frequency
k_e	Pitch tension spring stiffness
k_h	Heave spring stiffness
k_p	Pitch torsion spring stiffness
k_s	Parameter for adjusting heave perturbation
K^*	Non-dimensional torsion stiffness
L_4	Arm length of the tension force
M_p	Pitch moment
$M_{pte}(t)$	Pitch tension moment
$M_{pto}(t)$	Pitch torsion moment

P	Pivot point
Re	Non-dimensional Reynolds numbers
s	Foil span
Sr_c	Chord-length-based Strouhal number
Sr_D	Thickness-based Strouhal number
St_A	Amplitude-based Strouhal number
St_T	Path-length-based Strouhal number
t	Time
T	Wave period
T_{foil}	Trajectory length covered by the TE at the end of one oscillation period
u	Foil horizontal velocity
U_{wake}	Velocity of the wake
U_∞	Velocity of the incoming flow
v	Foil vertical velocity

Chapter 1

Introduction

1.1 Background

The escalating concerns over global warming have spurred a pressing need to transition from fossil fuels to sustainable energy sources. In this context, green energy solutions, especially those leveraging marine resources, have become key to our efforts in mitigating the impacts of global warming. Among the various kinds of renewable energy sources, ocean waves represent a vastly untapped resource. In fact, the oceans, which cover 71% of the Earth's surface (1), contain vast amounts of energy within their waves. For example, the UK's total wave resource is around 230 TWh/year (2), which, if fully utilised, could contribute more than 70% of the current electricity generation in the UK. Currently, technologies like Floating Offshore Wind Turbines (FOWTs) are deployed in marine environments, closely interacting with ocean dynamics. However, despite their proximity to this immense energy resource, FOWTs primarily harness wind power, leaving the vast wave energy largely unexploited.

Offshore wind energy plays a pivotal role in the global transition towards net-zero carbon emissions. By the end of 2022, the worldwide operational capacity of FOWTs had reached 64.3 GW, highlighting the rapid development within this sector to meet increasing energy demands sustainably (3). The sector is set on a path of ambitious growth, with plans to add more than 380 GW of offshore wind capacity across 32 markets by 2032 (3). This expansion is critical to achieving the ambitious targets set by various governments, including the UK's goal of achieving 50 GW of floating wind capacity by 2030 (3, 4).

Despite this optimistic outlook, the deployment of FOWTs faces significant obstacles, primarily due to the instability of offshore platforms under the influence of remote sea waves. This instability not only complicates operations and maintenance (O&M) activities but also increases the Levelized Cost of Energy (LCOE), making floating wind less competitive compared to its fixed-bottom counterparts (5). Against this backdrop, Wave Devouring Propulsion (WDP) emerges as a promising solution, offering a novel approach to enhancing platform stability and operational efficiency in the marine environment.

The WDP is a biomimetic design that harnesses the natural advantages found in aquatic animals. Inspired by the efficient hydrodynamics of whale fins, which employ foils to convert wave energy into propulsion (6). The concept of WDP represents an innovative green technology within the marine sector. Initially conceptualized in the mid-20th century (7), WDP has progressed through extensive theoretical, experimental, and

numerical research (8) to demonstrate its effectiveness in reducing wave resistance and alleviating vessel motions such as heave and pitch. The ability of WDP in improving propulsion efficiency while reducing fuel consumption establishes it as an eco-friendly option for the marine industry, highlighting its significant potential to aid in the transition to green energy.

The practical applications of WDP, as evidenced in ships like the Suntory Mermaid II (9) and various Unmanned Surface Vehicles (USVs), including the Wave Glider by Liquid Robotics and AutoNaut by AutoNaut, highlight its commercial viability and the benefits it brings to marine navigation and vessel efficiency. Despite these successes, the application of WDP in stabilising FOWTs represents a relatively unexplored area with considerable potential for innovation and development.

1.2 Aim and objectives

This study aims to

“To implement an innovative technology on semi-submersible platforms, to reduce carbon emissions and fuel usage in maintaining stationkeeping of FOWTs under severe marine conditions”.

To be more specific, this research seeks to explore a clean energy technology, ideally marine-related, capable of enhancing the stationkeeping of FOWTs under extreme wave conditions. The goal is to find a solution that can support or replace existing technologies, with the aim of extending the lifespan of FOWTs, reducing O&M costs, enhancing power generation efficiency, and alleviating or even substituting the economic burdens associated with mooring lines. By achieving these, the study hopes to not only improve the operational viability of FOWTs but also contribute to global sustainability efforts.

To achieve the stated aim, research objectives (I-IV) were devised. These are described as follows:

Objective I:

- To conduct a comprehensive literature review on the proposed innovative technology for FOWTs, summarising its potential advantages and foundational concepts.

Objective II:

- To develop a simulation tool for Fluid-Structure Interaction (FSI) analyses, tailored to evaluate the dynamics of the innovative technology under various marine conditions.

Objective III:

- To investigate the mechanisms of the proposed technology, focusing on the dynamics of sub-structures (e.g., foils) within a viscous numerical wave tank, to understand their interaction with wave forces.

Objective IV:

- To assess the practical applicability of incorporating the innovative sub-structure for FOWT stationkeeping, with an aim to provide a feasibility study and optimization strategies.

Fig.1.1 illustrates the relationship between the four objectives and the main structure of the thesis. The approach involves initial 2D flapping foil modelling in the wave tank to establish fundamental principles, which then inform and support subsequent 3D experiments.

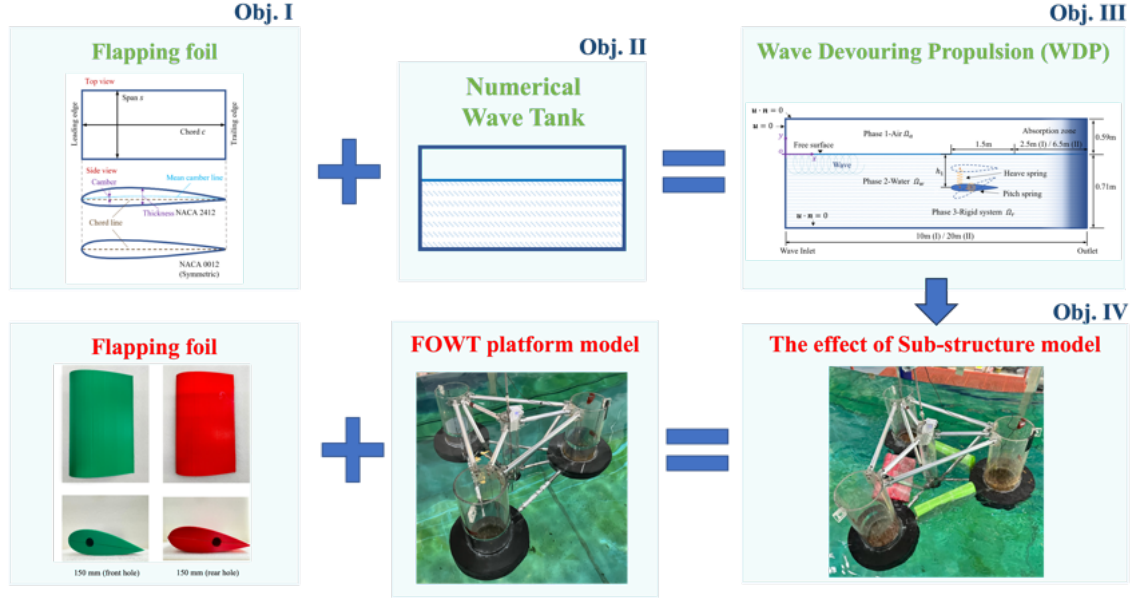


Figure 1.1: Conceptual framework illustrating the interconnection of the four objectives aimed at enhancing FOWT stabilisation technology.

1.3 Thesis structure

This thesis is comprised of a series of chapters (Fig.1.2), which were formatted as journal papers (published, submitted or in preparation) that address each one of the set objectives.

Chapter 2, corresponding to Objective I, provides a comprehensive review of the current advancements in flapping foil technology within Wave Devouring Propulsion (WDP). It identifies the gaps between this technology and offshore platforms. This analysis synthesizes existing literature and practical applications, examining WDP's operational aspects such as wave conditions, foil configurations, degrees of freedom, submergence depth, and surface effects, while highlighting current challenges and suggesting directions for future research in WDP.

Chapter 3, corresponding to Objective II, introduces a detailed three-phase Fluid-Structure Interaction (FSI) framework to evaluate the dynamics of the interaction between solid structures and ocean conditions. This framework lays the groundwork for Objective III, primarily serving as validation to demonstrate the effectiveness of the code. The chapter uses the well-known Simple Harmonic Motion (SHM) as a benchmark to validate the numerical spring's accuracy. By comparing experimental simulations of wedge dropping and oil boom response, this chapter confirms the feasibility of the numerical platform in multi-phase CFD simulations, establishing a solid foundation for subsequent chapter.

Chapter 4, corresponding to Objective III, investigates the mechanism of WDP, specifically the propulsion generated by flapping foils under wave. Utilizing the previously de-

veloped CFD framework, simulations in a numerical wave tank observe the generation of wake and the transformation from drag to thrust. Initial validation through established benchmarks and experimental data with the foil in uniform flow confirms the model's robustness and accuracy. The study further examines the impact of wave conditions and pitch stiffness on thrust generation, identifying conditions that significantly improve efficiency.

Chapter 5, corresponding to Objective IV, assesses the feasibility and potential of WDP for offshore platform stationkeeping. Experiments with a scaled model under various wave conditions, in both free-floating and moored states, were conducted to determine the optimal design. The investigation focuses on the influence of foil quantity, placement, pitch angle, and offset positioning. It further proposes an optimal integrated foil design for platforms and verifies its performance in reducing mooring line tension.

Chapter 6 summarises the PhD's findings, contributions, and future research opportunities, offering a cohesive conclusion to the research journey.

Chapter	Objective	Focus	Title	Status
1	-	Introduction	Introduction	-
2	I	Literature Overview	Wave devouring propulsion: an overview of flapping foil propulsion technology	Published <i>Renewable and Sustainable Energy Reviews</i> , 184, 113589.
3	II	Numerical Methodology	Advancing fluid-structure interaction (FSI) simulations in 2D: Insights from a three-phase model	Published <i>Ocean Engineering</i> , 278, 114356.
4	III	WDP	Numerical investigation of wave induced thrust on a submerged hydrofoil	Published
5	IV	WDP+Platform	Stabilisation of Floating Wind Platforms with Wave Energy: Lab Evidence	Under review
6	-	Discussion and conclusion	Discussion and conclusion	-

Figure 1.2: Thesis structure.

1.4 References

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Chapter 2

Wave devouring propulsion: an overview of flapping foil propulsion technology

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2.1 Abstract

A comprehensive review of flapping foils for Wave Devouring Propulsion (WDP) is presented. The flapping foil can effectively utilise wave energy and generate thrust. The development of WDP is discussed, followed by an introduction to the geometry, modes of motion, and operating principles. These research studies are classified as theoretical, experimental, and numerical and are provided in detail. They demonstrate that marine equipment with a flapping foil system can achieve high energy conversion efficiency and low resistance. Several prototypes of the combination of WDP with human-crewed and uncrewed vessels have been shown, including the latest initial concept models and company products. There is a huge prospect for self-driven, pollution-free propulsion of marine devices, and this paper suggests several future studies.

Keywords: Flapping foil; Wave Devouring Propulsion (WDP); Hydrodynamics; Fluid-Structure Interaction; Wave-powered propulsion; Biomimetic thrusters.

2.2 Introduction

The ocean waves have a tremendous amount of energy. The flapping foil, as a wave propulsion device, can use the kinetic energy of the waves to achieve self-propulsion without fuel. This green propulsion technique converts wave energy into thrust using

submerged flapping foils called Wave Devouring Propulsion (WDP). The human-crewed and uncrewed ships can be powered with WDP and achieve higher propulsion efficiency. The main purpose of this paper is to provide a comprehensive summary of the research on wave-induced flapping foils.

Flapping foils have been of great interest for a long time. The ability to generate thrust from surrounding flows was discovered and demonstrated as early as 1909. Knoller (1), and Betz (2) first explained the mechanism of thrust generation by a heaving foil in a uniform flow independently. The Knoller-Betz effect was verified experimentally by Katzmayer (3), who demonstrated that a foil could generate thrust in an oscillating flow and also assumed that this has the same effect as an oscillating foil in a uniform flow. For the uniform flow, McKinney and DeLaurier (4) tested the fluid as air instead of water. A windmill was analytically and experimentally investigated, which uses flapping airfoils to extract wind energy.

The phenomenon of propulsion is commonly observed in nature. The flight of birds and the forward movement of fish rely on the lift and thrust of their foil-shaped bodies. Lighthill (5) studied the biofluid dynamic behaviour of aquatic animal propulsion modes and concluded that the formation of the jet-like wake, also known as the reverse Von Karman vortex street (6), is an important phenomenon in the generation of thrust. This idea was subsequently confirmed by Koochesfahani et al. (7), Jones et al. (8), Buchholz et al. (9), Godoy-Diana et al. (10) and Lagopoulos et al. (11).

Flapping foils can also enhance traditional propulsion methods, which leads to the development of the wave devouring propulsion system (WDPS). It often utilises a spring-loaded foil mounted under the bow or stern of the vessel. Early experiments by Jakobsen (12) and Terao (13) conducted the feasibility of wave powering ships. Subsequently, further research through experiments and numerical simulations has been conducted to deepen the understanding. Commercial products have been developed, such as the Wave Glider by Liquid Robotics (14), Autonomous Surface Vehicles by AutoNaut (15, 16), and the M/F Teistin by Wavefoil, equipped with retractable bow foils (17).

In addition, WDPS operate at a lower frequency compared to traditional marine propellers, which can reduce the generation of underwater noise and its adverse effects on sea life, particularly marine mammals. Several numerical studies (18, 19, 20, 21, 22) used the acoustic wave equation Ffowcs Williams-Hawkings (FW-H) to understand the radiated noise levels. Belibassakis et al. (22) proposed that the hull geometry and acoustic properties of the vessel surface, as well as the effects of the sea surface and seabed, are crucial in determining the directionality of the generated noise.

Fig.2.1 illustrates the number of studies using theoretical, experimental and numerical approaches since 1954.

This review aims to present a comprehensive summary of the existing research and industrial applications related to wave-powered foil propulsion and provide directions for future work in this field. The paper is structured as follows: In Section 2.2, we provide a brief introduction to foil propulsion and WDP technology. Section 2.3 delves into the key parameters related to foil study, including information on geometry, motion modes, and dimensionless parameters. The theoretical, experimental, and numerical research on WDP is concluded in Sections 2.4, 2.5, and 2.6 respectively. Our aim is to provide an objective and comprehensive overview of the milestones and directions, focusing on wave-induced thrust efficiencies, such as the degrees of freedom of the foil, the relative

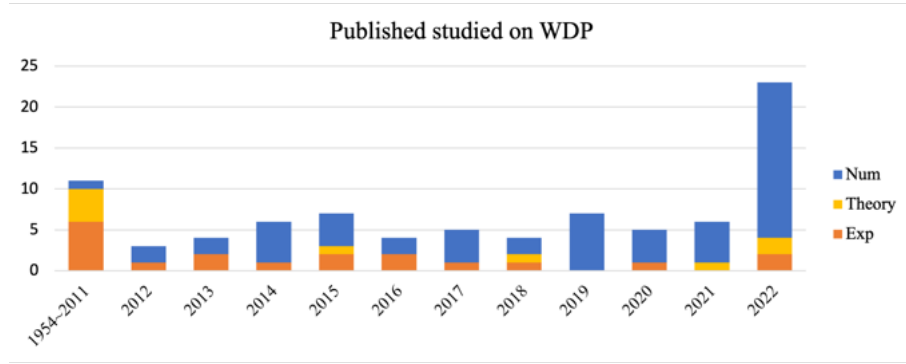


Figure 2.1: The number of papers on WDP since 1954.

position of the foil to the attached body, the free-surface effect, etc. In Section 2.7, we have conducted a comprehensive summary of the industrial development of WDP devices in table format, highlighting practical applications, products, and prototypes of foils and ships, foils and Unmanned surface vehicles (USVs), and foils and other connections. Finally, Section 2.8 suggests directions for future research.

2.3 Geometry and parameters

2.3.1 Foil Geometry

The design and geometry of the flapping foils are crucial for the performance of WDPs. One of the most commonly studied cross-sectional foil shapes is the NACA 4-digit foils, particularly the NACA0012 (23). Other NACA models have also been used, such as NACA0015 (24, 25, 26). The term “NACA” stands for National Advisory Committee for Aeronautics, and the numbers following it describe the profile of the foil. In the case of NACA0012, the first two digits, “00”, indicate that the foil has no camber and is symmetrical, while the last two digits, “12”, indicate that the foil has a ratio of 12% thickness to chord length. NACA foils are widely used in aerospace and marine engineering due to their systematic development, which provides standardized, reliable, and well-documented aerodynamic properties. Their proven performance, versatility, and extensive data make them essential for predictable outcomes and informed decision-making. It is noticed that in WDP, symmetric foils are often used. Although NACA, elliptical, and rectangular cross-sections have been studied, asymmetric shapes have not been widely explored. This is because WDP research has only recently become popular, making symmetric shapes a more intuitive starting point. The propulsion mechanism relies on symmetric foils to generate reverse Kármán vortices and thrust, and while asymmetric shapes may also be effective, they have not been thoroughly investigated. Moreover, underwater foils focus on stable thrust rather than lift, and asymmetric designs might produce uneven wake thrust, potentially affecting stability. However, further research into asymmetric designs is still needed to fully understand their potential.

Fig.2.2 illustrates the characteristics of symmetrical and asymmetrical foils. The front and back ends of the wings are referred to as the leading edge (LE) and trailing edge (TE), respectively. The flat line joining LE and TE is known as the chord, and its length is

denoted by c . The breadth of the foil is known as the span and is marked by s . The space between the upper and lower surfaces of the foil is known as the camber, and the line formed by the midpoint of their distance is known as the mean camber line. Additionally, it is noted that a foil cross-section does not necessarily have to be NACA-shaped, and a flat plate can also be considered a foil (27).

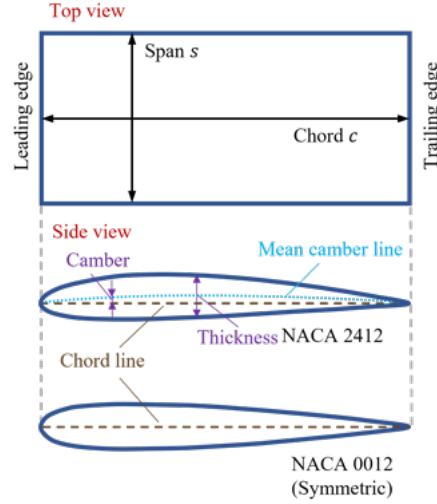


Figure 2.2: The characteristics of the foil.

Only the wave-induced foil propulsion response is addressed in this paper. This type of foil is commonly referred to as “wave foil,” “hydrofoil,” “foil,” “wing,” “plate,” etc. For the sake of convenience, they will be referred to collectively as “foil” in this paper.

2.3.2 Modes of motion

The flapping foils used in WDPSs have three modes of motion: heave, pitch, and flap. Heave motion refers to the vertical translation of the foil, pitch motion refers to rotation around a pivot point P , and flap motion is the result of a combination of heave and pitch motions, as illustrated in Fig.2.3.

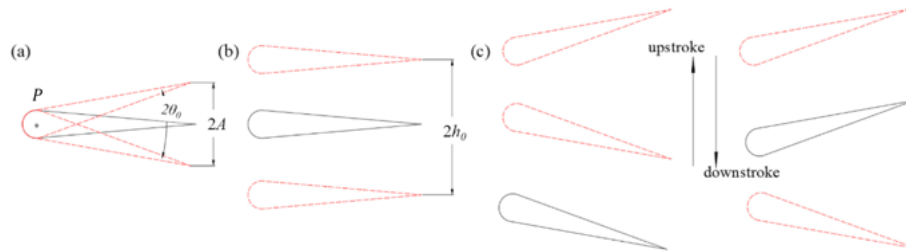


Figure 2.3: The motions of foil: (a) pitching motion, (b) heaving motion and (c) flapping motion of a foil (Wu et al. (23)).

It is possible to constrain the movement of the foils in different ways. There are three types of movement of the foils: passive, semi-passive/semi-active, and active. In passive foils, the flapping motion is entirely flow-induced, and in most cases, these foils are accompanied by springs that provide restoring forces to ensure that the equilibrium position

is restored. In semi-passive/semi-active foils, either heave or pitch is subject to a forced trajectory, while the other motion is induced by the flow. In active foils, both heave and pitch motions are enforced, also known as “flap”. Passive foils, while providing minimal disturbance to the flow, may not be as efficient as semi-passive or active foils. Semi-passive foils, on the other hand, can achieve a balance between efficiency and minimal disturbance to the flow. Active foils, while being the most efficient, may introduce more disturbance to the flow. Therefore, the choice of the type of movement and the control of the relative parameters will depend on the specific objectives.

For the passive foil, there have been several studies exploring the use of elastically deformed flapping foil propulsion. These works primarily focused on the chordwise deformation of the flapping foil, as demonstrated in studies by Anevlavi et al. (28) and Fernandez-Prats (29), among others. Additionally, Zhu (30) investigated the influence of spanwise flexibility, providing insights into the behaviour of the system when spanwise deformation is considered. By examining the impact of both chordwise and spanwise flexibility, these studies contribute to a deeper understanding of the potential benefits and challenges associated with incorporating flexible foils in WDP systems.

However, it is worth noting that in most cases, passive foils are rigid foils accompanied by springs. The inclusion of springs allows for the provision of restoring forces, ensuring that the foils return to their equilibrium positions. Eq.2.1 provides a general representation of the instantaneous heave force and pitch moment applied to the foil. Researchers can also experiment with different types of springs, such as torsion springs or tension springs, to achieve optimal results.

$$F_h(t) = k_h \Delta y \quad (2.1a)$$

$$M_{pto}(t) = k_p \Delta \theta \quad (2.1b)$$

$$M_{pte}(t) = k_e \Delta L L_4 \quad (2.1c)$$

where, $F_h(t)$ is the heave force, $M_{pto}(t)$ is the pitch torsion moment and $M_{pte}(t)$ is the pitch tension moment. k_h , k_p and k_e are the heave spring stiffness, pitch torsion spring stiffness and pitch tension spring stiffness. Fig.2.4 illustrates two different pitch springs, a torsion spring and a tension spring. Δy and $\Delta \theta$ are the vertical and rotational deviation of the foil from its initial position at time t , respectively. ΔL is the extended length of the tension spring, and L_4 is the arm length of the tension force.

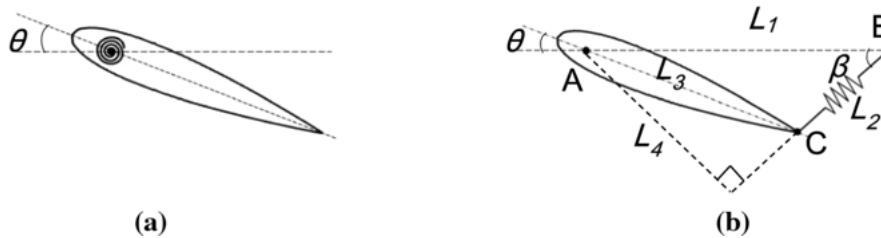


Figure 2.4: (a) torsion spring, (b) tension spring (Feng et al. (31))

The active foils have prescribed trajectories for the heave and pitch motions. This is achieved by setting the harmonic motion for both the heave and pitch, as described in the equations (Eq.2.2a) and (Eq.2.2b). For semi-passive foils, only one of the equations needs

to be satisfied, while the trajectory in the other direction is free.

$$h(t) = h_0 \sin(\omega t) \quad (2.2a)$$

$$\theta(t) = \theta_0 \sin(\omega t + \phi) \quad (2.2b)$$

where $h(t)$ is the instantaneous vertical displacement of the foil. $\theta(t)$ is the instantaneous angle formed between the incoming flow and the chord, also referred to as the angle of rotation. h_0 and θ_0 is the heave and pitch amplitude. ω is the angular frequency (rad/sec), equal to $2\pi f$, where f is the oscillating frequency, and ϕ is the phase difference between the pitch and heave motion. ϕ is an important parameter to consider, as it can impact propulsion efficiency. Platzer et al. (32), and Pedro et al. (33) suggested $\phi = \pi/2$ in uniform flow as the optimal phase difference between pitch and heave. De Silva and Yamaguchi (34) suggested that the optimal phase difference between wave and foil motion (ψ) = $-\pi/2$ (among $-\pi$ and π) while the optimal phase difference between heaving and pitching motion (ϕ) = $\pi/3$ (among 0 and π) in wavy flow.

Various perturbation signals can be added to the basic sinusoidal signal in Eq.2.2a for active heaving motion in order to achieve a broader variety of active heaving motion patterns (35, 36), as illustrated in Eq.2.3.

$$h(t) = h_0 \sin(\omega t) + h_s(t) \quad (2.3a)$$

$$h_s(t) = \begin{cases} \frac{h_{s0} \sin^{-1}[-k_s \sin(\omega t)]}{\sin^{-1}(-k_s)}, & -1 \leq k_s < 0 \\ h_{s0} \sin(\omega t), & k_s = 0 \\ \frac{h_{s0} \tanh[k_s \sin(\omega t)]}{\tanh k_s}, & k_s > 0 \end{cases} \quad (2.3b)$$

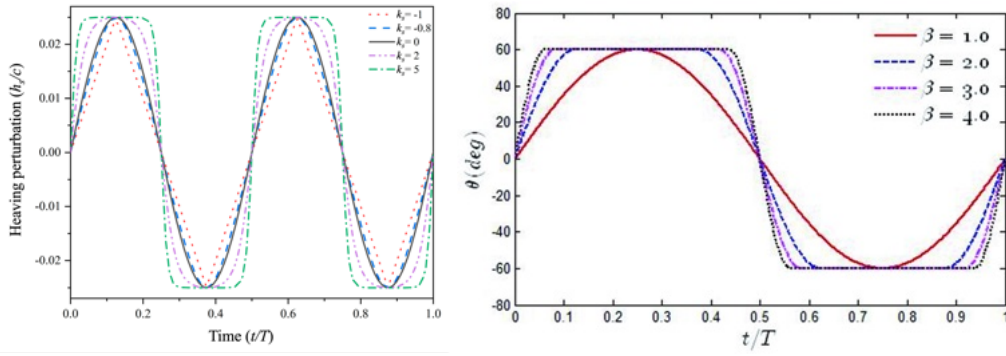
where $h_s(t)$ is the instantaneous heaving perturbation, h_{s0} is the maximum perturbation heaving amplitude, c is the chord length and k_s is an adjustable parameter. By adjusting the parameter k_s , different perturbation heave, including sawtooth, sine, and square, can be generated and added to the base heave trajectory $h(t)$. Increasing k_s leads to a change in the heaving perturbation from a sawtooth to a sine wave and finally to a square wave (Fig.2.5(a)). Apart from the heave signal, Qi et al. (37) considered different pitching trajectory by adjusting β (Fig.2.5(b)). β is defined as the ratio of the pitching frequency f_p to the heaving frequency f_h , i.e. $\beta = f_p/f_h$. When β increases, the pitching profile of the oscillating foil tends from sinusoidal to trapezoidal.

For foils subject to an incoming flow, the resulting angle of attack (AoA) is related to the heave velocity and pitch angle as illustrated in Fig.2.6 and expressed from Eq.2.4 (23, 38, 39).

$$\alpha_{eff}(t) = \theta(t) - \arctan\left(\frac{\dot{h}(t)}{U_\infty}\right) \quad (2.4)$$

where $\alpha_{eff}(t)$ and $\theta(t)$ is the instantaneous effective AoA and pitch angle and $\dot{h}(t) = \frac{dh}{dt}$ is the instantaneous heave velocity.

During such harmonic motions, foil undulations generate wake patterns that result in either drag or thrust. A von Kármán vortex street (vK) develops when the flow velocity in the wake falls below the ambient or incoming velocity, resulting in a drag-dominated state (6). In this scenario, the foil generates a wake pattern characterized by alternating vortices shedding from the trailing edge, leading to increased drag. In a neutral flow state, the wake



(a) Variation of the perturbation in one period at different k_s . (Gao et al. (36))
 (b) Time variation of $\theta(t)$ for different values of β in one period. (Qi et al. (37))

Figure 2.5: Non-sinusoidal heaving and pitching motion signal

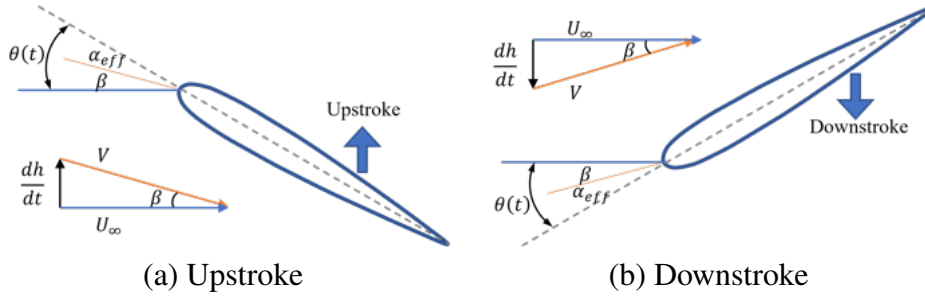


Figure 2.6: The illustration for the effective AoA α_{eff} of flapping foil in uniform flow ($\frac{dh}{dt}$ is the heave velocity, V is the velocity of the stream relative to the foil.).

velocity equals the ambient flow velocity, resulting in a balanced net force on the foil. This means there is neither significant thrust nor drag, and the wake does not contribute to additional resistance or propulsion. This state occurs when the vortex shedding at the trailing edge of the foil is aligned such that it neither enhances nor diminishes the overall thrust or drag. A reversed von Kármán vortex street (RvK) occurs when the wake velocity exceeds the ambient or incoming flow, producing thrust. In this condition, the foil sheds vortices in a reversed pattern compared to the vK state, leading to thrust generation. This drag-to-thrust wake transition stage has been extensively studied for flapping foils in uniform flows, and the foil transition towards thrust occurs with a delay relative to the drag-to-thrust wake transition stage (11, 40, 41, 42).

Fig.2.7 illustrates three fundamental wake patterns, where U_{wake} is the velocity of the wake and U_∞ is the characteristic flow velocity.

2.3.3 Dimensionless parameters

Foils that interact with viscous forces experience thrust, drag and lift forces. Previously, we discussed how the formation of an RvK vortex street is linked to the transition from drag to thrust wakes. Upon exceeding the ambient flow velocity, thrust jets are produced, causing the foil to experience a thrust force — total force acting on the x axis (usually horizontal axis) (43, 44, 11). Lift and thrust forces are generally expressed as dimension-

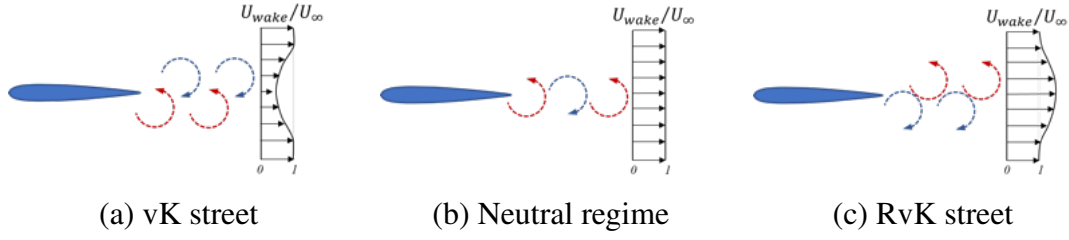


Figure 2.7: The illustration for the drag-to-thrust wake transition of flapping foil

less lift (C_l) and thrust coefficients (C_t). These coefficients are calculated by normalising the force acting on the foil with the dynamic pressure of the incoming flow at the platform area of the foil, as shown in Eq.2.5. Moreover, the power coefficient (C_P) can be present using an analogue expression (45). The similarity law applied here is based on Reynolds number similarity, ensuring that the flow conditions around the model are dynamically similar to those around the actual system, allowing for accurate scaling and comparison of results.

$$C_l = \frac{F_L}{\frac{1}{2}\rho U_\infty^2 cs} \quad (2.5a)$$

$$C_t = \frac{F_T}{\frac{1}{2}\rho U_\infty^2 cs} \quad (2.5b)$$

$$C_P = \frac{P}{\frac{1}{2}\rho U_\infty^3 cs} \quad (2.5c)$$

where ρ is the density of the surrounding flow. F_L and F_T is the lift and thrust force, c is the foil chord and s is the foil span. P is the power and can be calculated by the vertical velocities ($\frac{dh}{dt}$) and angular velocities ($\frac{d\theta}{dt}$), i.e. $P = -F_L \frac{dh}{dt} - M_p \frac{d\theta}{dt}$, where M_p is pitch moment. In addition, Eq.2.5 refers to instantaneous coefficient values and the time-averaged, e.g. over one wave period T , forces and coefficients ($\overline{F_L}$ and $\overline{C}$) are calculated as:

$$\overline{C} = \frac{1}{T} \int_0^T C(t) dt \quad (2.6)$$

$$\overline{F_T} = \frac{1}{T} \int_0^T F_T(t) dt \quad (2.7)$$

$$\overline{F_L} = \frac{1}{T} \int_0^T F_L(t) dt \quad (2.8)$$

where, $\overline{C}$ can be any of $\overline{C_l}$, $\overline{C_t}$, $\overline{C_P}$ and $\overline{C_P}$.

The interaction between a moving foil and the incoming flow is described through the non-dimensional Reynolds and Strouhal numbers. The chord-based Re is calculated from Eq.2.9a, while several expressions for the Strouhal number have been proposed. Two elements in the foil oscillation affect the wake pattern: the peak-to-peak oscillating amplitude, A , of the foil's trailing edge and its oscillating frequency, f . The current mostly used Strouhal number combines both, St_A , Eq.2.9b (46). To be dimensionless, the former element is normalised by the thickness D or chord length c to $A_D = A/D$ or $A_c = A/c$ and the latter becomes reduced frequency k , $k = U_\infty/fc$. Based on the reduced

frequency, a thickness-based Strouhal number $St_D = fD/U_\infty$ and a chord-length-based Strouhal number $St_c = fc/U_\infty$ also have been proposed by Godoy-Diana et al. (10) and Cleaver et al. (47), but St_A is often preferred to better characterise the wake pattern, i.e. RvK street, with a single factor. Recently, a path-length-based Strouhal number Eq.2.9c was proposed by Lagopoulos et al. (11), which should be higher than 1 in order for the foil to produce thrust. In contrast, Read et al. (38) reported optimum thrust generation for $0.2 < St_A < 0.4$.

$$Re = \frac{\rho c U_\infty}{\mu} \quad (2.9a)$$

$$St_A = \frac{fA}{U_\infty} \quad (2.9b)$$

$$St_T = \frac{fT_{foil}}{U_\infty} \quad (2.9c)$$

$$T_{foil} = \int_0^T \sqrt{u(x_{TE}, y_{TE}, t)^2 + v(x_{TE}, y_{TE}, t)^2} dt \quad (2.9d)$$

where ρ is the density of the surrounding fluid, μ is the dynamic viscosity of the fluid. T_{foil} is the trajectory length covered by the TE at the end of one oscillation period. u is the horizontal velocity of the foil and v is the vertical velocity of the foil. The (x_{TE}, y_{TE}) in the Eq.2.9d is the location of the TE. For forced harmonic oscillating motions, $2h_0$ is an estimate for the width of the foil wake. Thus A can be approximated with $2h_0$, and since $f = \frac{\omega}{2\pi}$ Eq.2.9b is also written as Eq.2.10.

$$St_A = \frac{\omega h_0}{\pi U_\infty} \quad (2.10)$$

Zurman-Nasution et al. (48) investigated the limitations of using 2D simulations to study 3D flapping foil motion through the boundary data immersion method (BDIM) and suggested that the valid two-dimensional simulation lies in the St_A range of 0.15 to 0.45.

Non-dimensional parameters are introduced to represent spring stiffness when the foil is restricted by spring force. Yang et al. (49) proposed a non-dimensional parameter frequency ratio (F_{ts}) for torsional stiffness in pitching motion, which is expressed in Eq.2.11a. Similar to the equation defining the coefficient in Eq.2.5, Thaweewat et al. (50) and Zhang et al. (51) and Chen et al. (52) proposed a non-dimensional torsion stiffness (K^*), see Eq.2.11b.

$$F_{ts} = \frac{f_n}{f_{encounter}} \quad (2.11a)$$

$$K^* = \frac{k_p}{\frac{1}{2}\rho U_\infty^2 cs} \quad (2.11b)$$

where f_n is the foil natural frequency and defined as $f_n = \sqrt{k_p/I}/2\pi$. I is the inertia moment of the foil. $f_{encounter}$ is the encounter wave frequency.

2.4 Theoretical study in WDP

2.4.1 Introduction of the WDP concept

The concept of WDP, or Wave Devouring Propulsion, was first proposed in the 1950s by Weinblum and Georg (53), who proposed a theoretical method for studying the seaworthiness of foils. They calculated the hydrodynamic forces of heaving or pitching foil systems without a hull in regular waves of small amplitude by theory, resulting in the simplification not involving severe errors in the cases considered.

Inspired by Weinblum and Georg's work, Wu et al. (54) found that oscillating foils could extract energy from the waves. They suggested that energy can only be extracted when the waves have a vertical velocity component to both the mean free stream and the span of the flapping foil (Fig.2.8). However, this study (54) ignored boundary effects, i.e. they placed the foil more than two chords away from each boundary and did not consider the effects of the free water surface and the solid bottom.

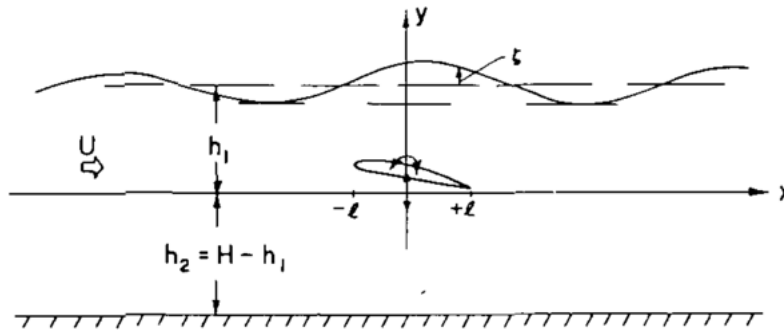


Figure 2.8: Body coordinate system for a two-dimensional foil heaving and pitching in a regular water wave (Wu et al. (54)).

Isshiki (55) extended Wu's theory by theoretically proving the possibility of thrust generation by a passive flapping foil in waves while taking into account the effect of the free surface. In fact, according to recent studies (56, 55, 57, 58), free surface effects should always be taken into account as the free surface strongly influences the vortex wake and the forces upon the foil. The effect of the free surface is only minor (below 2%) when the foil is submerged in the unbounded domain for more than 1.71 chord lengths (59).

Isshiki (55)'s study is also the first time the term 'Wave Devouring Propulsion (WDP)' has been introduced to summarise the fact that wave energy can be transformed by flapping foil to thrust generation. Isshiki then conducted a series of experimental studies on WDP in the following years, which will be continued in Section 2.5. Based on the experiment of the SR108 ship model in (60), Isshiki (61) roughly estimated the propulsive performance of HandyMax B/C under Beaufort 10 with the same boat length in (60) and pointed out that there was no need to extract foil out of the calm sea when running because the benefit gained in bad weather can compensate for the increased resistance of the boat in calm seas due to the foil.

2.4.2 Mathematical models

The mathematical models proposed to better understand the physics of WDP and to use it as an engineering tool in the marine field include those developed by Grue et al. (57), which used a vortex distribution along the centreline of the foil and the wake under infinite water depth, and resulted in good agreement with experimental results of Isshiki et al. (62). Lopes et al. (63) proposed an analytical model to better understand the physics of WDP, and after validation, it was shown to capture oscillating foil lift and thrust forces with enough precision, but does not consider the presence of a free surface. Rozhdestvensky et al. (64) presented a mathematical model of a ship with foil devices on the bow or stern and found that the ship in head seas had reduced heave and pitch motions, especially when the foil was attached to the bow. Their mathematical model included coupled equations of longitudinal ship motions, involving heaving (η_3) and pitching (η_5), are as follows:

$$(M + A_{33})\ddot{\eta}_3 + B_{33}\dot{\eta}_3 + C_{33}\eta_3 + A_{35}\ddot{\eta}_5 + B_{35}\dot{\eta}_5 + C_{35}\eta_5 = F_3 e^{i\omega_e t} \quad (2.12a)$$

$$A_{53}\ddot{\eta}_3 + B_{53}\dot{\eta}_3 + C_{53}\eta_3 + (I_5 + A_{55})\ddot{\eta}_5 + B_{55}\dot{\eta}_5 + C_{55}\eta_5 = F_5 e^{i\omega_e t} \quad (2.12b)$$

$$A_{33} = \int a_{33} d\xi - \frac{U}{\omega_e^2} b_{33}^A \quad (2.12c)$$

$$A_{55} = \int \xi^2 a_{33} d\xi + \frac{U}{\omega_e^2} A_{33}^0 - \frac{U}{\omega_e^2} x_A^2 b_{33}^A + \frac{U}{\omega_e^2} x_A a_{33}^A \quad (2.12d)$$

$$A_{35} = - \int \xi a_{33} d\xi - \frac{U}{\omega_e^2} B_{33}^0 + \frac{U}{\omega_e^2} x_A b_{33}^A - \frac{U^2}{\omega_e^2} a_{33}^A \quad (2.12e)$$

$$A_{53} = - \int \xi a_{33} d\xi + \frac{U}{\omega_e^2} B_{33}^0 + \frac{U}{\omega_e^2} x_A b_{33}^A - \frac{U^2}{\omega_e^2} a_{33}^A \quad (2.12f)$$

$$B_{33} = \int b_{33} d\xi + U a_{33}^A \quad (2.12g)$$

$$B_{55} = \int \xi^2 b_{33} d\xi + \frac{U}{\omega_e^2} B_{33}^0 + U x_A a_{33}^A + \frac{U}{\omega_e^2} x_A b_{33}^A \quad (2.12h)$$

$$B_{35} = - \int \xi b_{33} d\xi + U A_{33}^0 - U x_A a_{33}^A - \frac{U^2}{\omega_e^2} b_{33}^A \quad (2.12i)$$

$$B_{53} = - \int \xi b_{33} d\xi - U A_{33}^0 - U x_A a_{33}^A - \frac{U^2}{\omega_e^2} b_{33}^A \quad (2.12j)$$

$$C_{33} = \rho g \int b d\xi \quad (2.12k)$$

$$C_{55} = \rho g \int \xi^2 b d\xi \quad (2.12l)$$

$$C_{35} = C_{53} = -\rho g \int \xi b d\xi \quad (2.12m)$$

where A_{jk} are the added mass coefficients, B_{jk} are the damping coefficients, C_{jk} are the hydrostatic restoring force coefficients, and ξ is a spatial coordinate along the ship's length. The subscripts j, k can be $1, 2, \dots, 6$. η_j are the displacements in different direction: η_1 - surge, η_2 - sway, η_3 - heave, η_4 - roll, η_5 - pitch, and η_6 - yaw. F_3 and F_5 are the exciting

force and moment, ω_e is the frequency of encounter, M is the mass of the ship, and I is the moment of inertia of the ship.

They also noted that the energy efficiency design index (EEDI) for ships with foils was lower than those without foils, reflecting a reduction in CO₂ emissions due to fuel consumption in representative sea conditions, see in Fig.2.9. Bowker et al. (65) proposed a prediction method that shows the ship's length influences the efficiency of bow foils in various regions of the world, with the optimum conditions for efficient bow foil operation being waves with an average encountered wavelength similar to the ship's length. Rozhdestvensky (66) conducted a parametric analysis on underwater gliders and wave gliders using simplified mathematical models, and the wave gliders model can assess the effect of wave parameters, foil size, and shape on the efficiency of wave-induced thrust generation.

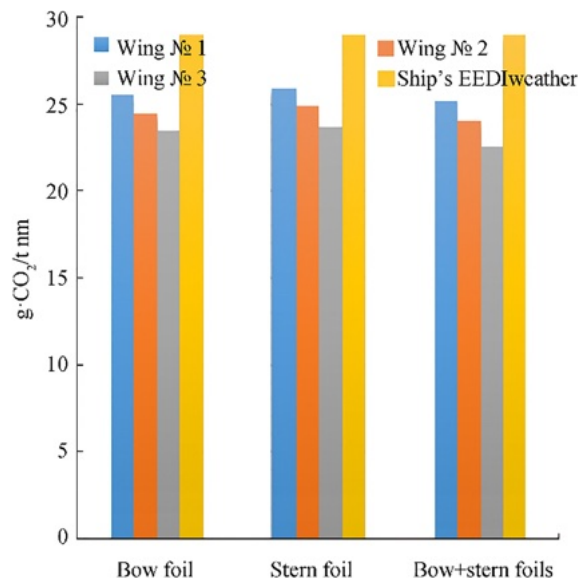


Figure 2.9: Results EEDI_{weather} of a container ship S-175 with wing devices at a representative sea condition (Ship speed=22 kn) (Rozhdestvensky et al. (64)).

2.5 Experimental study in WDP

2.5.1 Experiments on foil itself

The foil is the physical mechanism that allows WDP to take place. Several researchers and groups have focused on the foil itself, as well as innovative deformations of the foil and their hydrodynamic response under different wave conditions.

Isshiki et al. (62, 67) conducted a series of experiments and theoretical discussions about the WDP of a foil placed in waves with a shallow draft. The heaving and pitching motions of the foil were restricted by springs, and the results showed that as the wavelength increases, the passive foil's forward speed and heave amplitude tend to rise in short waves and fall in long waves. Their theoretical calculations also showed a higher forward speed for a deeper draft, which was later confirmed by the experimental work of

Hao et al. (68). These early studies were important for later analysis of numerical WDP (57, 34, 58, 69).

Hao et al. (68) proposed an innovation in wave propulsion devices by adding small floaters to adjust the pitch motion of the foil, named the floater-adjusted wave propulsion device (FAWPD) (Fig.2.10). They compared the FAWPD to a passive foil and found that when the wave-chord ratio is greater than 15, the average thrust of the FAWPD is significantly greater than that of the passive foil for the same wave conditions. This semi-active foil allowed for greater thrust in deeper submergence, in contrast to the behaviour of a sole passive foil.

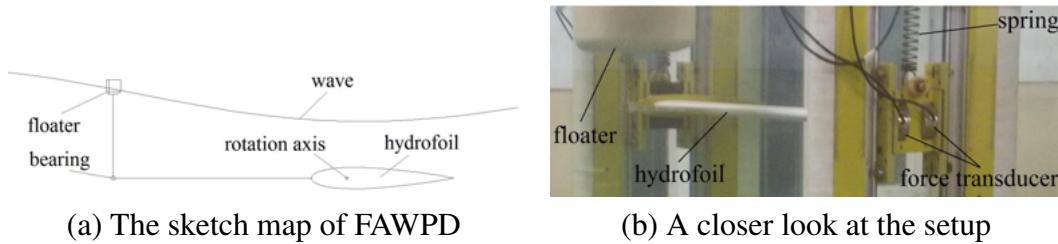


Figure 2.10: FAWPD setup (Hao et al. (68))

2.5.2 Experiments on WDP ship

In the early stages of WDP research, researchers were optimistic and ambitious in their expectations about the capabilities of the WDP. In 1981, Jakobsen (70) experimentally studied the movement of a ship model with foils against waves. Jakobsen's 1 m model was proved to move about 1 m/s under waves (wavelength=2.25 m, wave height=0.1m). Jakobsen (12) also experimented with full-size models but in a controlled environment and showed that if the energy of the waves is sufficient, the WDP produced ship speed could reach or even exceed the speed produced by the ship's conventional propulsion system. Terao (13) tested a floating body with foils on the fore and aft and the model moved toward the waves without any propelling power or control systems. Isshiki (71) suggested that the ship with flapping bow foil can be driven solely by wave power under certain conditions.

The use of foils for ship propulsion has gained attention in recent years, as research has demonstrated the positive effects of foils on ship performance. However, questions about the optimal number and placement of foils on ships have yet to be fully answered. Research has focused on the use of bow foils, which are foils attached to the bow of a ship. Studies have been conducted using both model ships and actual vessels. Isshiki and Murakami (60) conducted a free-running test of a 2 m long SR108 container ship model with rigid foil fixed to the bow under waves. They found that when the wavelength is about 1 m, which is half of the ship model length, the pitch of the ship model becomes prominent, and the ship model runs very fast. Using the same ship model in (60), Isshiki and Naito (72) conducted towing tests and theoretical calculations on the ship resistance and found the calculated results are slightly higher than the experimental ones while having a similar tendency. Turning to the actual sea state, the tests of a 15.7 m long fishing boat with bow foil were conducted by Isshiki et al. (73). The theory overestimated the effects of the foil as in (72), but the theory explained the experimental results qualitatively. Bøckmann

and Steen (74) conducted a model test of a 1:16 scale model ship with fixed a foil bow (Fig.2.11) in regular head waves with 3.7 m wave height and 8.6 s period. The results showed that when the ship's speed was 12 knots, the fixed foil resulted in a 60%, 42% and 45% reduction in ship resistance, heave and pitch motions. Bowker and Townsend (75) evaluated the bow foil effect on a generic cargo ship in waves and showed that the passive spring-loaded bow foil (Fig.2.12) reduces an average ship pitch and heave by 20% and 10% respectively, and the delivered power by up to 50% in regular head waves and estimated by 12% in irregular waves.

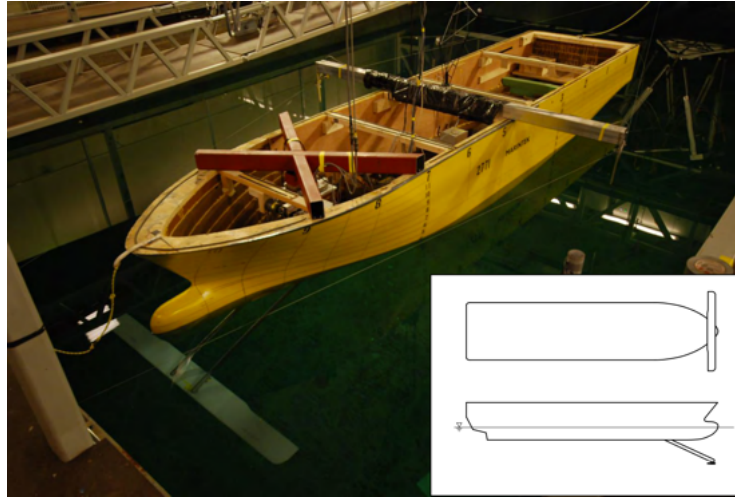


Figure 2.11: 1:16 scale model ship with a fixed foil bow (Bøckmann and Steen (74)).

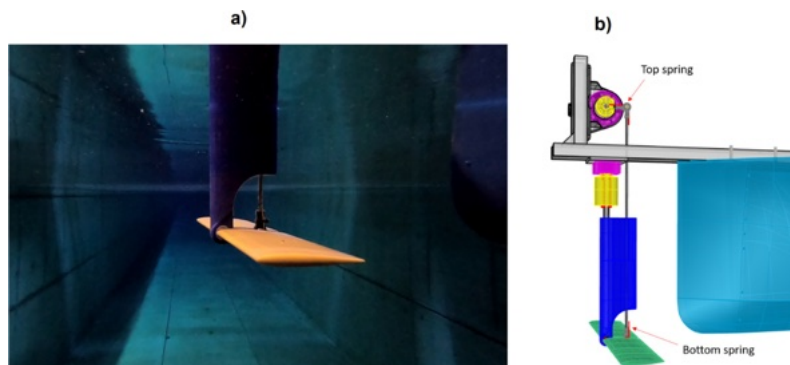


Figure 2.12: Neutrally buoyant, spring loaded foil mounted to the bow of the model (a) Underwater image (b) Computer-aided design (CAD) rendering (Bowker and Townsend (75)).

Instead of one foil, two foils at the bow can show a similar reduction effect even with different ship types and wave conditions. Feng et al. (76) studied the hydrodynamic performance of a container ship with a pair of foils fixed at the bow under head sea conditions. They developed a frequency domain model and conducted a model experiment in the towing tank, which proved theoretically and experimentally that 80% reduced the peak response, 30% and 25% for added resistance, heave and pitch, compared to the original hull. Long-term prediction of 3100TEU container ship at actual seas was also conducted

and showed a slight reduction in EEDI of this ship while the WDPS predicted to be effective in all the sea areas along the sailing route. Bøckmann et al. (25) conducted the model test and time-domain simulation on a tanker ship with two foils, called Wave Augmented Foil Technology (WAFT). The foils are fixed on the bow of the hull, see Fig.2.13. They found that the implementation of wavefoils has lowered the full-scale resistance of the ship by 9-17% in typical sea conditions. In the experiments of Huang et al. (77), a 1:50 scale model of a container ship was used for free running tests with fixed propeller speed (714 Revolutions per minute (RPM)). The effects of two fixed and active pitch oscillating of the bow foils were compared. The results show that in regular waves, the speed improvement is up to 6.24% (0.864 m/s when fixed) for a wavelength-to-boat length ratio of 1.0 and 11.86% (0.882 m/s when fixed) for a ratio of 1.3. Terao (78) carried out model

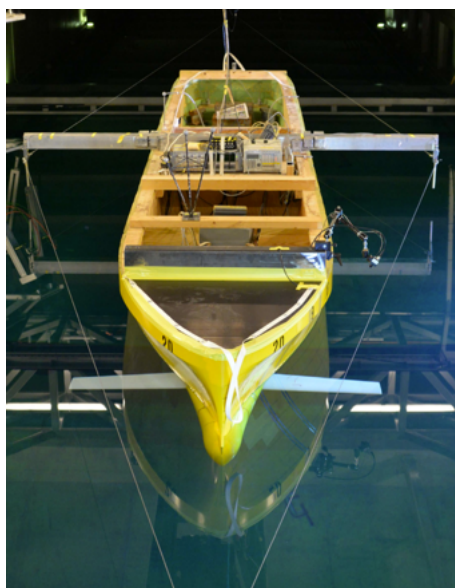


Figure 2.13: The tanker ship model with two fixed foils (Bøckmann et al. (25)).

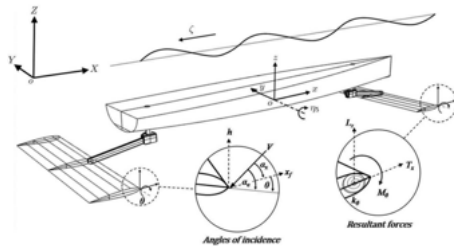
tests (Fig.2.14) for the Suntory Mermaid II—a wave-powered boat—as the experimental support of this CO₂-free vessel and summarised the relevant data from its voyage. For the WDPS design of this vessel, two foils were installed in tandem connection on the bow of this catamaran model and soft springs provided each foil's pitch restoring force to move individually.

The placement of foils on ships has not only been limited to the bow, but researchers have also explored the potential of foils placed on the stern. McGregor and Thomson (79) conducted full-scale sea trials on a 6 m long Caprice yacht with a foil mounted on the stern and found that the flexible foil propeller could convert wave energy into forward motion, acting as an aid to the yacht's movement. The use of foils on both the bow and stern of a ship has also been studied by researchers such as Bowker's team (80, 81). Bowker et al. (80) investigated the performance of an Autonomous Surface Vehicle (ASV) equipped with two passive foils submerged at the bow and stern in head waves in a towing tank and named this WDPS device Flapping Energy utilisation and Recovery (FLEUR) system, see Fig.2.15. They found that the FLEUR device can provide propulsion or power generation for the ASV, with an attainable output of 4 watts and an average of 1 watt. Using the same FLEUR prototype, Bowker et al. (81) presented a discrete time-domain numerical

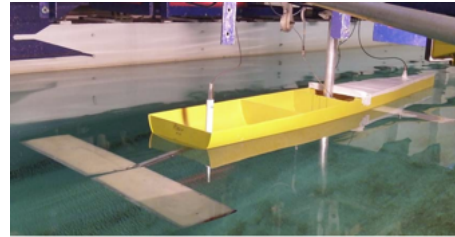


Figure 2.14: Test model of Suntory Mermaid II equipped with all sensors and WDPS (Terao (78)).

model that was validated with the experimental results and showed the dominance of the forward foil over the aft foil in the propulsion effect.



(a) Frames of reference



(b) Carriage setup of FLEUR

Figure 2.15: FLEUR (Bowker et al. (80, 81))

2.5.3 Experiments on other WDP structures

Offshore energy is an emerging technology in the marine field, which might benefit from WDP for resistance reduction and floating structure stabilisation. Terao and Sakagami (82) proposed two new applications of WDPS—floating wind turbine generator systems placed on the catamaran with a foil installed under the bow of the hull and another one based on an active control system. They suggested the WDPS can reduce the wave drift forces and pitching motion in waves. Without using any ship to apply WDPS, Yoshida et al. (27) directly connected the foil with a platform and presented the applicability of WDP for a floating platform. Three flat plate-type foils were horizontally attached to the lower beams of the semi-submersible model, see Fig.2.16. They studied the effects of the foil thickness, foil draft, wave directions, wavelength and attached foil directions and showed the top-view trajectory under different tests. Their research confirmed using a

foil attached to a floating body to support the dynamic positioning (DP) system with less energy consumption.

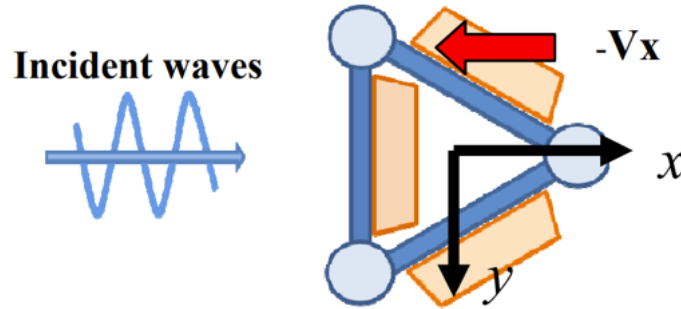


Figure 2.16: Incident wave angle and attached wing direction in the basic performance test of floating structure (Yoshida et al. (27))

2.5.4 Brief discussion of experimental result

The experimental studies on WDP have shown promising results in terms of reducing ship resistance and response magnitude during rough waves. The majority of the experiments were conducted in regular head waves and it has been found that the foils are more effective when attached to the bow than the stern. Researchers have primarily focused on fixed, passive, and semi-passive foils, the selection of which depends on the needs of the WDPS device. The results of these experiments have demonstrated that the WDP characteristics of the foil can not only be used as an auxiliary means of propulsion but also as a stabiliser on vessels and floating structures. Overall, the potential for utilising WDPS in marine transportation and other industries is significant and further research is needed to fully realize its potential.

2.6 Numerical study in WDP

In this section, we discuss the numerical studies on WDP, which can be divided into two groups of numerical solver, potential flow theory and viscous Navier-Stokes solver. Both approaches are widely used in WDP research, and we will present their advantages and limitations.

2.6.1 Potential flow solver

Boundary Element Methods (BEM), also known as Panel Methods or Boundary Integral Methods (BIM), is a widely used numerical simulation tool for studying active foil wave propulsion since 2012. BEM is based on the potential flow theory, which assumes that the fluid is an ideal fluid, and therefore viscosity is not taken into account in the simulation. BEM is used to solve boundary value problems by representing the fluid-structure interaction as a time-dependent problem. The flapping foil is represented by a moving boundary $\partial D_B(t)$ in an earth-fixed reference frame. The free surface and vortex wake are represented by ∂D_F and ∂D_W , respectively. A pressure-type Kutta condition is applied on the

trailing edge of the foil to model the vorticity shed by the foil. The hydrodynamic forces and moments on the foil can be directly obtained by instantaneous pressure integration over the body surface.

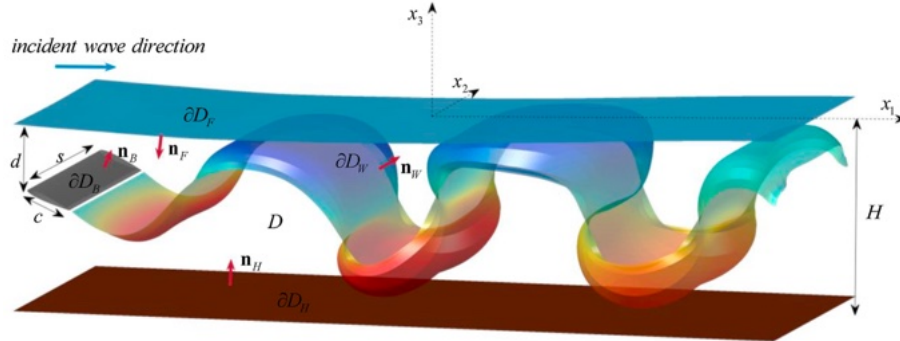


Figure 2.17: Definition of the wave flapping foil problem in the case of a foil (of chord c and span s) in flapping motion under the free surface (at mean submergence d) and in head waves over bathymetry with constant depth H (Filippas and Belibassakis (83)).

Studies using the BEM method have involved both 2D and 3D simulations of flapping foil. Filippas et al. (58, 84) analysed active flapping foils operating under free surfaces and in waves using the unsteady 2D developed-BEM and compared them with experimental results. Due to the development of wave resistance associated with the generation of the complex wave system, at the same pitch maximum angle, the infinite field C_t is maximum. When the submergence/chord is less than 2.5, the closer to the free surface, the lower the C_t . Xu et al. (85) studied the active flapping foil in fifth-order Stokes waves of infinite water depth and the effects of the frequency difference between foil motion and waves. Fig.2.18 shows the time histories of the horizontal and vertical hydrodynamic forces (f_1 and f_3) when the wave frequency (ω_e) differs from the active foil motion frequency (ω). The frequency difference between ω_e and ω indicates the interaction pattern between the wave flow and the foil. The corresponding frequency ratios ω/ω_e are 0.871, 0.974, and 1.15 respectively. It showed that when the frequency of the foil motion is close to the encountered wave frequency, clear envelopes of the force curves can be found, which means the foil's pitch varies violently. The force amplitude varies considerably for each period in Fig.2.18, indicating that the local flow velocity and the effective AoA of the foil differ substantially during the different oscillations. Also, the free surface effect becomes less significant with increasing heave amplitude and pitch angle (85). One notable work in the field of 3D BEM is by Belibassakis and Politis (86), which focuses on the hydrodynamic performance of flapping wings for enhancing ship propulsion in wave conditions through active control. They investigated two configurations: horizontal and vertical flapping wings positioned beneath the ship's hull. The findings revealed that both vertical and horizontal flapping wings contribute to thrust production, ranging from 10% to 50% of the ship's calm-water resistance at the same speed. Notably, their present system enhanced the ship's propulsive performance while providing significant dynamic stabilisation, particularly in the case of the horizontal flapping wing. Building upon this work, Belibassakis and Filippas (87) further explore the interconnected dynamics between hull motion and a horizontal flapping foil positioned below the keel in the presence of an irregular head wave associated with specific sea conditions.

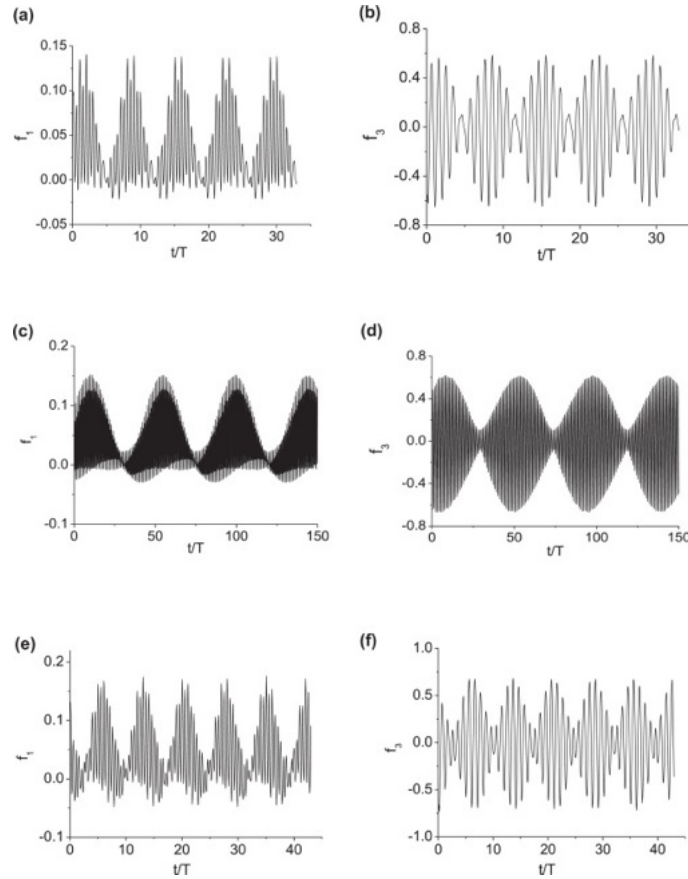


Figure 2.18: The time histories of the horizontal (f_1) and vertical (f_3) forces on the foil for different frequency ratios. (a),(b) $\omega/\omega_e = 0.871$; (c),(d) $\omega/\omega_e = 0.974$; (e),(f) $\omega/\omega_e = 1.15$ (Xu et al. (85)).

They found that when the wave height increased significantly, more thrust could be produced (see Fig.2.19(a)). Moreover, Fig.2.19(b) indicated a decrease in thrust/resistance ratios at lower submergence. According to Isshiki et al. (67), the decrease of the draft results in increased thrust, which is acceptable as lower submergence means more wave energy that can be transformed into thrust. However, lower submergence also means more wave resistance due to the effect of the free surface. In addition, the vertical oscillating motion of the foil caused by the vessel's pitching is reduced due to the smaller radius of rotation. Thus, it is suggested in this case that the growth of thrust does not exceed the growth of resistance because of the significant wave resistance from the attached ship. Furthermore, it brings to our attention that when the size of the boat is too large to a certain extent, the submergence depth of the foil should not be too shallow. In practice, the optimum submergence depth of the foil should be chosen in relation to the size of the boat for higher propulsion efficiency. Recently, there have been advancements in utilising graphics processing units (GPUs) for research purposes. Filippas and Belibassakis (83) introduced the use of GPUs for time-domain BEM analysis, specifically focusing on the hydrodynamic performance of 3D submerged flapping foils, also known as biomimetic thrusters. Their work demonstrated that employing GPUs in mixed precision instances resulted in superior computational speed without significant compromise in accuracy.

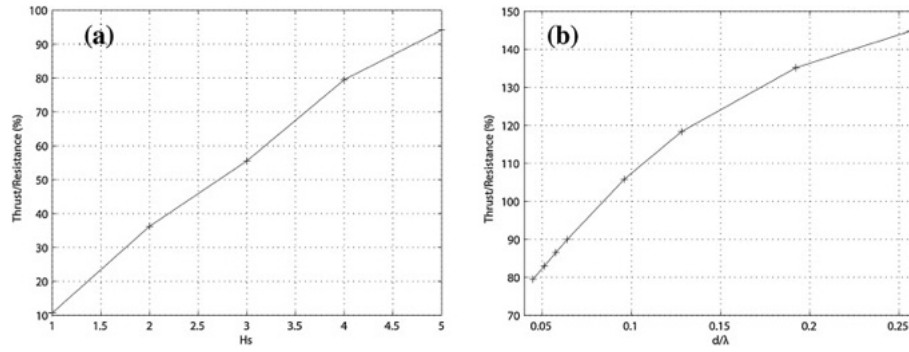


Figure 2.19: Mean thrust to calm water resistance ratio as (a) function of sea conditions represented by significant wave height, and (b) as function of the mean foil submergence to modal wavelength ratio (Belibassakis and Filippas (87)).

2.6.2 Navier-Stokes flow solver

Computational fluid dynamics (CFD) solves Navier-Stokes equations with suitable turbulence models. It extends the applicability of potential flow theory by taking into account the effects of viscosity. In Tab. 2.1, a chronological overview of previous CFD studies on WDP is presented, revealing distinct research objectives that can be broadly categorized into four main application areas: wave-propelled physical mechanisms, ASV, ship propulsion, and other innovative marine systems.

Table 2.1: The CFD studies of WDP.

Investigators		Foil related information				Numerical methodology						Mesh technology	
Reference	Foil shape	Flexibility of foil	Motion	Environment	Application	2D/3D	CFD type	Method	Solving equation	Turbulence model	Free surface deformation	Grid	
(88)	NACA0012	Rigid	Active	Regular waves and irregular waves	Ship propulsion	3D	In-house BEM and CFD	FVM	RANS	SST k- ω	VOF	Unstructured grid	-
(89)	NACA0008	Rigid	Semi-active	Regular waves	Physical mechanisms	2D	FLUENT	FEM	RANS	RNG k- ϵ	No free surface	Unstructured grid	-
(90)	NACA0008	Rigid	Active and semi-active	Regular waves	Wave-powered vehicle	2D	FLUENT	-	RANS	RNG k- ϵ	No free surface	Unstructured grid	Dynamic mesh
(52)	NACA0012	Rigid	Semi-active	No wave	Wave glider	2D	FLUENT and STAR-CCM+	-	RANS	SST k- ω	No free surface	Structured grid	Overset mesh
(31)	NACA0008	Rigid	Semi-active	Uniform flow	Wave glider	2D	FLUENT	-	RANS	RNG k- ϵ	No free surface	Unstructured grid	Dynamic mesh
(91)	NACA0009 and NACA0006 as a winglet	Rigid	Active	Uniform flow	Wave glider	3D	STAR-CCM+	-	RANS	k- ω	No free surface	Structured grid	Overset mesh
(92)	NACA0012	Rigid	Static to the ship	Clam water and regular waves	Ship propulsion	3D	In-house CFD	FVM	RANS	-	VOF	Structured grid	-
(93)	NACA0012	Rigid	Semi-active	Regular waves	Wave glider	2D	AQWA (without foil) and FLUENT	-	RANS	SST k- ω	No free surface	Unstructured grid	Dynamic mesh
(94)	NACA0012	Rigid	Passive	Regular waves	Wave glider	2D	FLUENT	-	RANS	k- ϵ	No free surface	Unstructured grid	Dynamic mesh
(95)	NACA0015	Rigid	Semi-active	Regular waves	Wave glider	2D	FLUENT	FVM	RANS	SST k- ω	No free surface	Structured grid	Overset mesh
(96)	NACA0012	Rigid	Semi-active	Regular waves	Wave glider	2D	Custom C++	FDM	LES	-	No free surface	Cartesian grid	-
(97)	NACA0015	Rigid	Passive	Regular waves	Physical mechanisms	2D	In-house CFD	FVM	RANS	-	LSM	Cartesian grid	-
(51)	NACA0012	Rigid	Semi-active	No wave	Wave glider	2D	FLUENT	FVM	RANS	SST k- ω	No free surface	Structured grid	Overset mesh
(98)	-	Rigid	Passive	Regular waves	Retractable foil for ship propulsion	3D	Simulink and STAR-CCM+	-	-	-	VOF	Unstructured grid	-
(26)	NACA0015	Rigid	Passive	Irregular waves	Retractable foil for ship propulsion	3D	Simulink and STAR-CCM+	-	RANS	RNG k- ϵ	VOF	Structured grid	Overset mesh
(36)	NACA0012	Rigid	Active	No wave	Physical mechanisms	2D	FLUENT	FVM	RANS	S-A	No free surface	Unstructured grid for inner domain; structured grid for outer domain	-
(99)	NACA0012	Rigid	Passive	Regular waves	Wave-driven catamaran	3D	STAR-CCM+	-	RANS	RNG k- ϵ	VOF	Structured grid	Overset mesh
(45)	NACA0015	Rigid	Active	Regular waves	Ship propulsion	2D	FLUENT	-	RANS	SST k- ω	-	Structured grid	Overset mesh
(100)	NACA0012	Rigid	Semi-active	No wave	Wave glider	2D	FLUENT	FVM	RANS	S-A	No free surface	Unstructured grid for fluid; structured grid for solid	-
(101)	NACA0008	Rigid	Semi-active	No wave	Wave glider	2D	FLUENT	FEM	RANS	-	No free surface	Unstructured grids	Dynamic mesh
(39)	NACA0012	Rigid	Active	Regular waves	AUV	3D	Star CCM+	-	RANS	k- ϵ	No free surface	Structured grid	Overset mesh
(102)	NACA0015 with elastic plate	Rigid	Active	Still water and short wave	stationkeeping system of a floater	2D	FLUENT and MECHANICAL	FVM	RANS	SST k- ω	VOF	Unstructured grid for fluid; structured grid for solid	-

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Investigators Reference	Foil related information				Numerical methodology					Free surface deformation	Grid	Mesh technology
	Foil shape	Flexibility of foil	Motion	Environment	Application	2D/3D	CFD type	Method	Solving equation	Turbulence model		
(37)	NACA0012	Rigid	Active	No wave	Wave glider and physical mechanisms	2D	FLUENT	FVM	RANS	S-A	Unstructured grid	Dynamic mesh
(103)	-	Rigid	Active	Regular waves	Wave-powered unmanned catamaran ship	3D	AQWA (without foil) and FLUENT	FVM	-	SST k- ω	Unstructured grid	Dynamic mesh
(49)	NACA0012	Rigid	Semi-active	Uniform flow	Wave glider	2D,3D	STAR-CCM+	-	RANS	-	Structured grid	Overset mesh
(69)	NACA0012	Flexible	Active	Regular waves	Unmanned underwater vehicles propulsion	2D	FLUENT	-	RANS	RNG k- ϵ	Unstructured grid	Dynamic mesh
(104)	NACA0012	Rigid	Semi-active	Regular waves	Wave glider	3D	FINE/Marine and STAR-CCM+	FVM	RANS	SST k- ω	Unstructured grid	Overset mesh, cut-cell mesh, sliding mesh
(105)	NACA0018	Rigid	Active	Regular waves	Ship propulsion	2D	AQWA (without foil) and FLUENT	-	RANS	SST k- ω	Unstructured grid	Dynamic mesh
(106)	NACA0012	Rigid	Active	Regular waves	Wave glider	2D	FLUENT	Cell-centered FVM	RANS	RNG k- ϵ	Unstructured grid	Dynamic mesh
(107)	NACA0015	Rigid	Active	Uniform flow	WDPS concept	2D	-	-	RANS	RNG k- ϵ	-	-
(108)	Flat plate	Rigid	Active	Uniform flow	Underwater vehicles	2D	-	FDM	-	-	Boundary-fitted grid	-
(109)	NACA0012	Rigid	Active	No wave	Ship propulsion	2D	FLUENT	-	RANS	k- ϵ	Unstructured grid	Dynamic mesh
(34)	NACA0015	Rigid	Active	Regular waves	Physical mechanisms	2D	FLUENT	Cell-centered FVM	RANS	SST k- ω	Unstructured grid	Dynamic mesh

A comprehensive understanding of the underlying physical mechanisms is essential for its widespread application in marine vehicles and platforms. These studies typically focus on the movement of foils and employ CFD simulations to explore a wide range of conditions that cannot be easily achieved through experimental means. For instance, Gao et al. (36) studied different heave perturbation based on sinusoidal heave motion by adjustable parameter k_s such as sawtooth ($k_s=-1$), sine ($k_s=0$), and square ($k_s=5$), mentioned in Fig.2.5(a). Gao et al. (36) investigated fluid dynamics around the heaving foil with finite volume method (FVM) and found that adding any perturbation increased the axial thrust. As the perturbation frequency increases, all wake structures first go through an asymmetric wake phase (at perturbation frequency = 2 Hz) and finally evolve into a chaotic wake. While Gao et al. (36) focused on the heave motions, Qi et al. (37) studied active oscillating foils with sinusoidal and non-sinusoidal pitch (Fig.2.5(b)) using the dynamic and moving mesh. For the sinusoidal pitching ($\beta=1$), as the reduced frequency increased, both the mean output power coefficient and the optimal pitching amplitude increased. For the non-sinusoidal pitching, when the pitching amplitude was small, as β increases, the propulsive performance of the flapping foil gradually improves, with the trapezoidal-like pitching profile ($\beta = 4$) being more effective. De Silva and Yamaguchi (34) investigated the active 2D oscillation foil in wavy flow for marine propulsion by using the custom Fluent solver with the dynamic mesh technology and the Volume of fluid (VOF) method. They suggested when the ratio of wave amplitude to foil chord length is between 1/14 and 3/14, over 50% of the wave energy can be converted into propulsion energy.

In recent years, the utilisation of foils in ASVs has garnered significant attention due to advancements in technology. Researchers are increasingly recognizing the immense potential of foils, particularly in the form of wave gliders, for various ASV applications. A notable characteristic of wave glider simulations is the omission of the free surface. This omission is justified by the fact that the foil is typically submerged under the surface by more than four times its chord length (93). Consequently, the influence of the free surface can be considered negligible, allowing the domain to be treated as an infinite fluid without incorporating the ship and the free surface in the simulation. Yang et al. (104) predict the dynamic performance of the wave glider with six pairs of 3D tandem foils in head seas. The interaction of flow and the boat-foil system was simulated by FINE/Marine with VOF model and the passive eccentric rotation of foils was simulated by STAR-CCM+ with overset mesh. The passive rotation was restricted by the spring force that restored them to the original position. Fig.2.20 was the vortex structure of the six tandem foils in one cycle. Foils rotated counterclockwise as the glider descends (a-e) and clockwise as the glider ascends (f-g). In addition, when the glider was in a trough, the clockwise rotation was greater than the counterclockwise rotation. They suggested that wave surge forces from the surface vessel and passive eccentric rotation of the foils were factors that affected the propulsive efficiency of the wave glider. Following the study of Yang et al. (104), Yang et al. (49) further focused on the six tandem foils with passive pitch and conducted parametric studies. The results reported that for a 2D single foil the best pivot axis on it was the leading edge and for the six foils, the optimum torsional spring stiffness k_p was 0.8 N·m/rad (frequency ratio=25) in 2D simulation and 11.8 N·m/rad (frequency ratio=17) in 3D. Qi et al. (100) studied the mass ratio effect—density ratio of the foil to water—of a 2D flapping foil on a wave glider with FVM. The foil was semi-active with a

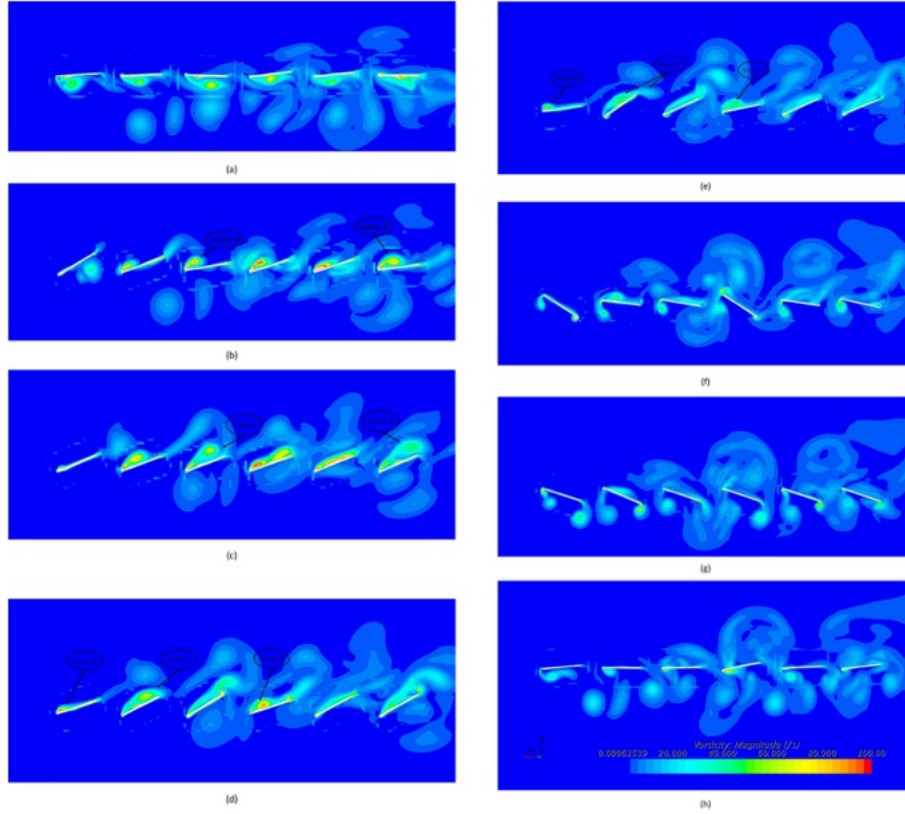


Figure 2.20: Time series of the vorticity magnitude development in one cycle of the foil (a) $t = 0$, (b) $t = 0.175$, (c) $t = 0.3$, (d) $t = 0.4$, (e) $t = 0.5$, (f) $t = 0.66$, (g) $t = 0.8$, (h) $t = 1.0$. (Yang et al. (104))

full-prescribed harmonic heaving motion and a passive pitching motion. With the increase in the mass ratio, the mean output power of the flapping foil decreases monotonously at high reduced frequency k and increases at low k , whereas as the mass ratio increases in the case of critical k , the average output power coefficient increases and decreases. Feng et al. (31) compared the characteristics of linear (torsion) and nonlinear (tension) springs mounted on the foils of wave glider. Fig.2.21 illustrated that, for the same wave condition, the forward speed and propulsion force of a propulsion mechanism with a nonlinear spring is greater than that of a linear spring. Sang et al. (93)'s numerical model simulated the float response of another USV type (named single-body architecture wave glider (SAWG), see Fig.2.22) by AQWA and the derived motion of semi-active flapping foils attached to the USV by Fluent without wave surface. They presented the average forward velocity under different spring constants and wave frequencies (Fig.2.23). After the verification of the model by tank experiment and sea trial, they suggested there is an optimum spring constant at different wave frequencies for maximum forward velocity, for example, an optimum spring constant of 6 N·m/rad at a wave frequency of 0.4 Hz Qingdao offshore.

Wang et al. (96) proposed a mechanism that allowed the foil in the wave glider to harvest energy while propulsion and simulated the energy harvesting performances by a custom C++ procedure using immersed boundary and large eddy simulation methods. The harvested energy was used to provide the energy required to actuate the pitch of the semi-

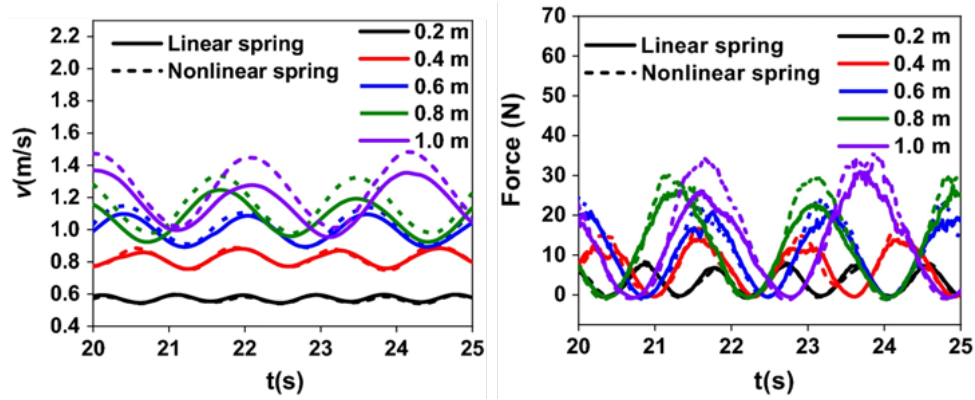


Figure 2.21: Forward velocity and propulsion force (right) under different sea conditions. (Feng et al. (31))

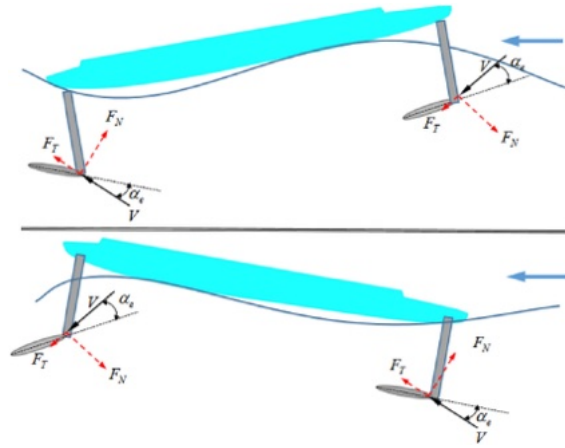


Figure 2.22: The wave propulsion principle of the SAWG. (Sang et al. (93))

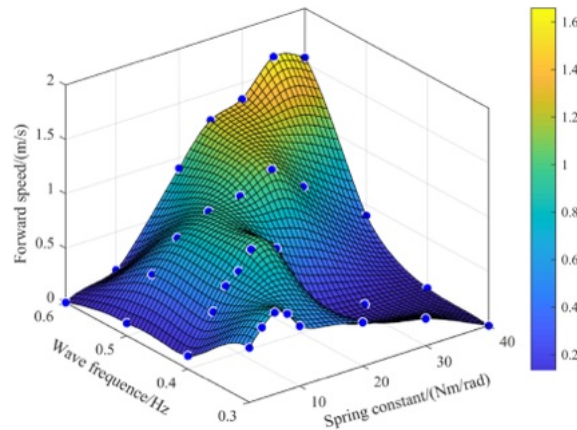


Figure 2.23: The mean forward speed responses for varied spring constants and wave frequencies. (Sang et al. (93))

active foil. When the wave glider was travelling upstream, the oscillating foil can only function as a propeller; when the wave glider was travelling downstream, the oscillating foil can act as a propeller and an energy generator. Sang et al. (94) presented a self-

adjusting lower limit (SALL) mechanism for wave glider optimisation. This mechanism can passively control the maximum swing angle of foil in a counterclockwise direction according to the umbilical cable inclination angle. The CFD simulation results showed that the thrust of the SALL underwater glider increased by 17.78%, 7.42% and 20.70% compared to a conventional underwater glider when the stiffness coefficient of the torsion spring was $k_p=4 \text{ N}\cdot\text{m/rad}$, $k_p=6 \text{ N}\cdot\text{m/rad}$ and $k_p=8 \text{ N}\cdot\text{m/rad}$ respectively. While wave gliders continue to be the primary focus of research, it is worth noting that foils are also being explored for their application in other types of ASVs. Ci et al. (39) discussed about the structural optimization of the foil applied on the autonomous underwater vehicles (AUV), such as REMUS 100 AUV, using 3D CFD approaches—Star CCM+ and suggested that trapezoid foil with the NACA0012 section, i.e. the ratio of LE to TE of the foil=1:2, had better performance. Liu et al. (69) studied the performance of a 2D flexible flapping foil in regular waves for unmanned underwater vehicles propulsion. The method was validated by rigid foil experiments and further analysis also compared with rigid foil under the same flow conditions. In the case of larger wave amplitudes or smaller wavelengths, both fixed and flexible foils are able to generate higher thrusts, see Fig.2.24. The flexible deformation of the foil not only helps to increase the vortex strength, but also reduces the dissipation of vortex energy in the flow field. Higher performance is achieved when the foil moves at a frequency equal to that of the wave it encounters. The phenomenon has likewise been observed by Xu et al. (85) in their study of active rigid foil in nonlinear waves.

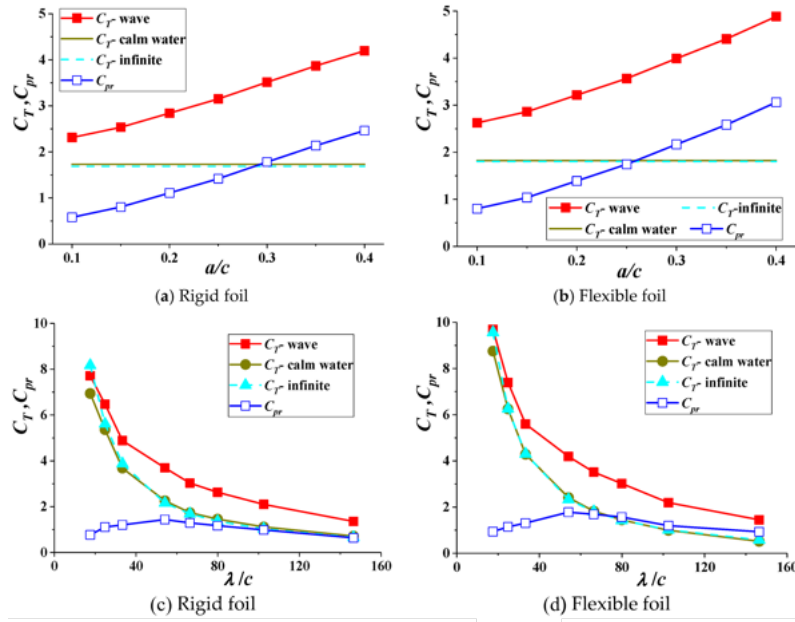


Figure 2.24: (a) (b) Average thrust and wave power utilisation coefficients for the foil versus non-dimensional wave amplitudes. (c) (d) The average thrust and wave power utilisation coefficients for the foil versus non-dimensional wavelengths. (Liu et al. (69))

Furthermore, researchers have also directed their attention towards investigating the utilisation of foils in ship propulsion, expanding the scope of foil applications beyond ASVs. Zhang et al. (26) proposed a novel type of the retractable wave foil. 4 pairs tandem foils placed on both sides of the ship (Fig.2.25). The foils can unfold in rough

seas and passively change the angle of attack. The CFD results by STAR-CCM + were validated by mathematical simulation in Simulink. Fig.2.26 showed the ship propulsion and anti-pitching moments generated by the foil for different wave encounter angles. It was noticed that when the wave encounter angle is 180° , the thrust and moment of the ship were the largest. Therefore, the wave excitation to the ship is the maximum in the head wave, which was also confirmed by Li et al. (110) with the study of Natural-energy-driven surface vehicle (NSV). Zhang et al. (98) introduced a novel way for the foil to

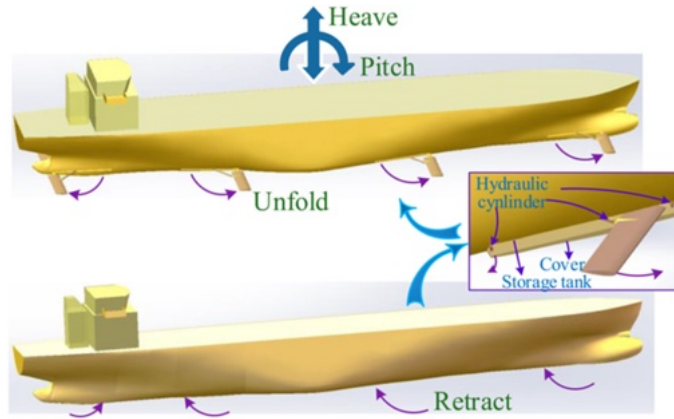


Figure 2.25: The ship with retractable flapping foils. (Zhang et al. (26))

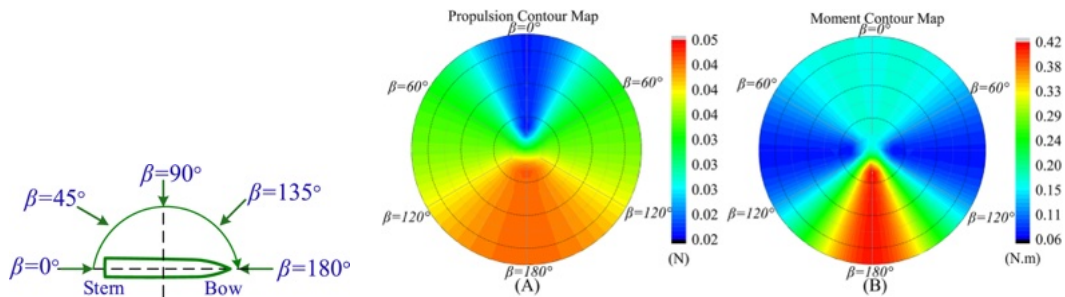
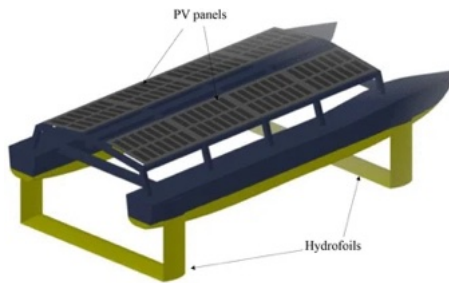


Figure 2.26: The contour map for the propulsion (A) and moment (B) with different encounter angle. (Zhang et al. (26))

be equipped to the hydraulic device, the pitch of the wave foil was passively adjusted by spring damping and the wave foil can be retracted and expanded by a strut connected to the hull. The time-domain simulations and wave-ship-foil coupling simulations were conducted by Simulink and STAR-CCM+. The CFD results showed that passive pitching adjustment facilitates better extraction of wave energy and anti-pitching performance. By retracting the foils when not in use, ships can reduce drag and increase fuel efficiency. However, in the case of conventional foil designs that are attached and non-retractable, different considerations come into play. Moreira et al. (45) presented a 2D parametric study of dual flapping foils system. They focused on evaluating the ship propulsion and energy harvesting performance of foils and placed the dual foils in various unaligned or out-of-phase positions. Under the same sea state, higher obtained energy and hydrodynamic coefficients (C_t , C_l and C_p) were obtained for foils with larger horizontal spacing (cases 1–2) compared to those with smaller horizontal spacing (cases 3–4). Similar to

Mermaid II, the catamaran type ship brings to attention for wave propulsion. Wang et al. (103) analysed 3D low-speed wave-powered unmanned catamaran ship (Fig.2.27) motion without/with horizontally fixed foils under head wave through both potential theory (AQWA) and the CFD method (Fluent). The results showed horizontal forces acting on the tandem foils provided thrust to propel the ship while vertical force on the catamaran with foils was smaller than that of without foils therefore decreased the heave and pitch motions of the ship and the wave-added resistance on the ship. Zhang et al. (99) simulated the same type of catamaran by the combination of multi-body dynamics model and STAR-CCM+. Just as the bowfoil worked better than the sternfoil, the foil thrust was found to be much greater at the bow of the catamaran than at the stern, as the front foil heaved more than the aft foil. They also concluded that the motion of the catamaran completely driven by wave and its advancing speed was positively correlated with wave height and negatively correlated with wavelength. In the context of numerical simulation of bow



(a) The illustration



(b) The photograph

Figure 2.27: The unmanned catamaran ship completed driven by foils (Wang et al. (103))

foil, an innovative project on ship dual-fuel engines and propulsion is proposed, called the SeaTech Horizon 2020 project, H2020 project for short (111). Fig.2.28 illustrated the model of bow foil in the H2020 project. This project provides a renewable-energy-based hybrid solution for the reduction of fuel consumption and operational costs. Belibassakis et al. presented a preliminary analysis of the proposed bow foil and ship engines in waves as part of this project (112, 113, 114) using lab test and BEM and CFD simulation. In the study, the additional thrust generated by the foil allowed the engine to run at partial load without affecting the speed of the ship and resulting less CO₂ emissions. Ntouras et al. (92) also confirmed the feasibility of a static foil bow with a non-zero AoA as a trim-pitch stabiliser in head waves by experimental data in towing tank and in-house CFD simulation result. Both numerical and experimental results showed the significant decrease in added resistance in the waves of the 12–34% compared to the bare hull. Furthermore, the economics of bow foil thrusters were studied by Ventikos et al. (115). Their proposed Life Cycle Cost Analysis (LCCA) framework includes construction, operation, maintenance and end-of-life. The operating cost model result proved that the foil thruster proposed by H2020 Project is economically promising through the expected engine power demands.

Moreover, the research endeavours extend to exploring the implementation of foils in other innovative marine systems, showcasing the diverse range of applications for foils beyond ASV and ship propulsion. Kumar and Shin (116, 117) conducted a series of experiments to explore the practical application of foils in designing stationkeeping systems for deep-water floaters. They (116, 117) investigated and measured the thrust force gen-

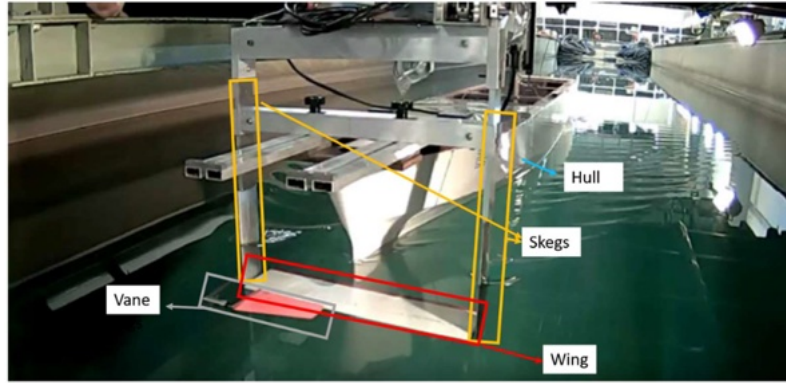


Figure 2.28: SeaTech Horizon 2020 project: Experimental configuration of the dynamic wing arranged at the bow of the ferry ship model tested. (Ntouras et al. (92))

erated by a rigid foil attached to an elastic plate, which underwent flapping motion in a wave flume. Their empirical studies led to the development of an empirical formula that incorporates azimuth angles, allowing for the prediction of thrust force for a flapping foil. This formula was derived through multiple regression analysis of their experimental data. Additionally, Kumar and Shin (102) extended their predictions by employing CFD method. utilising a Two-way Fluid-Structure Interaction (FSI) method, they examined the interaction between a submerged actively flapping foil and the surrounding fluid in short waves. The geometry in (102) was the same as in experiments, see Fig.2.29. The thrust forces were evaluated at five different flapping frequencies and the results showed that the positive force from the active foil was dominating and then can benefit the design of a stationkeeping system of a floater in short waves while passive foil proved ineffective in the experiment (116).

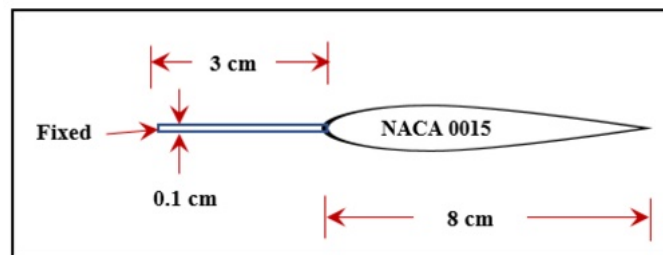


Figure 2.29: The flapping foil with an elastic plate attached to the leading edge. (Kumar and Shin (116, 102))

2.6.3 Brief discussion of numerical result

In the reported numerical study of wave-induced propulsion, the influence of the flapping foil was detailed and various parameters were studied. Both the potential flow solver and the Navier-Stokes flow solver were used. The submergence depth was found to be a significant factor, with shallower depths generally converting more wave energy and generating more thrust, but wave resistance becoming a significant issue in some cases. The presence of the free surface was also found to be important, but its effect became less

Table 2.2: The applications of WDPS on manned vessels.

Year	Creator	Location	Ship	Ref.
1895	Herman Linden	Naples, Italy	‘Autonaut’	(71)
1935	Unkown	Long Beach, California, USA	18-inch model	(119)
1950	John S. McCubbin	Victoria, Australia	-	(120)
1966	Joseph A. Gause	Burlington, Ontario, Canada	Gausefin I	(121)
1981	Einar Jacobsen	-	Kystfangst	(122, 123)
2008	Yukata Terao	Tokai University, Japan	Suntory Mermaid II	(78)
2012	MARIN	Netherlands	O-foil	(124)
2013	Rolls-Royce, MOST and SeaSpeed	UK	Wave Augmented Foil Technology (WAFT)	(25)
2014	Dolprop	Ekerö, Sweden	Fluke	(125)
2019	WAVEFOIL	Norway	SmartWings	(17)

significant with increasing heave amplitude and pitch angle. Finally, the results showed that head waves performed better for wave excitation propulsion and generated more anti-pitching moments to boost ship stability.

2.7 WDP Applications (WDPS)

This chapter aims to provide an overview of the various industrial applications of wave propulsion technology, specifically those that have reached full scale size and have been proven in practice. The focus will be on the use of WDP in unmanned surface vehicles, the shipping industry, and other industries such as the offshore industry. The knowledge gained from the theoretical, experimental, and numerical studies discussed in previous chapters will be applied to these real-world scenarios to demonstrate the potential of wave propulsion technology.

2.7.1 Foil with manned vessels

A full list of applications for foil-attached manned vessels is provided in Tab. 2.2. The idea of using wave-powered foils for marine propulsion dates back to 1895, when Linden proposed a British patent for a ship equipped with a flexible foil at the bow (71). One of the most notable examples of this technology in practice is the Suntory Mermaid II (78), a wave-powered boat that successfully completed a 110-day voyage from Honolulu, Hawaii to the Kii Channel, Japan under the leadership of Captain Kenichi Horie. This journey demonstrated the potential of foil propulsion for eco-sailing and gained international attention. More recently, the WAVEFOIL company has developed technology for retractable bow foils attached to ships (118) and has expanded its applications from 2019 onwards. These foils are used on a wide range of vessels, from large 45 meter passenger ships to small 23 meter ambulance vessels.

2.7.2 Foil with Unmanned surface vehicles (USV)

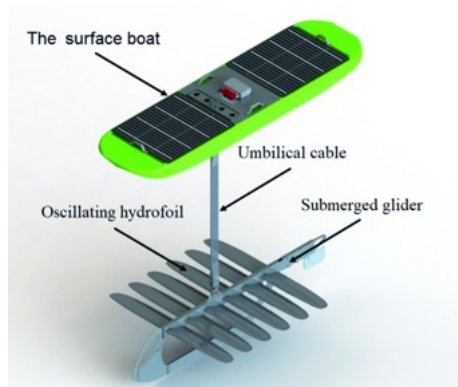
Foil-attached unmanned vessels have been gaining increasing attention as a promising solution for oceanic applications, e.g. monitoring. These vessels utilise the energy from

Table 2.3: The applications of WDPS on unmanned vessels.

Year	Creator	Location	USV	Ref.
2007	Liquid Robotics	USA	Wave Glider	(14)
2013	AutoNaut	UK	AutoNaut series	(16)
2014	Team of Ocean Mobile Observation Technology (TOMOT)	China	Black Pearl	(127)
2021	Li Ye et al.	China	NSV	(110)
2022	Hongqiang Sang et al.	China	SAWG	(93)

ocean waves to provide propulsion, enabling them to perform long-duration missions with limited energy consumption. A comprehensive list of these applications can be found in Tab. 2.3.

There are two primary mechanisms for underwater foil wave propulsion: heave-induced gliding and pitch-induced flapping (126). The heave-induced mechanism uses the motion of the vessel to generate a flow over submerged foils, similar to an aerial glider with umbilical cables. The pitch-induced mechanism utilises the movement of the vessel to drive the foil in a flapping motion, interacting with the incoming wave's oscillating flow to produce thrust. Commercial USVs, such as the Liquid Robotics' Wave Glider and AutoNaut's AutoNaut, have utilised both of these propulsion methods (14, 15). A wave glider typically consists of a floating body, an underwater propulsion mechanism, and a connecting cable between the two, as shown in Fig.2.30(a). The floating body captures wave energy, converting it into heave motion to drive the underwater propulsion mechanism through the connecting structure. Placing the glider in a deeper position helps to avoid surface wave turbulence and maintain stability. An AutoNaut, on the other hand, is an USV with submerged flapping foils fixed to the hull both at the bow and stern via a rigid arm, as shown in Fig.2.30(b). The motion of the vessel caused by waves drives the spring-loaded submerged foils, forcing them to flap at the frequency of the incoming waves. In recent years, multi-energy source NSVs have also been proposed, as demonstrated by Li et al. (110). These vessels can be powered by multiple natural energy sources, including wave, wind, and solar energy, as illustrated in Fig.2.31. When sailing at its design speed of 1.5 m/s, it is capable of absorbing natural energy to replenish its own energy consumption.



(a) Wave Glider (Qi et al. (37))

(b) Caravela (an AutoNaut vessel) (Siddle et al. (128))

Figure 2.30: The commercial USVs from Liquid Robotics and AutoNaut

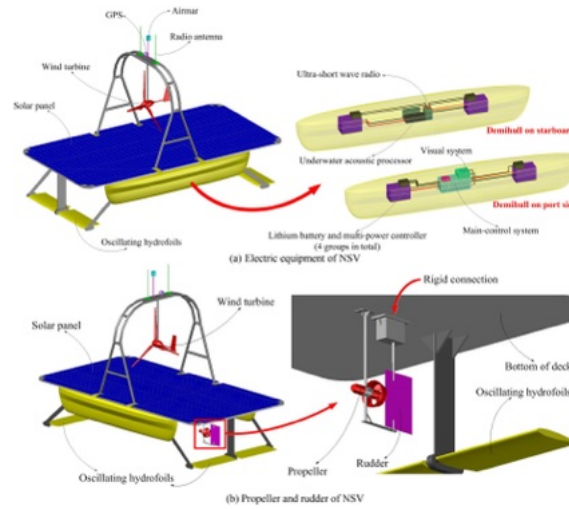


Figure 2.31: Basic structure and electric equipment of NSV. (Li et al. (110))

2.7.3 Foil with other structures

The utilisation of foils for wave propulsion is not limited to marine vessels alone. Another application of this technology is the wave energy converters (WECs). One such example is the wave-bladed cyclorotor, which uses foils to generate lift force by the wave-induced water velocities, and then rotates around an axis continuously. For instance, the LiftWEC concept, which uses two foils to spin, has been proposed as a novel WEC (129). Further information on cycloidal WECs can be found in works such as (129, 130).

Offshore platforms, including both offshore wind and offshore Photovoltaics (PV) platforms, operate in wave conditions and are subject to wave loads and wave-induced motions. These motions are undesirable as they increase maintenance and mooring costs. While WDPS has the potential to propel or stabilise floating systems using flapping foils, there have been no documented examples of WDPS being applied to offshore platforms. The authors believe that this is an area with significant potential and is in need of further research and development.

2.8 Conclusion

WDP has the potential to be highly efficient and environmentally friendly propulsion, but more research and development are needed to overcome the challenges and limitations. Future research directions include exploring the generation of the reversed von Kármán vortex street, understanding the foil-ship hull-wave interaction, the optimal number and placement of foils on ships, optimizing the ship size, foil size, and submergence depth, analysing the economic aspects of WDP, optimizing the 3D foil design, developing an efficient control mechanism, examining the effect of spring and wave conditions on passive foils, the investigation of foil flexibility effects on the performance enhancement and stability of the system and further developing applications in the offshore industry.

2.9 Acknowledgement

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Chapter 3

Advancing fluid-structure interaction (FSI) simulations in 2D: Insights from a three-phase model

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3.1 Abstract

This study presents a novel three-phase Fluid-Structure Interaction (FSI) model for simulating the containment of oil spills. The model uses Level Sets to capture the evolution of multiple interfaces and incorporates spring forces on the structure under hybrid wave-current boundary conditions. The implementation of spring forces has been validated through simple harmonic motion models while a wedge falling simulation demonstrates the model's ability to handle multi-phase deformation. The study compares numerical results with experimental data to study the response of oil spills to wave-current hybrid conditions. Our simulations reveal that when the current exceeds 0.2 m/s, the movement of the boom is dominated by the current and not by the waves or their inertia, providing important information for the design of effective oil spill containment systems.

Keywords: Multi-phase Level Sets; One-fluid formulation; Fluid-structure interaction (FSI); Numerical tank; Spring mooring.

3.2 Introduction

Fluid-Structure Interaction (FSI) involves the interaction between fluid flow and structural movement, where the fluid influences the structure's deformation and vice versa. The complexity of FSI increases significantly when multiple fluid phases are involved, as these different phases interact with each other and with solid structures. This type of multi-phase FSI is especially relevant in situations like offshore engineering, where oil, water, and gas may interact simultaneously with floating platforms and pipelines, or in biomedical engineering, where blood components interact with vascular walls (1, 2). Oil spills are a common occurrence in marine transportation, with a volume of over 15,000 tonnes reported in 2022 according to the International Tanker Owners Pollution Federation (ITOPF) (3). In addition, natural seepage accounts for 45% of the yearly release of petroleum into the environment (4, 5). These spills can cause severe pollution to freshwater and marine environments, posing a significant threat to wildlife and resources (6, 7). To mitigate the spread of oil spills, the use of oil containment booms is common practice (8). These booms aggregate waterborne pollutants, facilitating the isolation and disposal of contaminants. The schematic diagram for the interaction between the rigid boom and the ocean is shown in Fig.3.1. The oil containment process is a multi-phase fluid-structure interaction (FSI) problem, involving the evolution of multiple fluid and solid interfaces, fluid-structure interaction with spring forces, hybrid wave-current boundary conditions, and the design of the boom, which is modelled as a rigid structure.

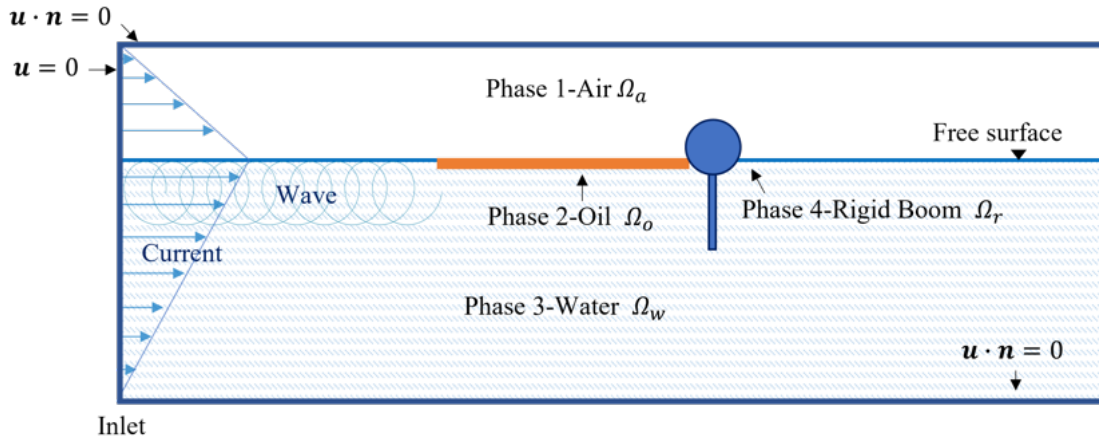


Figure 3.1: Schematic diagram of the oil boom simulation.

The system in Fig.3.1 includes four phases, air Ω_a , oil Ω_o , water Ω_w and rigid boom Ω_r . The whole domain can be written as $\Omega = \Omega_a \cup \Omega_o \cup \Omega_w \cup \Omega_r$.

We present the use of the numerical wave tank (NWT) to simulate the multi-phase fluid-structure interaction problem in oil spill containment. The system in the NWT consists of multiple regions (Ω_a , Ω_o , Ω_w , and Ω_r) that are separated by their interfaces. The method of capturing these interfaces can be either mesh-based or meshless (9). The Level Set Method (LSM) is a well-established mesh-based method introduced in 1988 (10). In the past three decades, quite a few researchers have made more in-depth developments on LSM and it is widely used in computational geometry (11), microfabrication (12), fluid dynamics (13), image processing (14), and computer vision (15). Apart from LSM, the

Volume of Fluid (VOF) (16, 17, 18) is another mesh-based Eulerian method for interface capturing (19). VOF binds the continuity, momentum and energy conservation status using volume fraction equations. This method is very convenient for the study of two-phase flow, with various applications in cloud modelling (20), bubble motion (21), fuel cells (22) etc. However, due to the discontinuity of the fraction function, it is difficult to achieve high-order accuracy (higher than two) of the curvature. Smoothed Particle Hydrodynamics (SPH) is a popular approach within meshless methods. The concept was originally proposed to solve interstellar hydrodynamic problems in astrophysics (23) and has since been extended to cover many other research areas such as solid mechanics (24, 25), multi-phase flows (26), granular media flows (27), heat conduction (28), high speed impacts (29) and underwater explosions (30), etc. However, the fluid is weakly compressible (31). For explicit accurate solutions, the time step (Δt) needs to be small (32). It is also due to the nature of particles that it is relatively difficult to deal with boundary conditions, especially inlets, outlets and pressure boundary conditions, compared to mesh-based methods (33).

The simulation of fluid-structure interaction in numerical simulations can be performed using two approaches, namely the body/boundary fitted methods and the immersed type methods (34). The body/boundary fitted approach updates the mesh as the interface changes (35). However, this method is computationally expensive, especially when working in 3D (36). In contrast, the immersed type methods are performed on a fixed Eulerian Cartesian grid and do not require mesh updates or regeneration as the interface evolves (36, 37, 38). The original concept of the Immersed Boundary Method was developed by Peskin (36) and involves the representation of the valve's motion through a force field. The method is originally limited to solids satisfying Hooke's law (39, 40, 41, 42), but later, with the consideration of strain energy, it is expanded to include immersed hyperelastic solids (43). Currently, the solid structure can be extended to any continuum through the introduction of the potential energy equation (44). The fluid's spatial discretization can be built on either structured Eulerian grids or unstructured grids and adaptive local mesh refinement can be used to improve the accuracy of the method (8, 45). To address excessive numerical diffusion caused by the interpolation/spreading process, the Immersed Structural Potential methods were proposed (46, 47).

The 'one-fluid' formulation is a recent addition to the family of immersed methods in fluid-structure interaction simulation (34, 48, 49, 50, 51, 52). This approach differs from traditional methods, where the dynamics of a structure are represented by a set of Newton-Euler equations, by using Lagrange multipliers to constrain the rigid motion and treating the simulated objects as fluids. The simulations are conducted on a fixed Eulerian Cartesian grid, and the Heaviside/LSM is used to capture the fluid-structure interface and eliminate indeterminate regions between the two phases. The motion of the rigid body is modelled using weighted linear least squares (49). The main advantage of the one-fluid approach is its computational efficiency, as it only requires a fluid solver and can handle topological changes without the need for mesh updates or re-meshing.

This paper focuses on investigating the hydrodynamic fluid-structure interaction (FSI) model using the one-fluid framework and examining its response to oil booms. The unique aspect of this study is modelling the behaviour of the mooring-boom system through a spring-mass system. The pendulum case demonstrates that the boom can rotate around a fixed point freely, while the combination of vertical and horizontal motions limited by elastic forces simulates the expected motion of a floating boom affected by waves and

currents. The wedge-dropping case is designed to analyse the interaction between the rigid structure and interface deformation. The fluids in this study are treated as viscous, with clear interfaces, isothermal, and negligible surface tension effects.

The paper is organized as follows. Section 3.2 introduces a detailed background multi-phase FSI model. Section 3.3 presents a comprehensive description of the numerical method. The solutions to multi-phase interaction and computational equations are proposed. The governing equations, interface descriptions and boundary conditions are defined using the one-fluid, three-phase level set, and hybrid conditions methods. Two Simple harmonic motion (SHM) cases and the wedge-dropping model have been successfully tested using these methods. Specific descriptions and analysis are given in section 3.4.1 and section 3.4.2, respectively. Section 3.5 represents the simulation of the rigid boom in response to oil spills at sea under wave-current conditions. Finally, in section 3.6, the conclusions are summarised.

3.3 Numerical solver

The numerical solver used in this study is an ‘in-house’ developed code that employs the ‘one-fluid’ formulation, recently proposed by Yang et al. (34, 49, 50, 52). The ‘one-fluid’ formulation is a part of the immersed methods family. In contrast to traditional approaches, which typically represent structural dynamics through a set of Newton-Euler equations, this formulation utilises Lagrangian multipliers to impose constraints on motion, thereby ensuring rigidity. Simulations are performed on a fixed Eulerian Cartesian grid with a uniform mesh size. The ‘one-fluid’ formulation uses the Level Set Method (LSM) to capture the interface between different phases. Compared to other formulations, a key advantage is it only uses fluid solver for the FSI. Additionally, this formulation does not need mesh updating or re-meshing. Both of these features contribute to improved computational efficiency.

Fig.3.2 shows the calculation logic of ‘one-fluid’ under one single step, which is validated through several benchmarks and experimental tests (53). It is worth mentioning the part related to the solution of the velocity field. The framework considers the elastic force as the external force of the system. The connection between the spring force F_{ext} and the velocity u_r of the rigid body can be described as Eq.3.1, according to the law of conservation of linear momentum.

$$F_{total} = F_r + F_{ext} = m \frac{du_r}{dt} + m \frac{du_{ext}}{dt} \quad (3.1)$$

where F_{total} is the total force on the rigid body at each time step Δt , u_{ext} is the velocity generated by the external force (i.e. spring force in this case). For more information, please refer to paper (53).

The simulation platform in this study employs a high-performance computing system featuring an Intel Core i7-10700 CPU 2.90GHz, 32GB of RAM, a 954GB SSD, and a 1.8TB HDD, running on Windows 10 Enterprise (version 21H2). This setup provides robust computational power and ample memory and storage capacity, essential for conducting CFD simulations efficiently.

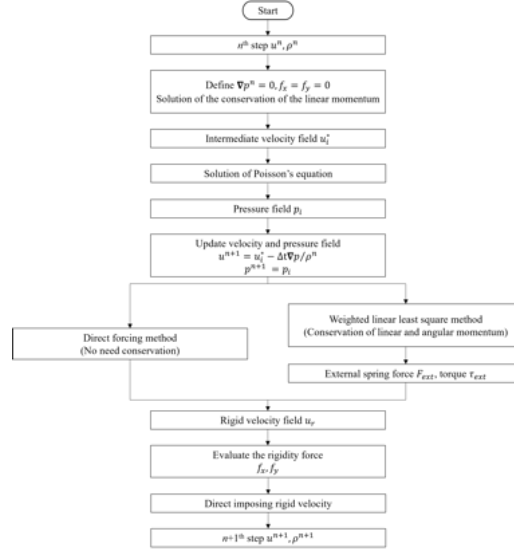


Figure 3.2: The flow chart of the ‘one-fluid’ framework.

3.3.1 Governing equation – ‘one-fluid’ formulation

The dynamics of the four-phase system are governed by incompressible Navier-Stokes equations. Ignoring the surface tension, the conservation of the linear momentum and mass is given by Eq.3.2. The different material properties are described by the deviatoric stress f , see Eq.3.2c.

$$\rho \left[\frac{\partial u}{\partial t} + (\nabla u)u \right] = -\nabla p + f + \rho g \quad (3.2a)$$

$$\nabla \cdot u = 0 \quad (3.2b)$$

$$f = \begin{cases} \nabla \cdot 2\mu_a \mathbf{D}(u) & \text{in } \Omega_a \\ \nabla \cdot 2\mu_w \mathbf{D}(u) & \text{in } \Omega_w \\ \nabla \cdot 2\mu_o \mathbf{D}(u) & \text{in } \Omega_o \\ \rho \left[\left(\frac{\partial \mathbf{P}(u)}{\partial t} \right) + (\nabla \mathbf{P}(u))\mathbf{P}(u) \right] + \nabla p - \rho g & \text{in } \Omega_r \end{cases} \quad (3.2c)$$

where $\mathbf{D}(u) = \frac{1}{2} (\nabla u + (\nabla u)^T)$ is the strain rate tensor and $\rho = \rho_a H_a + \rho_w H_w + \rho_o H_o + \rho_r H_r$ is the density. The symbols a , w , o and r stand for air, water, oil and rigid boom. u is the velocity vector field, p is the pressure, f is the force on each phase Ω , g is the gravitational acceleration, μ is the dynamic viscosity. The characteristic function χ of any region Ω_α can be uniformly expressed as

$$\chi = H_\alpha(\phi) \quad \alpha = a, w, o, r \quad (3.3)$$

where

$$H_\alpha(x) = \begin{cases} 1 & \text{if } x \in \Omega_\alpha \\ 0 & \text{if } x \notin \Omega_\alpha \end{cases} \quad (3.4)$$

is a Heaviside function for any phase α (49). $x = (x, y)$ is the coordinate of any point in the domain. H_α is introduced to identify different phase regions. Once the Heaviside function has been constructed, the corresponding density and viscosity field can be assigned.

3.3.2 Multi-level Set Method

3.3.2.1 Level Set Method

LSM describes the interfaces implicitly. An object is represented by a zero contour of a level set function (LSF), which is one dimension higher than the object (54). The LSF itself is a signed distance function (SDF) (55). The evolution of the SDF formulates the motion of this object (56), see Eq.3.5.

$$\phi(x) = \text{sgn}d(x, \Gamma) \quad (3.5a)$$

$$\Gamma = \{x \in \mathbb{R}^3 \mid \phi(x, t) = 0\} \quad (3.5b)$$

$$\phi(x, t) \begin{cases} > 0 & x \in \Omega_1 \\ < 0 & x \in \Omega_2 \end{cases} \quad (3.5c)$$

The value of ϕ in Eq.3.5a is the shortest distance from a point x to the interface Γ . Taking a two-region system as an example (Eq.3.5c), the sign of this function represents two regions with one interface Γ , positive for region Ω_1 and negative for region Ω_2 .

In non-conservative form, the evolution equation of ϕ with time can be defined by the following Eq.3.6a. The LSF for three materials can be expressed as Eq.3.6b.

$$\frac{\partial \phi(x, t)}{\partial t} + u(x, t) \cdot \nabla \phi(x, t) = 0 \quad (3.6a)$$

$$\frac{\partial \phi_\alpha(x, t)}{\partial t} + u(x, t) \cdot \nabla \phi_\alpha(x, t) = 0 \quad \alpha = 1, 2, 3 \quad (3.6b)$$

where $u(x, t)$ is the velocity of the interface. As the interface evolves, the LSF loses its signed property. To maintain this property and guarantee mass conservation, the LSF is reinitialised after each time step (57).

3.3.2.2 3-phase Level Set Method

For one-phase LSM, the expression of SDF ϕ is divided into two groups due to the location of x , as shown in Fig.3.3(a). ϕ is a negative number when x is inside the target region and a positive number outside. The same procedure can be adopted if two regions are within the research domain. Only need to set

$$\phi_1 = \phi \quad (3.7a)$$

$$\phi_2 = -\phi. \quad (3.7b)$$

They represent two regions in one domain, as shown in Fig.3.3(b). Region 1 and region 2 correspond to those negative points in Ω_1 and Ω_2 .

When extending to an arbitrary number of regions, one LSF per region is supposed to be adopted for convenience and intuitively maintain the depictions. There should be only one ϕ less than 0 (negative) in one region, while all other ϕ should be greater than 0 (positive). When the phases increase to 3 or more, and the ordinary LSM is used, the ideal distribution of regions can be demonstrated in Fig.3.4(a). However, an arbitrary ϕ will introduce the vacuum and overlap (Fig.3.4(b)). It is due to the generation and accumulation of numerical errors approximated by numerical solutions during the evolutionary

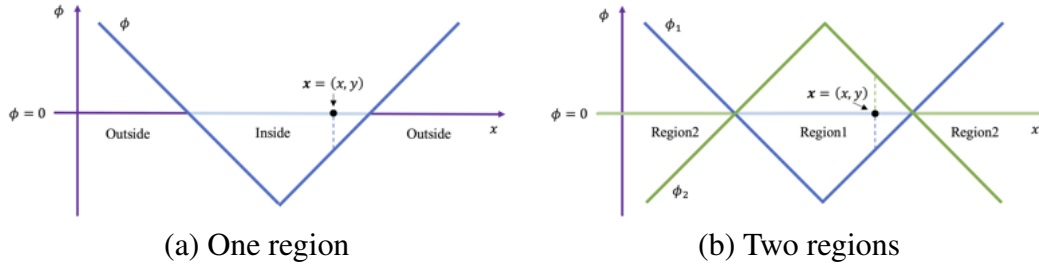


Figure 3.3: Illustration of LSM in less than two regions.

process. More specifically, the vacuum appears when none of ϕ is negative, as shown in Fig.3.5(a). No ϕ lays claim that this area belongs to any region in the domain. On the contrary, the overlap appears when more than one of ϕ is negative, as shown in Fig.3.5(b). That indicates this area belongs to more than one region in the research domain. Vacuum and overlap are abnormalities during the simulation, which need to be avoided.

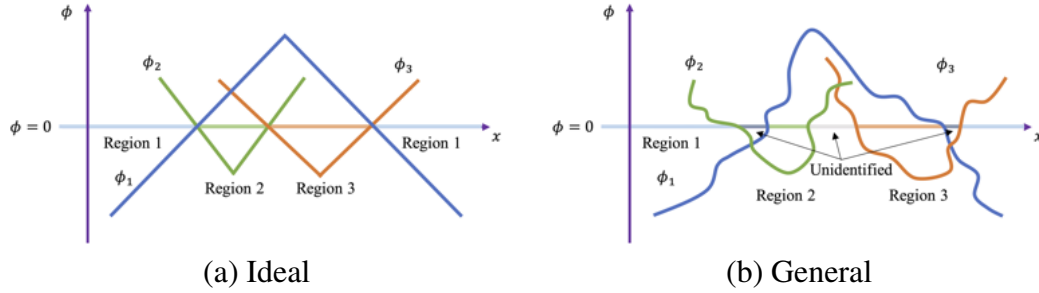


Figure 3.4: Comparison of the ideal and the general situation in Multi-LSM.

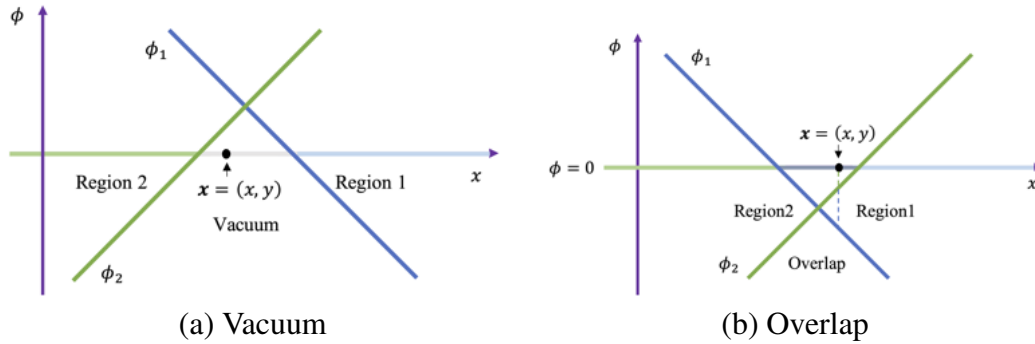


Figure 3.5: Illustration of the abnormalities in Multi-LSM.

Combining the previous experience of other researchers (8, 58, 59, 60, 61, 62, 63), we will use a projection scheme for the multi-LSM. To distinguish the location of the interface, each domain can be defined by the smallest level set function. For any point x , if $\forall i \neq j, \phi_i < \phi_j$, then x is one point from region i . In this way, vacuum and overlap do not occur whether there are no negative ϕ or multiple negative ϕ for a given region, see Fig.3.6(a) and 3.6(b). A projection method is proposed and applied before the standard LSM. Briefly, this projection scheme has two steps. First, find and compute the average of the two smallest level set values ϕ for every location x (Fig.3.6(c)); Second, subtract

this average from the original ϕ (Fig.3.6(d)). Each level set can be transformed into ideal cases, like Fig.3.4(a). The following Fig.3.7 provides a visual representation of the

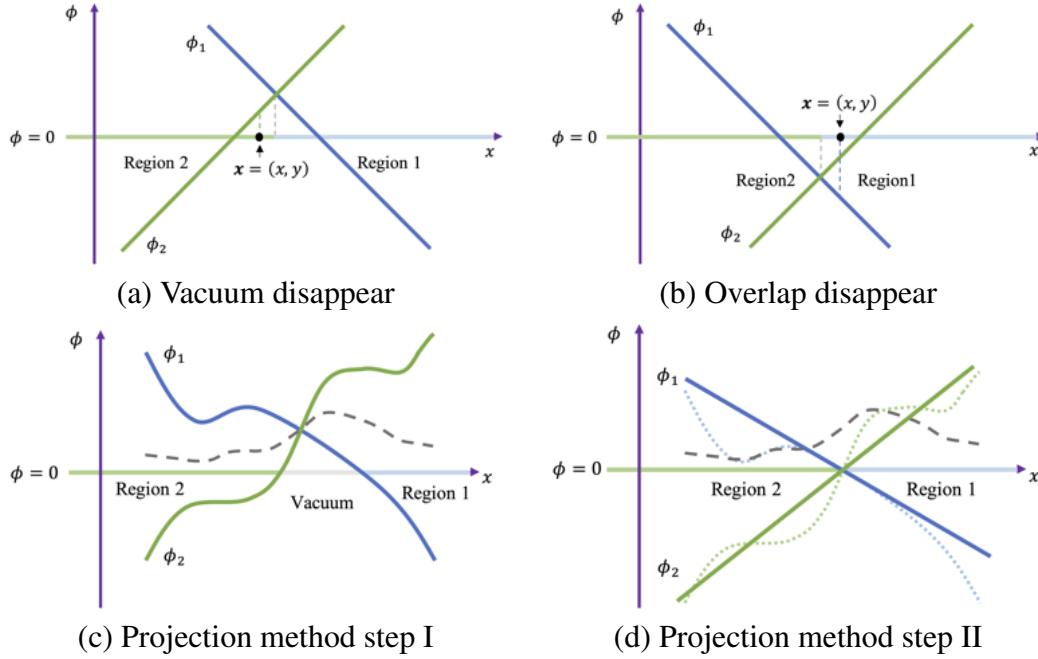


Figure 3.6: Illustration of the Multi-LSM.

combination of the multi-LSM and the Heaviside Functions applied to the oil containment process model.

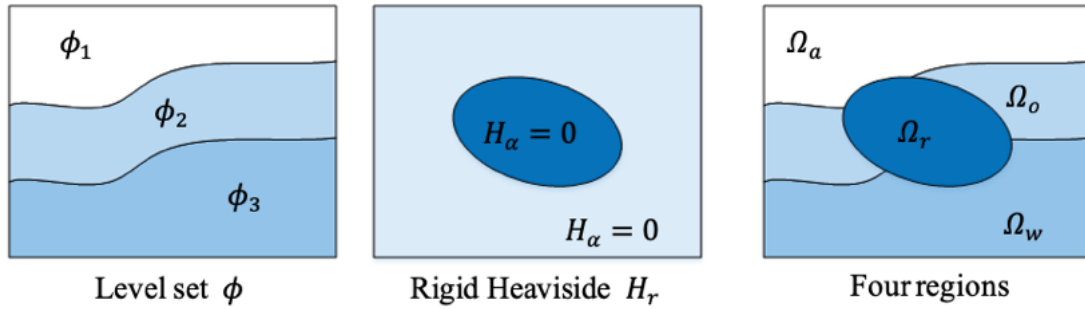


Figure 3.7: The combination of the Heaviside Function and the LSM.

In summary, we will use LSM to represent interfaces. It can automatically handle topological changes (34) and make it easier to calculate the normal direction and curvature of the interface. Special attention is needed for multiple Level Sets because the vacuum and overlap will appear during the simulation (61). Projection and initialisation procedure is added for every time step.

3.3.3 Numerical and discretisation scheme

The Cartesian staggered Finite Volume scheme (Marker-and-Cell grid) is adopted for the spatial discretisation of the study domain. For example, Fig.3.8 shows one zoom-in part

from the most refined Cartesian grid for the boom simulations. The convective terms of Navier-Stokes equations are discretised with a second-order Total Variation Diminishing (TVD) Runge-Kutta scheme in the conservative framework. The fourth-order accurate Runge-Kutta scheme processes the time treatment of level set equations. Adaptive time steps are applied to govern the Courant-Friedrichs-Lewy (CFL) number. The CFL number is kept as 0.5 in this paper, which will not be repeated.

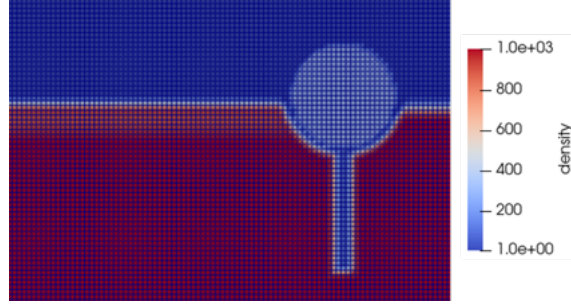


Figure 3.8: Part Cartesian grid from the finest case for the rigid boom simulation ($\Delta x = 0.0042$ m).

3.3.4 Boundary conditions

Dirichlet type, the common inlet boundary condition for free surface flow simulations, is considered the boundary condition. Dirichlet boundary condition specifies the values of related physical quantities at the boundary. For the oil containment process model, the initial wave and current profiles are based on experimental inlet settings. Wave is generated from the point of air/water interface at the inlet, and this point is set to be the origin. Its constant motion propagates through the free surface in the positive x-axis direction, with the flow direction changing periodically. This method can generate both regular and irregular waves through the velocity equation description, but this paper only considers regular waves to maintain consistency with the experiment. In terms of the linear airy wave, the horizontal velocities u , vertical velocities v and wave profile ζ at time t can be expressed as follow (Eq.3.8) (57).

$$u(x, z, t) = \frac{\pi H}{T} \frac{\cosh[k_w(z + d)]}{\sinh(k_w d)} \cos \theta \quad (3.8a)$$

$$v(x, z, t) = \frac{\pi H}{T} \frac{\sinh[k_w(z + d)]}{\sinh(k_w d)} \sin \theta \quad (3.8b)$$

$$\zeta(x, z, t) = \frac{H}{2} \cos \theta - z + d \quad (3.8c)$$

where d is the initial still water level (water depth), and z is the vertical coordinate of the wave forefront. Regarding the characteristic parameters, H is the wave height, T the wave period, $\theta = k_w x - \omega_w t$ the wave phase, $k_w = \frac{2\pi}{L}$ the wave number, L the wavelength, $\omega_w = \frac{2\pi}{T}$ the angular wave frequency. Their relationship follows the linear dispersion equation, Eq.3.9.

$$L = \frac{gT^2}{2\pi} \tanh(k_w d) \quad (3.9)$$

The current profile (Eq.3.10) is described by a linear equation with two segments.

$$u_c = -\frac{v_0}{s_0}s + v_0 \quad (3.10)$$

where u_c is the current velocity at any point; the value of u_c achieves its maximum v_0 at the air-water interface; s_0 is the thickness of the single fluid layer (air or water); s is the vertical distance from the point at the left boundary to the original air-water interface. The numerical walls of the flow domain are considered to be smooth and have the no-slip condition. The inlet of the current-wave numerical tank is the left boundary, while the right boundary is an open outlet for the flow instead of a closed wall. The sum of the regular wave and current constitutes the hybrid boundary condition. The vivid expression of the hybrid boundary conditions can be found on the left side of Fig.3.1.

The ghost cell method was employed to enforce boundary conditions. A no-slip boundary condition was enforced at the bottom of the numerical wave tank by setting the horizontal and vertical velocity components (u and v) to zero in the ghost cells adjacent to the boundary.

3.4 Benchmark cases

3.4.1 The SHM simulation

3.4.1.1 Introduction

This section aims to verify the linear mooring force on the rigid structure (boom). The motion constraints include translation under spring forces and free rotation. Under the action of restoring force, the reciprocating motion of an object near its equilibrium position is called Simple Harmonic Motion (SHM). The object is subjected to the restoring force F . The force can be expressed by Eq.3.11.

$$F = -kx \quad (3.11)$$

The force magnitude is proportional to the object displacement x , and the direction always points to the equilibrium position, where $F = 0$. For example, the simple pendulum movement, molecular microscopic motion, the movement of the object subjected to the force of the spring, et al. Such motion is characterized by periodicity. The general expressions of object x - t are

$$x = A \cos(\omega t + \varphi_0) \quad (3.12)$$

where x is the object's displacement, A is the amplitude, ω is the number of complete vibrations within 2π seconds, φ_0 is the initial phase when time $t = 0$. The vibration period of the object can be expressed by $T = \frac{2\pi}{\omega}$. The angular frequency ω of this mechanical system is

$$\omega = \sqrt{\frac{k}{m}} \quad (3.13)$$

which is determined by the mechanical properties of the motion system itself. m is the mass of the object.

3.4.1.2 The spring-mass oscillator

A system with a spring linked to an object is called a mass spring. Within the elastic limit, the elastic force of the spring satisfies Hooke's law. According to Eq.3.11, k is the spring stiffness coefficient. Fig.3.9 depicts the motion of the spring-mass system over one period. In order to replicate SHM motion and build a spring-mass model, we abstract the spring by removing its entity and retaining only the characteristics of the force. That is to say, the spring itself is invisible, while the force exerted on the object will remain. The object is initially positioned at any one point in the domain. The deviation of the object x is marked with the distance from this specified point to the position at time t . The direction always points to the position calibrated at the initial moment.

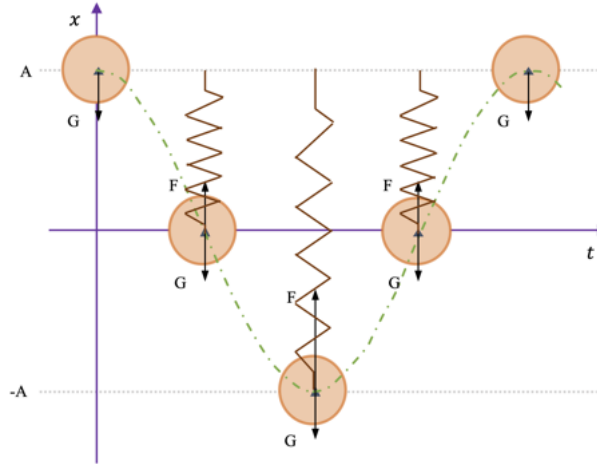


Figure 3.9: A schematic representation of the variation of displacement over one cycle of the spring-mass system.

The specific velocity and acceleration variations of the motion are not repeated here. When the acceleration is zero, the gravitational and restoring force on the object are balanced, and the ideal amplitude without damping can be obtained by

$$A = \frac{mg}{k} \quad (3.14)$$

The initial phase φ_0 is 0. g is the acceleration of gravity. The effect of the resistance is taken into account and is numerically summarised as the damping ratio ζ . Therefore, in this case, the Eq.3.12 evolves into

$$x = \frac{mg}{k} e^{-\zeta t} \cos \left(\sqrt{\frac{k}{m}} t \right) \quad (3.15)$$

The framework considers the elastic force as the external force of the system. The connection between the spring force F_{ext} and the velocity u_r of the rigid body can be described as Eq.3.16, according to the law of conservation of linear momentum.

$$F_{total} = F_r + F_{ext} = m \frac{du_r}{dt} + m \frac{du_{ext}}{dt} \quad (3.16)$$

Table 3.1: Grid resolution for numerical comparison of spring-mass model.

	Mesh size (m)	Numbers of cells ($x \times y$)
1	$\Delta x = \frac{1}{320} = 3.13 \times 10^{-3}$	80×128
2	$\Delta x = \frac{1}{640} = 1.56 \times 10^{-3}$	160×256
3	$\Delta x = \frac{1}{1280} = 7.81 \times 10^{-4}$	320×512

where F_{total} is the total force on the rigid body at each time step Δt , u_{ext} is the velocity generated by the external force (i.e. spring force in this case).

The target of the simulation in this section is the spring-mass system. The three-dimensional simulation system is conceptualized into a two-dimensional circle with the ‘invisible’ force F_{ext} . The whole domain is filled with still air. Check Tab. 3.3 for its physical characteristics. In order to eliminate the effect of domain boundaries, the space around the object is deliberately left enough for its motion. This simulated object has 1 DOF, i.e. translation in the vertical direction. The main concern is the effectiveness of the simulation in applying the elastic force.

The domain size is $0.25 \text{ m} \times 0.40 \text{ m}$. The object has a mass m of 1 kg and a radius r of 0.02 m. The gravitational acceleration g is 9.81 m/s^2 . The spring stiffness is set to 100 N/m. The value of the damping ratio ζ , in this case, is approximately 0.05, according to multiple tests. Refer to the Eq.3.15, the theoretical displacement-time results of this simulation can be expressed by Eq.3.17. The mass vibrates for a period of $T = \frac{2\pi}{\omega} = 0.63 \text{ s}$.

$$x = 0.098e^{-0.05t} \cos(10t) \quad (3.17)$$

To verify the convergence and accuracy of the numerical results, the simulation was conducted three times using structured grids of varying sizes. More details about grids can be found in Tab. 3.1. Specifically, regarding conservation, mass is inherently satisfied because the system involves only solid and liquid phases. The solid phase is directly defined, ensuring mass conservation for the fluid phase as well.

Fig.3.10 shows the results of the numerical SHM displacement of the mass over more than ten natural periods for three grid conditions. It can be seen that the results of different grid conditions tend to converge instead of diverging at each time step, indicating that this SHM numerical model has excellent convergence. Furthermore, the numerical results are in good agreement with the theoretical results, where the data for the ascending and descending motion of the object almost coincide.

From the second period and after ($t > 0.63 \text{ s}$), the mass is incapable of returning to the starting position. This exemplifies the influence of the frictional damping of the surrounding air and the numerical damping of the simulation model. Besides, the frequency of the object motion does not vary significantly while the amplitude gradually decreases. In addition to the mutual conversion of kinetic and potential energy, the object constantly consumes energy to compensate for the energy dissipated by the damping. Ultimately, the object is expected to become stationary, and the spring-mass system reaches a steady state, which is a normal phenomenon in reality. If the spring-bound object is required to return to stationary more rapidly, it is only necessary to increase the spring stiffness. This phenomenon is exactly what we expect and is an essential theoretical base for why the spring system can be considered an under-damped harmonic oscillator. The spring’s restoring force can be mounted on the object as a mooring force in the following CFD

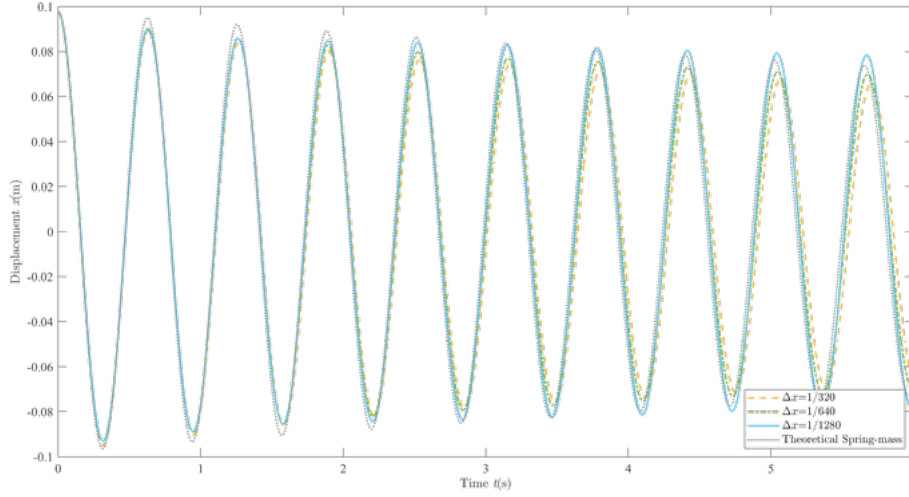


Figure 3.10: Validation of convergence of spring-mass simulations and comparison with theoretical solutions ($\zeta=0.05$).

simulation.

3.4.1.3 The simple pendulum

The simple gravity pendulum is an ideal physical model. A point mass (such as a ball in 3D or a circle in 2D) is connected by a massless cord and suspended from a fixed point in the gravitational field, see Fig.3.11. When the mass is pulled away from the equilibrium position, i.e. the angle θ between the cord and the plumb line across the point of suspension $\neq 0$, the mass vibrates back and forth under the influence of the net restoring force $F = mg \sin \theta$. The time of one vibration is one period T . Generally speaking, when the angle θ is less than 5° , $\sin \theta \approx \frac{x}{l}$, the motion of the pendulum is linearly approximated as SHM. In this case, the restoring force is $F = -mg \frac{x}{l}$, the coefficient is $k = \frac{mg}{l}$, where m is the body mass, g is the acceleration of gravity, x is the displacement and l is the cord length, as shown in Fig.3.11. In summary, the motion of a single pendulum is described by the formula as Eq.3.18.

$$A = \theta_0 \quad \theta|_{t=0} = \theta_0 \quad (-5^\circ \leq \theta_0 \leq 5^\circ) \quad (3.18a)$$

$$\theta = \theta_0 \cos \left(\sqrt{\frac{g}{l}} t + \varphi_0 \right) \quad (3.18b)$$

$$T = \frac{2\pi}{\omega} = 2\pi \sqrt{\frac{l}{g}} \quad (3.18c)$$

where the amplitude A is the magnitude of the initial pendulum angle θ_0 . The initial phase is $\varphi_0 = 0$. The period T is only dependent on the length l and the local acceleration of gravity g , and is irrelevant to θ_0 .

The single pendulum model has 1 DOF, i.e. the rotational motion. It does not translate in the horizontal or vertical direction. The massless cord is not visually labelled; only the fixed centre of rotation and its tension are retained.

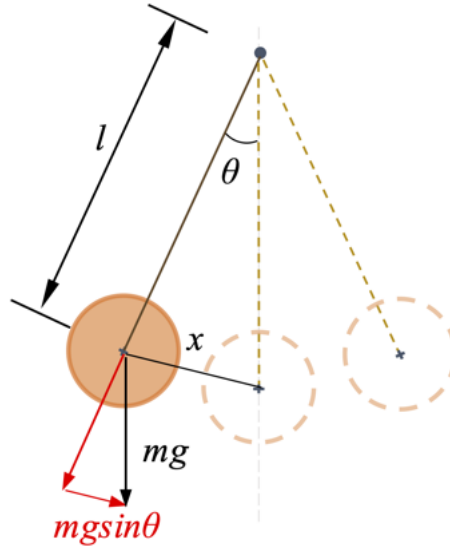


Figure 3.11: Single pendulum model approximated as SHM.

Table 3.2: Grid resolution for numerical comparison of pendulum model.

	Mesh size (m)	Numbers of cells ($x \times y$)
1	$\Delta x = \frac{1}{640} = 1.56 \times 10^{-3}$	192×128
2	$\Delta x = \frac{1}{1280} = 7.81 \times 10^{-4}$	384×256
3	$\Delta x = \frac{1}{2560} = 3.91 \times 10^{-4}$	768×512

The domain size is $0.30 \text{ m} \times 0.20 \text{ m}$. The object has a mass m of 1 kg and a radius r of 0.01 m. The cord length l is 0.5 m. The initial pendulum angle θ_0 is -5° . The value of the damping ratio ζ , in this pendulum model, is approximately 0.14, according to multiple tests. Refer to the Eq.3.18, the theoretical angle of rotation-time results for this simulation can be expressed by Eq.3.19. The mass vibrates for a period of $T = 1.42 \text{ s}$.

$$\theta = -5e^{-0.14t} \cos(4.43t) \quad (3.19)$$

The convergence is tested three times as well. The information of grids and the result is illustrated with Tab. 3.2 and Fig.3.12.

Fig.3.12 demonstrates that as the grid is doubled and refined, the angle-time ($\theta - t$) curves converge more and more closely towards the theoretical values. In particular, it shows an extremely high degree of coincidence at $\Delta x = \frac{1}{2560} \text{ m}$. This proves the convergence of the present simple pendulum model. The mass is subject to the cord and gravity, oscillating near its equilibrium position around a fixed point. After one cycle of oscillation, the object returns to the position where it started to be released.

3.4.2 The dropping wedge simulation

3.4.2.1 Introduction

This chapter explores the capabilities of the one-fluid method for simulating fluid-structure interaction in multi-phase fluid environments. The wedge-dropping case is used as a test

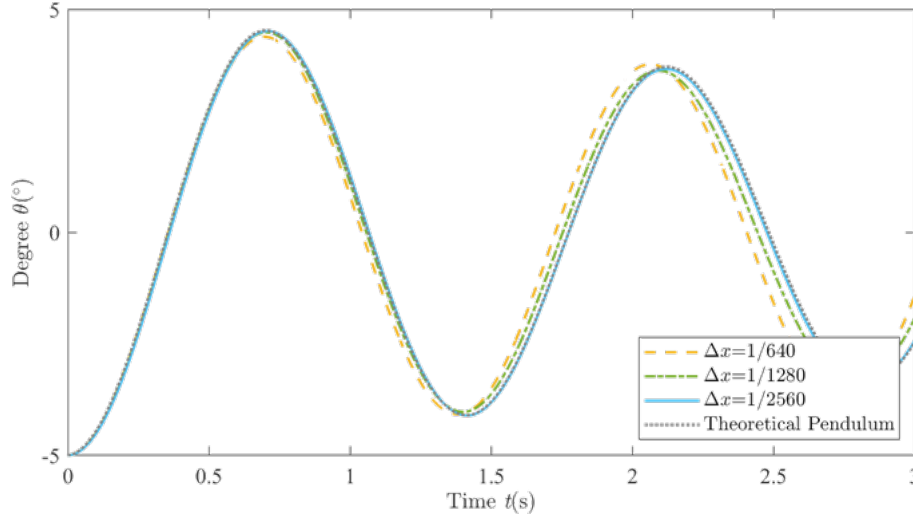


Figure 3.12: Validation of convergence of pendulum simulations and comparison with theoretical solutions ($\zeta=0.14$).

scenario to demonstrate the accuracy of the one-fluid method in modelling the interactions between rigid objects and multi-layer fluids. The wedge, as a solid body, is dropped into a calm state of a two-layer oil-water fluid, causing a disturbance and resulting in splashing. The numerical simulation will examine the effects of various physical parameters, such as the density and velocity field, on the dynamics of the fluid and the solid.

3.4.2.2 Numerical model setup

In this section, we present the numerical model setup for simulating the free-fall movement of a wedge-shaped object into a two-layer oil-water fluid system. Our model is based on the experimental setup conducted by the Tianjin Research Institute for Water Transport Engineering (TIWTE) (26). Our model is set up the same as the experiments conducted by TIWTE. Fig.3.13 is a schematic diagram of the initial state position of the wedge and fluid. The entire study domain size is $0.40 \text{ m} \times 0.22 \text{ m}$. The upper air part is 0.07 m thick, the middle oil layer is 0.05 m thick, and the lower water part is 0.10 m thick. The physical parameters of the oil used in the simulation are consistent with corn oil, as used in the TIWTE experiments, and are detailed in Tab. 3.3.

The wedge is modelled as a freely falling object with 3 degrees of freedom (DOF), i.e. translation in the vertical and horizontal direction, and rotational motion. The wedge has an isosceles right triangle cross-section with a right-angled side of 0.03 m and a mass of 0.2 kg . Its density is 2339.18 kg/m^3 , and it starts at a distance of 0.03 m from the air-oil surface.

In order to prevent reflections from the boundaries, absorption zones with a thickness of 0.02 m are placed on the lower, left, and right sides of the domain using the relaxation method proposed by H.Bihs, et al. (57). The velocity components inside the absorption

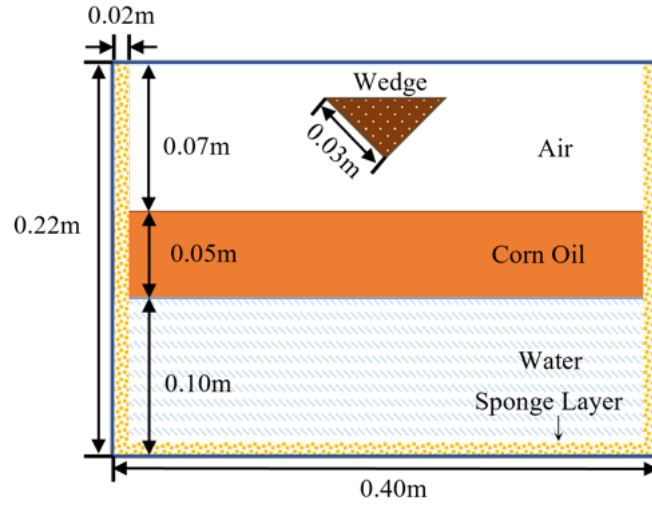


Figure 3.13: A schematic diagram of the wedge dropping simulation.

Table 3.3: Physical factors of the elements in the models.

	Density ρ_α (kg/m ³)	Kinematic viscosity ν (m ² /s)	Dynamic viscosity μ (kg/m · s)
Air (20°C)	1.204	1.516×10^{-5}	1.825×10^{-5}
Water	1000	1.000×10^{-6}	1.000×10^{-3}
Corn Oil	848	7.000×10^{-5}	5.936×10^{-2}
CKC680	890	1.625×10^{-3}	1.446

zone are relaxed using the equation in Eq.3.20.

$$p(\tilde{x}) = 1 - \frac{e^{\tilde{x}^{3.5}} - 1}{e - 1} \quad \text{for } \tilde{x} \in [0, 1] \quad (3.20a)$$

$$u_r = p(\tilde{x})u_o \quad (3.20b)$$

$$v_r = p(\tilde{x})v_o \quad (3.20c)$$

where $p(\tilde{x})$ is the relaxation function and $\tilde{x}$ is a scale factor, i.e. the distance from the wave inlet boundary to the absorption zone over the entire domain length. u_r and v_r are the relaxed velocity components inside the absorption zone in the horizontal (x) and vertical (y) directions, respectively, while u_o and v_o are the original computational ones without relaxation.

The simulation was run on various grid sizes, as detailed in Tab. 3.4, and the results are analysed in Fig.3.14 and Fig.3.15.

In summary, the numerical model setup in this section accurately represents the experimental setup, while incorporating appropriate boundary conditions to ensure accurate and reliable results.

3.4.2.3 Result and discussion

Fig.3.14 presents a series of snapshots of the wedge-dropping experiment (first row), simulation results (second row) and the corresponding velocity field (third row). The com-

Table 3.4: Grid resolution for numerical comparison of wedge dropping model.

	Mesh size (m)	Numbers of cells ($x \times y$)
1	$\Delta x = \frac{1}{300} = 3.33 \times 10^{-3}$	120×66
2	$\Delta x = \frac{1}{600} = 1.67 \times 10^{-3}$	240×132
3	$\Delta x = \frac{1}{1200} = 8.33 \times 10^{-4}$	480×264

parison charts provide a clear visual representation of the wedge's interaction with each phase at different moments. The wedge block maintains its degrees of freedom in all three directions, with no sway or rotation detected. This indicates that the horizontal direction force and the turning moment to which the wedge block is subjected are balanced. The acceleration is generated by the difference between the self-gravity and the buoyant force of the fluid. The fluid movement at the wall is effectively reduced.

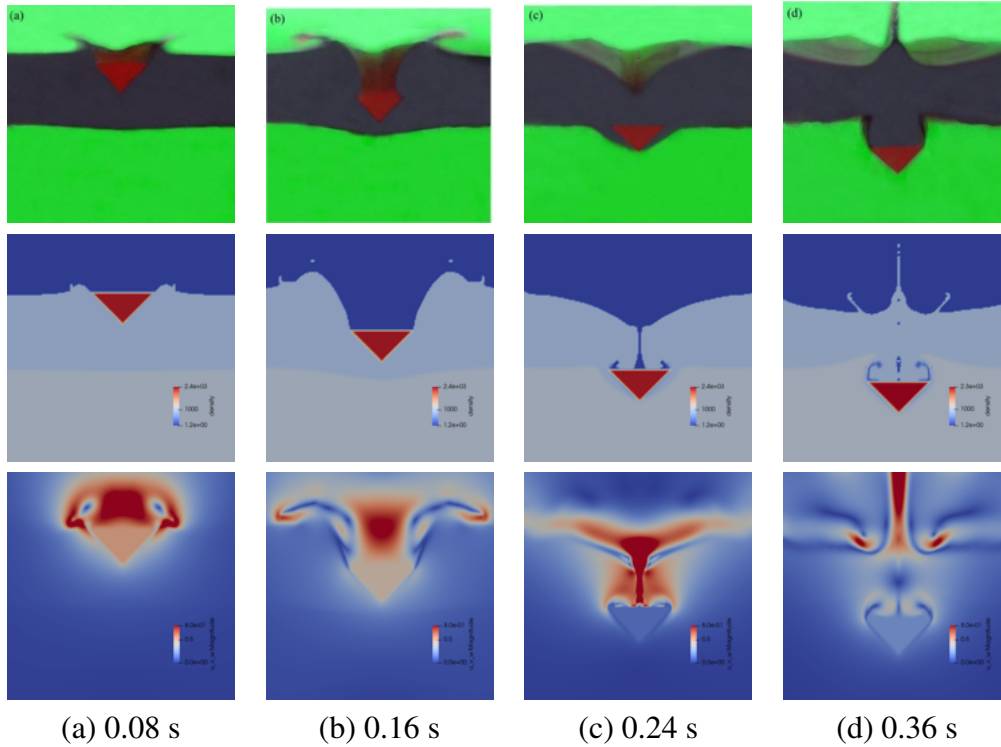


Figure 3.14: Comparisons between the experiment (the 1st row) and wedge-dropping simulation ($\Delta x = \frac{1}{1200}$ m).

The displacement D and velocity U of the simulated wedge over time are shown in Fig.3.15 and demonstrate reasonable agreement with the TIWTE experiment (26). As the mesh size decreases, the differences among the results become smaller, indicating a convergent drop wedge model. The slight quantitative differences near the maximum velocity can be attributed to either the 2D simulation or numerical viscosity.

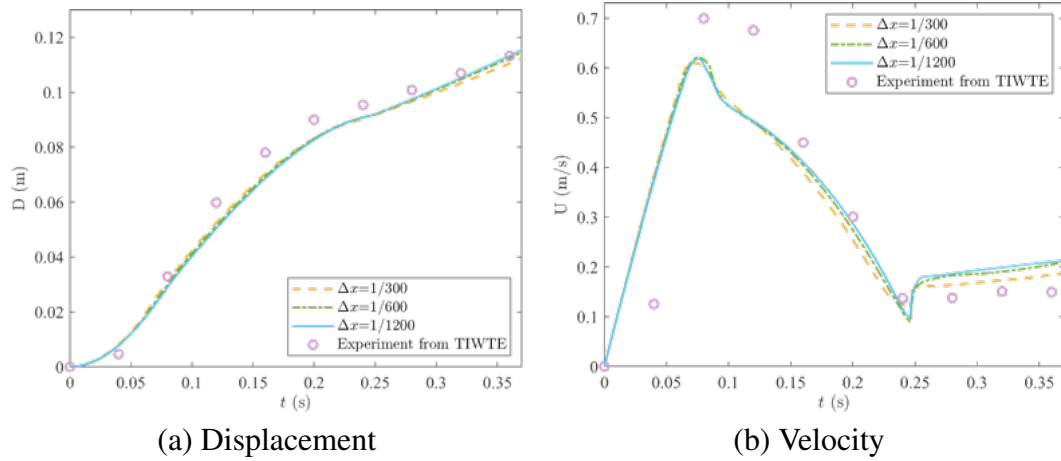


Figure 3.15: The displacement and velocity of the wedge with time t .

3.5 Validation case

3.5.1 Oil spill numerical model setup

3.5.1.1 Fluid Properties

The simulation of oil spill response at sea is a representation of the interactions among air, currents, waves, oil spill and oil booms, as outlined in Section 3.5. The numerical model configuration replicates the experimental setup described in TIWTE (26), and the validity of the model is verified by comparing the simulation results to the experimental data. The details of the experiments and equipment can be found in the literature (26).

The overall inlet is set to the left boundary, which is the generation side of wave and current. Given that the oil boom plays a role not only in gathering the oil but also in blocking wave propagation, there is no absorption area on the right boundary. The schematic diagram of the simulated initial state is shown in Fig.3.16(a). The entire tank domain is 8.0 m long and 1.3 m high, with 0.5 m thick air in the upper part and 0.8 m thick water in the lower part. The oil layer is formed by the industrial gear lubricant CKC680. It floats on the surface of the water, with a length of 1 m and a thickness of 0.02 m. The boom is placed close to the right side of the oil layer, positioned 5.0 m from the left side of the tank. See Tab. 3.3 for the physical properties of the air, water and oil.

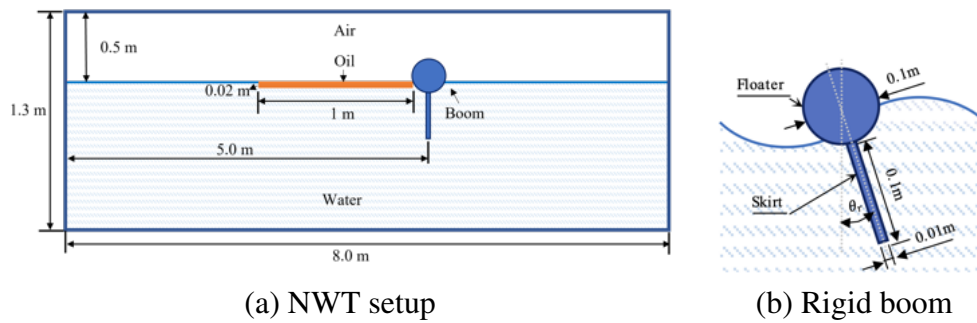


Figure 3.16: A schematic diagram of the dimensions of the oil spill response simulation.

3.5.1.2 Rigid Boom

The rigid boom model used in the simulation is a cylindrical floater with a diameter of 0.1 m and a skirt with a length of 0.1 m and a thickness of 0.01 m, as shown in Fig.3.16(b). The boom has a mass of 1.988 kg and a rotational inertia of $5.34 \times 10^{-3} \text{ kg} \cdot \text{m}^2$. The floater and skirt are treated as a single piece, and the boom's movement is represented by the whole structure's movement. The boom has 3 DOF — vertical, horizontal, and rotational motions, also described as heave, sway, and roll with rotation angle θ_r .

3.5.1.3 Mooring systems

Each boom is confined by an individual numerical spring force to simulate the elastic mooring system, as shown in Fig.3.17. The spring attached to the boom restricts the vertical and horizontal motions, while the rotational motion is constrained by the boom's inertia, which is set to be the same as that used in the experiment. The external spring force F_{ext} , represented by Eq.3.21, changes as the boom deviates from its initial position and depends on the magnitude of the deviation. The deviation is measured by the change in position of a randomly selected point, marked as the anchor in the simulation, from the boom's initial position. After t seconds, the position of the anchor is the offset distance of the boom from its initial position, δx and δy .

$$F_{ext} = \begin{bmatrix} k_1 & 0 \\ 0 & k_2 \end{bmatrix} \cdot \begin{bmatrix} \delta x \\ \delta y \end{bmatrix}, \quad \begin{cases} k_1 = 5000 \text{ kg/s}^2 \\ k_2 = 5000 \text{ kg/s}^2 \end{cases} \quad (3.21)$$

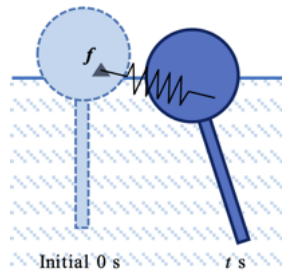


Figure 3.17: The schematic representation of the mooring system.

3.5.1.4 Fixed Cartesian grid

The size of the structural grid cell is 0.0042 m ($\frac{1}{240}$ m), with a total of 1926×318 cells (both including 6 ghost cells), see Fig.3.8. The workstation is the same as above.

3.5.2 Result and discussion

3.5.2.1 Effect of current

The numerical framework was validated by comparing it to the experimental results under current conditions only. Six simulations were performed with different current speeds

Table 3.5: Current conditions in the oil spill response simulations.

Simulation	1	2	3	4	5	6
Velocity (m/s)	0.06	0.15	0.27	0.4	0.48	0.66

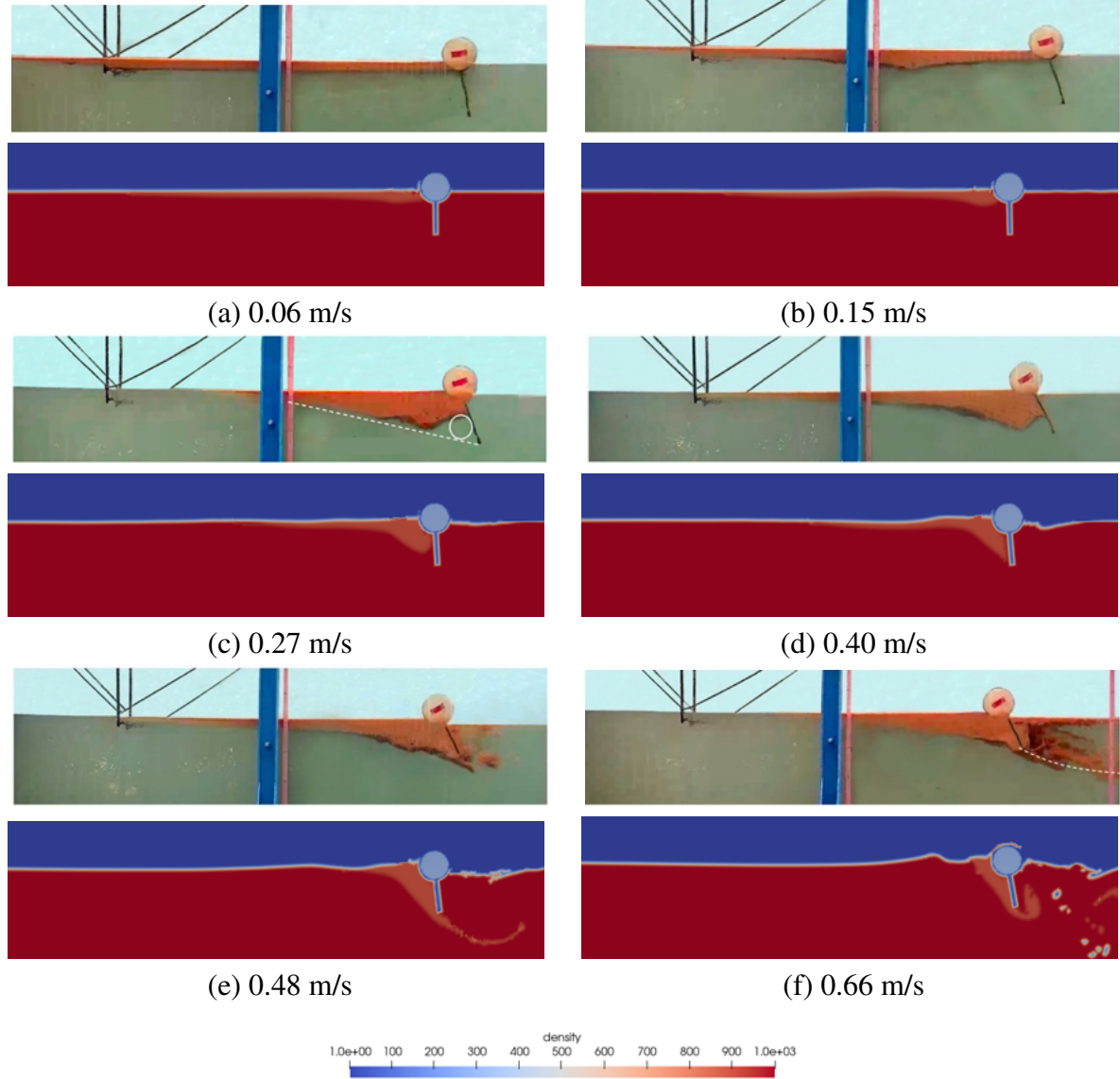


Figure 3.18: The snapshots of experimental (upper) and numerical (lower) results of rigid booms at different current speeds.

and the specific parameters used are listed in Tab. 3.5. The numerical simulations showed good agreement with the experimental results, as shown in Fig.3.18.

As the current velocity increased from 0.06 m/s to 0.66 m/s, the oil layer grew and accumulated towards the boom. The oscillation of the boom intensified, which helped to counteract the thrust of the thicker oil layer and prevent oil spread. No failure modes were observed in Fig.3.18 (a) to (d) for current speeds between 0 and 0.4 m/s. This indicates that the hybrid situation does not negatively impact the performance of the oil boom at

these speeds. However, when the velocity increased to 0.48 m/s (Fig.3.18(e)), small oil droplets started to escape the boom beneath the skirt and spread to the sheltered area behind it, indicating a drainage failure. At a velocity of 0.66 m/s (Fig.3.18(f)), the boom moved vigorously and the oil storage capacity was weak, causing a noticeable drainage failure mode. Unlike the calm water surface in the experimental setup, the water surface in the numerical wave tank becomes more disturbed as the flow velocity increases. This is because, before the experiment, the entire system had already reached a state of equilibrium, where the boom and water surface would not have any changes under no external driving force. In the numerical wave tank, the boom is initially placed near the equilibrium state, and when the simulation starts, it tries to maintain equilibrium in the presence of the current. Due to the restoring force of the spring, the boom begins to fluctuate up and down, causing ripples in the surrounding water surface. When the flow velocity is small, the influence of the current on the equilibrium state is minimal, and the boom quickly reaches a state of equilibrium, exhibiting a great and reasonable oil containment effect. However, at high flow velocities, the interaction becomes intense, and the boom is greatly affected, even leading to Splash Over at a flow velocity of 0.66 m/s.

3.5.2.2 Effect of waves

Table 3.6: Wave conditions in the oil spill response simulations.

Type	Water depth d (m)	Height H (m)	Period T (s)	Length L (m)
Airy	0.8	0.06	1.20	2.20

Table 3.7: Coordinates of the wave gauges while setting wave propagation as positive direction.

No. of WG	WG1	WG2	WG3
Relative position to floater initial centre (m)	-1.0	-0.5	+0.5

This section investigates the effects of waves in a hybrid environment on the response pattern of a floating boom model. To validate the present FSI method, a quantitative validation case including waves and currents ($U_c = 0.1$ m/s) is presented. The parameter settings of the inlet airy wave are listed in Tab. 3.6. Three resistance-type wave gauges were used in the experiment to observe the water surface elevation, and their positions are shown in Tab. 3.7. Simulated water surface changes with the same relative position as the wave gauges and boom in the experiment were recorded and compared with the wave gauge results, see in Fig.3.19. The comparison indicates a favourable agreement between the numerical simulation and experiment. Nevertheless, the amplitude of the numerical simulation slightly exceeds that of the experiment, which can be attributed to the three-dimensional effects of the experiment. Additionally, the numerical amplitude of 0.03 m is closer to the prescribed wave amplitude (i.e., a wave height of 0.06 m). These findings demonstrate the reliability and accuracy of the FSI method used in simulating oil spill response.

To account for reinitialisation at every time step, the LSF in Eq.3.6 is non-conservative. However, the preservation of mass balance is still achieved and demonstrated through the

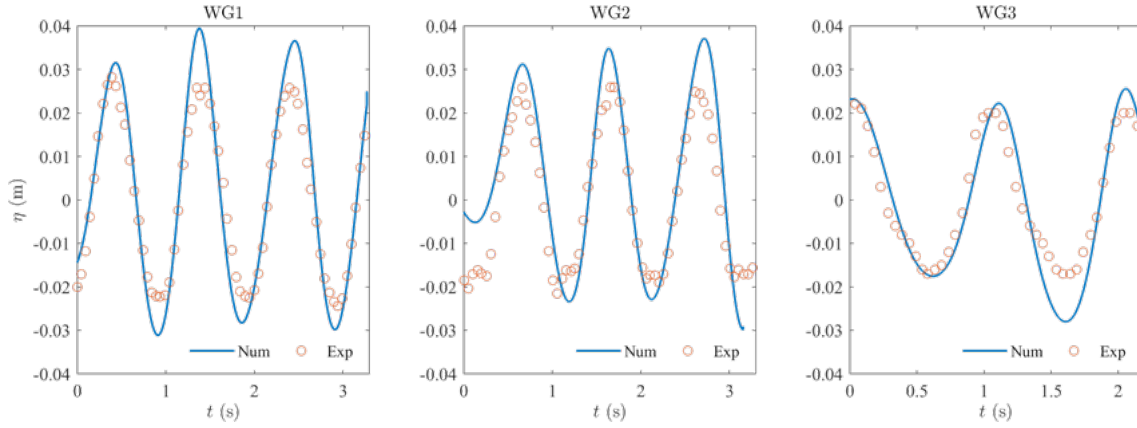


Figure 3.19: Comparisons between numerical and experimental water surface elevation ($U_c = 0.1$ m/s).

calculation and presentation of the mass error percentage over time, illustrated in Fig.3.20. The mass error percentage is calculated as:

$$\text{error} = \frac{\text{mass at the current time step} - \text{mass at the initial time step}}{\text{mass at the initial time step}}$$

Here, the mass at the initial time step refers to the mass of the oil phase at the start of the simulation. The mass error percentage remains within a very small range ($\pm 0.20\%$), indicating that the mass error is minimal and acceptable.

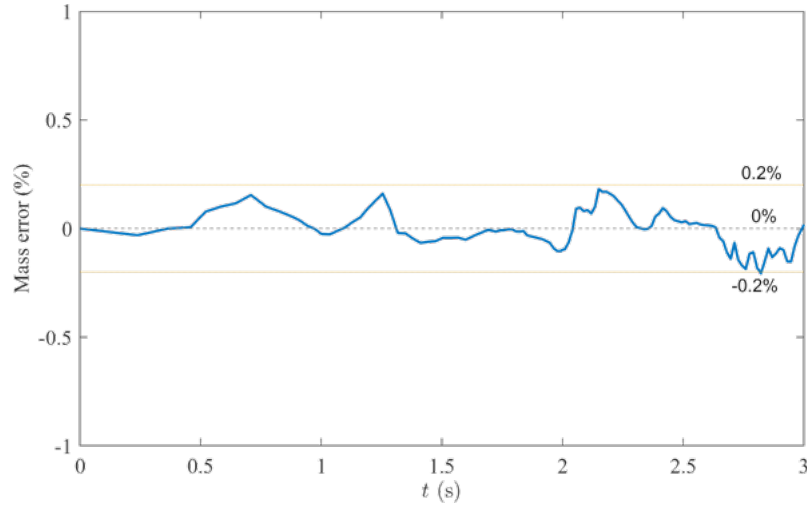


Figure 3.20: The oil mass percentage error over time (Current velocity $U_c = 0.1$ m/s and airy wave in Tab. 3.6).

Next, the boom's response pattern under the influence of waves is explored for the same wave as in the validation case (Tab. 3.6) with currents up to 0.2 m/s. One wave cycle is selected to investigate the boom position under wave crest and trough, as shown in Fig.3.21.

At the start of the cycle (t_1), the boom was floating upright on the water surface. Between t_1 and t_2 , the boom was subject to inertia and rotated counterclockwise in a small

range. At t_2 , there was a slight bulge on the offshore side of the boom due to the water blockage phenomenon. At this time, the upper part of the boom was impacted by the wave crest and the lower skirt was driven to rotate clockwise. By t_3 , the boom continued to rotate counterclockwise with an increased degree of inclination, despite the periodic change in wave direction. These results suggest that at a current speed of 0.2 m/s, the motion of the boom was dominated by the current rather than the wave or its inertia. The boom continued to rotate under the wave trough until the end of the observation period (t_5) with no significant deviations, indicating that the numerical spring force effectively restrained the boom under the combined wave and current forces.

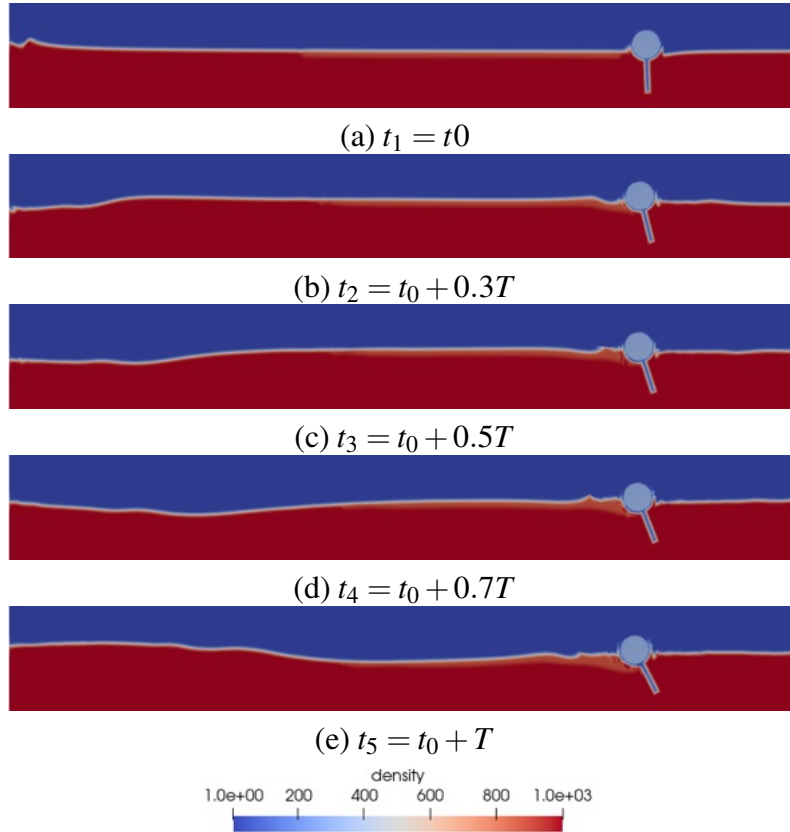


Figure 3.21: The poses of a rigid boom in one wave period with 0.2 m/s current.

The observed response pattern of the boom under the combined influence of waves and currents is driven by several physical mechanisms, including inertia, water blockage, and wave-induced rotation. Inertia plays a role in the initial rotation of the boom, as it tends to keep the boom stationary at the moment of wave strike, effectively preventing the initial rotation. Water blockage is another factor that affects the motion of the boom. As the water hits the boom, it creates a pressure differential, which increases the hydrodynamic force acting on the boom. This increased force, combined with the inertia of the boom, helps to alleviate the severity of the rotation between t_1 and t_2 . Wave-induced rotation is another mechanism that contributes to the overall motion of the boom. For the wave alone situation in (26), the periodic change in wave direction leads the boom to rotate counterclockwise and then reverse, showing a periodic oscillation. However, no reverse rotation is observed in Fig.3.21, suggesting that the motion of the boom was dominated

by the current rather than the wave or its inertia when the current was above 0.2 m/s. By understanding these mechanisms, it is possible to design booms with features that enhance their ability to collect oil effectively in the presence of waves and currents. For instance, increasing the boom's inertia by adding weight could reduce the initial rotation caused by wave strikes, thereby improving its stability. Additionally, designing a boom with a rotational mechanism that takes advantage of wave-induced rotation could help collect oil more effectively. These design considerations could lead to more effective and reliable oil spill response strategies, ultimately reducing the environmental impact of oil spills.

3.6 Conclusions

The numerical framework presented in this study provides a reliable solution to multi-phase flow problems, especially when simulating the interactions between flows and rigid structures. To maintain the conservation of mass and momentum, 'one-fluid' formulation was adopted as the governing equation. It is based on the incompressible Navier–Stokes equations and regards all the objects in the study domain as fluids. The different regions are represented by different Heaviside functions. The evolution of interfaces is captured by the 3-phase LSM to ensure the conservation of geometry. It is necessary to eliminate the vacuum and overlap and reinitialise the LSM.

Two experiments were conducted to validate the numerical model. The first experiment involved the free-fall motion of a wedge-shaped block into a two-layer water-oil fluid system. The left, bottom, and right sides of the tank were equipped with three absorption zones that simulated the effect of sponge layers. The results showed that the drop speed of the wedge rapidly increased during free fall, decreased slightly upon impact with the oil layer, and later moved at a relatively constant rate in the water layer. The second experiment was an oil spill response scenario in which a rigid boom was used to contain the spill. The boom was attached to a mooring system via a spring force. As the current speed increased from 0.06 to 0.66 m/s, the oil collected on the offshore side of the boom. Drainage failure was observed after the current speed reached 0.48 m/s. The final experiment involved the application of a hybrid wave-current boundary condition at the left inlet. The effect of the wave was studied via the performance of the rigid floating boom during a wave cycle. Further failures under this condition require further investigation.

The next step of this research will be to extend the numerical framework to three-dimensional simulations and further validate the accuracy of the results by comparing them with more experimental data. Additionally, the framework can be extended to study the impact of different parameters, such as wave height, current velocity, and oil viscosity, on the containment performance of the oil boom. These efforts will contribute to the development of effective strategies for oil spill response and environmental protection.

3.7 Acknowledgement

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Chapter 4

Numerical investigation of wave induced thrust on a submerged hydrofoil

(Under review: <https://doi.org/10.48550/arXiv.2209.05551>)

4.1 Abstract

Submerged flapping hydrofoils have the capability to directly convert wave energy into thrust, offering a sustainable approach to marine propulsion. This research employs Computational Fluid Dynamics (CFD) to analyse the propulsion mechanism of wave-induced flapping hydrofoils. Initially validated through established benchmarks and experimental results with foil in uniform flow, the CFD model was then applied to examine the generation of thrust by flapping hydrofoils in heading regular waves. The study reveals a distinct transition from drag to thrust, characterized by the patterns of vortex flow. For the first time, the influence of pitch stiffness on this propulsion process is extensively explored, identifying optimal wave conditions and pitch stiffness for the application of future eco-friendly marine systems.

Keywords: Wave Devouring Propulsion (WDP); Flapping foil; Thrust generation; Computational Fluid Dynamics (CFD); Fluid-Structure Interaction; Spring stiffness.

4.2 Introduction

Addressing the global challenge of climate change, the development of green ocean propulsion systems plays a pivotal role in curtailing CO₂ emissions. Ocean waves represent an underutilized energy source that flapping hydrofoils can convert into thrust, providing both propulsion and stabilisation for marine vessels. This study presents a 'one-fluid' Computational Fluid Dynamics (CFD) framework to understand the dynamics of wave-foil interactions and the resulting thrust, with the goal of improving the environmental and economic efficiency of marine field.

Flapping hydrofoils can generate thrust from flows, a concept first explained by Knoller (1) and Betz (2). Wu et al. (3) explored energy extraction from wave-foil interactions, while Isshiki (4) introduced wave devouring propulsion (WDP) using passive flapping hydrofoils. Lighthill (5) and subsequent researchers like Koochesfahani et al. (6) confirmed the formation of a jet-like wake, facilitating thrust. Studies have since employed numerical simulations to understand WDP's physics for marine applications, analysing active flapping hydrofoils under various conditions. Filippas et al. (7, 8) and Gao et al. (9) investigated the impact of hydrofoil movement and perturbations on thrust generation. Qi et al. (10) compared sinusoidal and non-sinusoidal pitching effects on propulsion, finding that certain pitching profiles enhance propulsive performance. De Silva and Yamaguchi (11) studied active oscillation hydrofoils in waves, noting efficient wave energy conversion into propulsion under specific conditions. Wang et al. studied the use of WDP for stabilizing a floating wind platform (12) and for reducing fuel consumption in shipping (13).

Hydrofoils can transform flow energy into thrust using their flapping motion, often controlled with springs to adjust heave/pitch movements. This enables various flapping modes, where an infinite stiffness results in pure heave or pitch motion, while reducing the stiffness leads to a coupled flapping motion. Prior research has mainly focused on active flapping, which our study uses as a benchmark to validate our CFD code. Yet, the passive thrust generation by wave-induced flapping remains less explored.

This research presents a novel 'one-fluid' CFD framework to analyse thrust production from passive, wave-induced flapping motions, with a focus on how changes in pitch spring stiffness influence thrust—a topic previously unexplored with CFD simulations. Additionally, the developed methodology can be readily applied to study the impact of heave spring stiffness on thrust generation.

The paper is structured as follows: Section 4.3 details the 'one-fluid' framework, hydrofoil geometry, motion modes, flow simulation, and key parameters. Model convergence and validation tests in uniform flow are covered in Section 4.4. Section 4.5 explores the effects of wave and pitch stiffness on thrust in passive hydrofoils under heading waves. Conclusions are summarised in Section 4.6.

4.3 Material and Method

4.3.1 Numerical solver

The numerical solver used in this study is an 'in-house' developed code that employs the 'one-fluid' formulation, recently proposed by Yang et al. (14, 15, 16, 17). The simulation platform in this study employs the same as illustrated in Chapter 3.3.

4.3.1.1 ‘One-fluid’ formulation - Governing equations

Vortices are key to thrust generation from the hydrofoil (18). Our aim is to solve the incompressible Navier-Stokes equations to capture these phenomena.

$$\rho \left[\frac{\partial u}{\partial t} + (\nabla u)u \right] = -\nabla p + \nabla \cdot 2\mu \mathbf{D}(u) + \rho g \quad \text{in } \Omega \quad (4.1a)$$

$$\nabla \cdot u = 0 \quad \text{in } \Omega \quad (4.1b)$$

$$\mathbf{D}[u] = 0 \quad \text{in } \Omega_r \quad (4.1c)$$

where $\rho = \rho_a H_a + \rho_w H_w + \rho_r H_r$ is the density field, u is the velocity vector field, g is the gravitational acceleration, μ is the dynamic viscosity and $\mathbf{D}(u) = \frac{1}{2} (\nabla u + (\nabla u)^T)$ is the strain rate tensor. The subscripts indicate the different phases, i.e., a for air, w for water, and r for rigid body.

The LSM is used for capturing the interface between the fluid phases in the numerical domain, while the characteristic function χ of any region Ω_α can be expressed as

$$\chi = H_\alpha(\phi) \quad \alpha = a, w, r \quad (4.2)$$

where

$$H_\alpha(x) = \begin{cases} 1 & \text{if } x \in \Omega_\alpha \\ 0 & \text{if } x \notin \Omega_\alpha \end{cases} \quad (4.3)$$

is a Heaviside function for a phase α . $x = (x, y)$ is the coordinate of any point in the domain. H_α is introduced to identify different phase regions. Once the Heaviside function has been constructed, the corresponding density and viscosity field can be assigned. For more information refer to Yang’s study (15).

4.3.1.2 Numerical and discretisation scheme

All simulations use a fixed Eulerian Cartesian grid and a grid-based immersed method for numerical discretization. The Cartesian staggered Finite Volume scheme (Marker-and-Cell) is adopted for spatial discretization. The convective terms of the Navier-Stokes equations are discretized with a second-order Total Variation Diminishing (TVD) Runge-Kutta scheme in a conservative framework. Time integration of the computational momentum and level set equations is performed using a second-order accurate Runge-Kutta scheme. Adaptive time steps are used to maintain the Courant-Friedrichs-Lewy (CFL) number at 0.5 for all simulations (15).

4.3.1.3 Boundary conditions

This study impose Dirichlet boundary condition for the wave generation and absorption. The initial flow profiles are derived from experimental inlet settings. Specifically, the uniform flow and regular heading waves are referenced from the studies by Read et al. (19) and Isshiki et al. (20, 21), respectively.

Convergence tests and initial validation of the computations, see in Section 4.4, are performed with one hydrofoil subject to the uniform flow. The flow velocity U_{flow} is the

same throughout the domain and is described by Eq.4.4 and Fig.4.5.

$$u(x, y, t) = U_{flow} \quad (4.4a)$$

$$v(x, y, t) = 0 \quad (4.4b)$$

The current flows steadily along the x positive direction with zero velocity in the y direction. The flow enters the domain from the left boundary and exits through the right boundary. To address concerns regarding the trailing wake, it is important to note that both the left and right boundaries are positioned sufficiently distant. It ensures uniform flow at both boundaries, maintaining the conservation of mass while minimising the influence of wake dynamics on the boundary conditions.

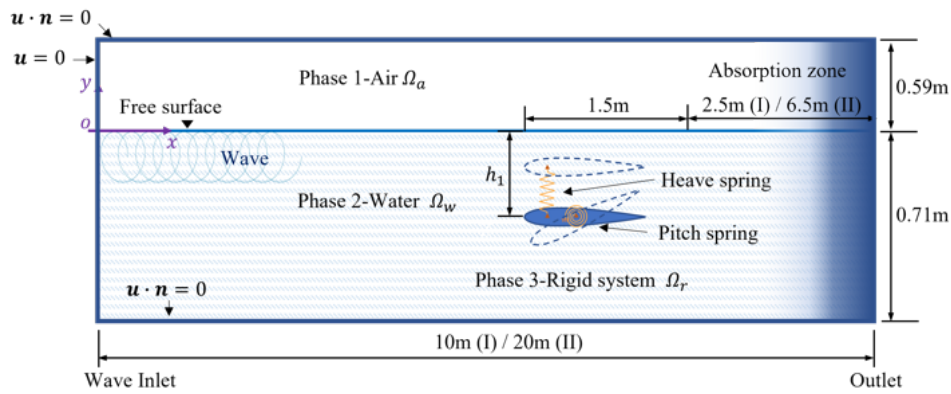


Figure 4.1: The schematic diagram of the passive hydrofoil simulation.

For the study of wave-induced performance, see in Section 4.5, the hydrofoil is exposed to regular waves. The schematic diagram for passive foil simulations is shown in Fig.4.1. For these simulations, wave generation and absorption are set on the left and right-hand side of the numerical domain, see also Fig.4.1. The flow conditions, in the form of horizontal and vertical velocity, at the inlet are calculated using linear wave theory as:

$$u(x, y, t) = \frac{\pi H}{T} \frac{\cosh[k(y + d)]}{\sinh(kd)} \cos(kx - \omega_w t) \quad (4.5a)$$

$$v(x, y, t) = \frac{\pi H}{T} \frac{\sinh[k(y + d)]}{\sinh(kd)} \sin(kx - \omega_w t) \quad (4.5b)$$

where u and v are the instantaneous horizontal and vertical particle velocities at (x, y) location, H is the wave height, T is the wave period, d is the water depth, $k = \frac{2\pi}{\lambda}$ is the wave number, $\omega_w = \frac{2\pi}{T}$ is the angular wave frequency, and λ the wavelength calculated with the linear dispersion equation, Eq.4.6. Using linear wave theory, the wave conditions in the numerical domain were tuned to match the wave measured in the experiments of Isshiki et al. (21). It is noted, that the use of linear wave theory for generating non-linear waves is a limitation of this study, which can lead to the generation of spurious long wave components. The application of a more precise wave generation procedure goes beyond the scope of the present work and the spurious effects are partly mitigated through the use

of an absorption zone that the end of the numerical fume.

$$\lambda = \frac{gT^2}{2\pi} \tanh(kd) \quad (4.6)$$

The relaxation zone is applied to gradually decrease the velocities u and v in the absorption zone to zero. This technique is known to mitigate spurious wave reflections and improve the quality of the simulation results. The length of the wave absorption zone at the outlet (right hand side boundary) varies in accordance with the incoming wavelength and it is set to 2.5 m for $\lambda < 3.75$ m and 6.5 m for $\lambda < 7.70$ m. At the absorption zone wave reflection is prevented through the application of the relaxation method proposed by Bihs et al. (22), Eq.4.7.

$$\Gamma(\tilde{x}) = 1 - \frac{e^{(\tilde{x}^{3.5})} - 1}{e - 1} \quad \text{for } \tilde{x} \in [0, 1] \quad (4.7a)$$

$$u_r = \Gamma(\tilde{x})u_c \quad (4.7b)$$

$$v_r = \Gamma(\tilde{x})v_c \quad (4.7c)$$

Here, $\Gamma(\tilde{x})$ represents the relaxation function, with $\tilde{x}$ being a scale factor ranging from 0 to 1, denoting the normalized distance from the wave inlet boundary to the absorption zone. In the absorption zone, u_r and v_r are the relaxed velocity components, contrasting with u_c and v_c , the original computational velocities.

4.3.2 Hydrofoil characteristics

4.3.2.1 Geometry and motions

The study validates simulation results against experimental findings by Read et al. (19) and Isshiki et al. (20, 21), focusing on two NACA hydrofoil types: NACA0012 and NACA0015. Each hydrofoil is described by its chord length c and maximum thickness D , with D being a percentage of c . While 3D applications consider the hydrofoil's span s , this 2D analysis normalizes s to 1.

Hydrofoil motions in simulations are either active oscillations (enforced around a fixed position) or passive flapping (flow-induced combining heave and pitch), as depicted in Fig.4.2. Active motions include vertical heave or rotational pitch around a pivot point P . In accordance with the experiments of Read et al. (19), P is set at the chord and a distance of $c/3$ from the leading edge of the hydrofoil. Pure heave and pitch motions are dictated in the numerical domain through Eq.4.8a and 4.8b. The flapping hydrofoil motion emerges from the combination of heave and pitch.

$$h(t) = h_0 \sin(\omega t) \quad (4.8a)$$

$$\theta(t) = \theta_0 \sin(\omega t + \phi) \quad (4.8b)$$

where $h(t)$ is the displacement of the hydrofoil in the y direction or in other words the instantaneous vertical position of the hydrofoil axis. $\theta(t)$ is the instantaneous angle formed between the incoming flow velocity (U_∞) and the chord, also referred to as the angle of rotation. h_0 and θ_0 is the heave and pitch amplitude. ω is the angular frequency (rad/sec),

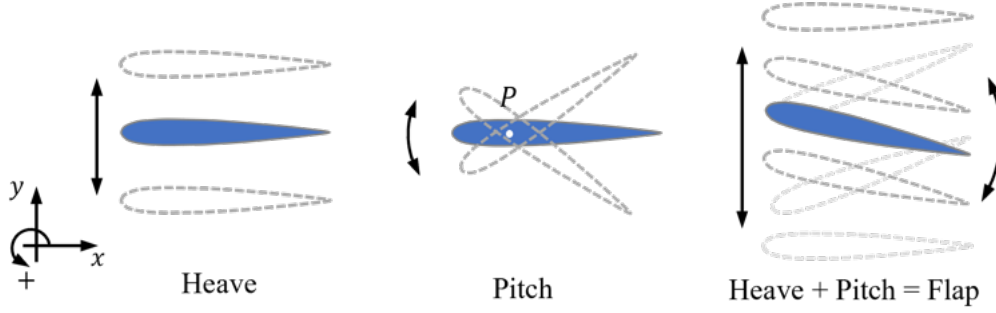


Figure 4.2: The motions of the hydrofoil.

$2\pi f$, where f is the oscillating frequency, and ϕ is the phase difference between the pitch and heave motion. When it comes to propulsion efficiency, Platzer et al. (23) suggested $\phi = 90^\circ$ as the optimal phase difference between pitch and heave.

A hydrofoil can generate either a drag force or a thrust force, depending on the wake patterns. When a von Kármán vortex street (vK) (24) develops, the velocity of the flow within the wake is diminished relative to the incoming or ambient flow velocity. This leads to the hydrofoil entering a regime dominated by drag. However, when the vortices in the wake align at the hydrofoil's trailing edge, a unique condition arises. The velocity of the flow within the wake now matches the incoming or ambient flow velocity, and the hydrofoil experiences a regime of neutrality. Further evolution of these conditions can lead to thrust generation. When a reversed von Kármán vortex street (RvK) develops, it accelerates the flow behind the hydrofoil, leading to thrust production. The emergence of a reversed Rvk wake signals the onset of the drag-to-thrust wake transition. However, there's a delay in the hydrofoil's transition towards thrust generation. This delay occurs because the minor velocity surplus cannot overcome the profile drag, velocity fluctuations, or pressure differences (25, 26, 27, 28). The three fundamental wake patterns are illustrated in Fig.4.3.

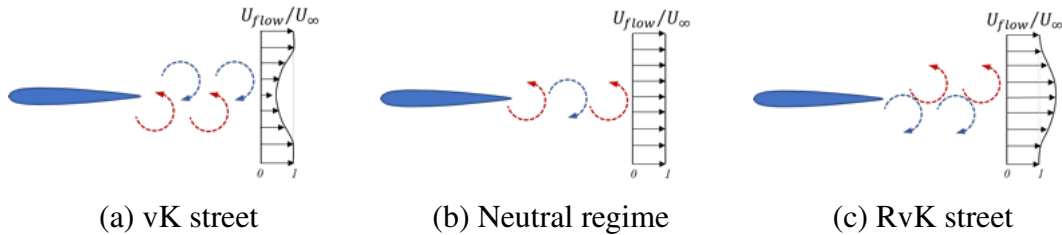


Figure 4.3: The illustration for the drag-to-thrust wake transition of the flapping hydrofoil.

For passive hydrofoils controlled by springs, the switch between drag and thrust depends on spring stiffness, affecting pitch angle change and amplitude. Simulations represent this effect by applying an instantaneous force and moment on the hydrofoil, as outlined by Eq.4.9 and depicted in Fig.4.1.

$$F_h(t) = k_h \Delta y \quad (4.9a)$$

$$M_p(t) = k_p \Delta \theta \quad (4.9b)$$

where, $F_h(t)$ is the heave force and $M_p(t)$ is the pitch moment. Δy and $\Delta \theta$ are the vertical and rotational deviation of the hydrofoil from its initial position at time t , respectively,

while k_h and k_p are the heave and pitch stiffness. This definition enables the direct comparison with the experiments of Isshiki et al. (20, 21) for which k_h is set to 10388 N/m and k_p to 129.24 N·m/rad. The pitch stiffness effect is then explored by keeping k_h constant and varying k_p for different incoming flow conditions. This topic will be revisited in Section 4.5.2. To remove the effect of the buoyancy force, the hydrofoil's density has been matched to that of the fluid $\rho = \rho_{hydrofoil} = \rho_{fluid}$. We have included the gravity force for the wave dynamics.

4.3.2.2 Main and dimensionless parameters

In fluid dynamics, hydrofoils are subject to both drag and lift forces, illustrated in Fig.4.4. As elaborated earlier, the formation of an RvK vortex street indicates a shift from drag to thrust, occurring when the wake's velocity surpasses the surrounding flow's speed, thus generating thrust. In this study, the thrust force opposes drag and is denoted as F_x , while the lift force, acting vertically upward, is labeled F_y . These forces are quantified using dimensionless coefficients for lift (C_l) and drag (C_d), calculated by normalizing the force on the hydrofoil against the dynamic pressure of the incoming flow and the hydrofoil's platform area, as shown in Equation 4.10.

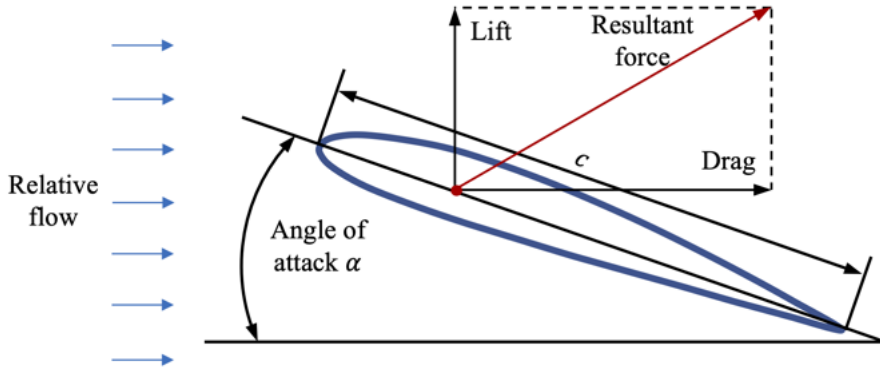


Figure 4.4: The forces on the hydrofoil facing flows.

$$C_l = \frac{F_L}{\frac{1}{2}\rho U_\infty^2 c} \quad (4.10a)$$

$$C_d = \frac{F_D}{\frac{1}{2}\rho U_\infty^2 c} \quad (4.10b)$$

where, F_L and F_D is the lift and drag force and c is the hydrofoil chord. $F_T = -F_D$ and $C_t = -C_d$ are the thrust force and coefficient. In addition, Eq.4.10 refers to instantaneous coefficient values and the time-averaged, e.g. over one wave period T , forces and coefficients ($\overline{F_D}$, $\overline{F_L}$ and $\overline{C}$) are calculated as:

$$\overline{C} = \frac{1}{T} \int_0^T C(t) dt \quad (4.11)$$

$$\overline{F_x} = \frac{1}{T} \int_0^T D(t) dt \quad (4.12)$$

$$\overline{F_y} = \frac{1}{T} \int_0^T L(t) dt \quad (4.13)$$

where, $\overline{C}$ can be any of $\overline{C_l}$, $\overline{C_d}$ and $\overline{C_t}$.

The interaction between a moving hydrofoil and the incoming flow is described through the non-dimensional Strouhal and Reynolds numbers. The chord based Re is calculated from Eq.4.14a, while several expressions for the Strouhal number have been proposed. Typically, the peak-to-peak oscillating amplitude, A , of the hydrofoil's trailing edge and its oscillating frequency, f , are normalised with the characteristic flow velocity, U_∞ , for estimating St_A , Eq.4.14b (29, 18). Read et al. (19) reported optimum thrust generation for $0.2 < St_A < 0.4$.

$$Re = \frac{\rho c U_\infty}{\mu} \quad (4.14a)$$

$$St_A = \frac{fA}{U_\infty} \quad (4.14b)$$

where ρ is the density of the surrounding fluid, U_∞ is the characteristic flow speed, μ is the dynamic viscosity of the fluid, f is the oscillating frequency and A is the oscillating amplitude, which is often taken as the width of the wake. For uniform flow, U_∞ is defined as the horizontal velocity component in the x-direction relative to the foil. In the case with waves, U_∞ is taken as the x-direction horizontal velocity component of the wave. Although this may not be entirely accurate as it does not account for the instantaneous forward speed of the foil, it is the most appropriate approximation given that the foil's instantaneous speed is constantly changing and cannot be predicted beforehand due to its passive motion. For forced harmonic oscillating motions, $2h_0$ is an estimate for the width of the hydrofoil wake, thus A can be approximated with $2h_0$, and since $f = \frac{\omega}{2\pi}$ Eq.4.14b is also written as Eq.4.15.

$$St_A = \frac{\omega h_0}{\pi U_\infty} \quad (4.15)$$

4.4 Hydrofoil in uniform flow

4.4.1 Fixed hydrofoil

This section assesses the convergence of drag and lift coefficients for a NACA0012 hydrofoil in uniform flow, depicted in Fig.4.5. The simulation uses a numerical tank sized 7 m by 7 m, positioning the hydrofoil vertical centrally, one meter from the inlet on the left boundary. Fig.4.5 shows the coordinate system starting at the hydrofoil's leading edge, extending positively towards the trailing edge and upward from the front. The hydrofoil's density matches the fluid's at $\rho=1 \text{ kg/m}^3$, with a dynamic viscosity of $\mu=10^{-3} \text{ kg/m} \cdot \text{sec}$. The incoming flow velocity is set at $U_\infty=1 \text{ m/s}$, with an angle of attack (AoA) α of -15° and the chord length c is 1 m. Consequently, the Reynolds number is calculated as $Re = 1000$, fitting within the typical range for flapping hydrofoils of $100 < Re < 10000$ (30). Direct Numerical Simulation is utilised for these low Reynolds number flows in the fixed hydrofoil scenario.

Convergence tests were conducted using three different mesh sizes, as detailed in Tab. 4.1. The results for both lift coefficient (C_l) and drag coefficient (C_d) are presented

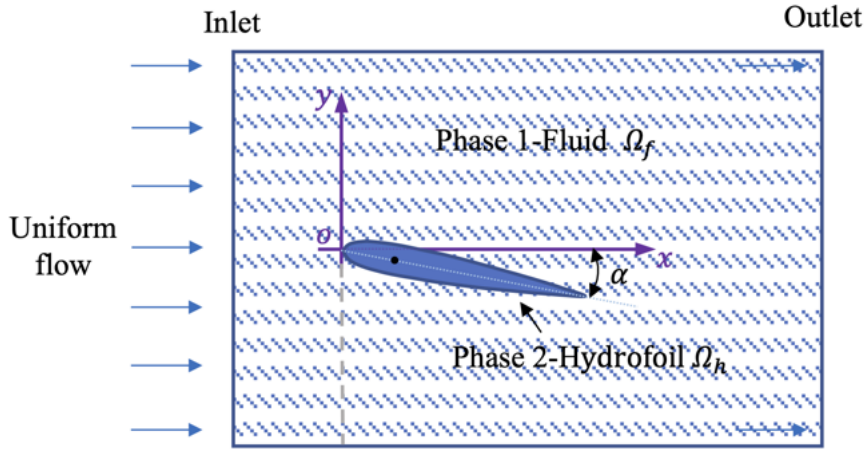


Figure 4.5: A schematic diagram of the simulation for the fixed hydrofoil in uniform flow.

Table 4.1: Grid resolution for numerical comparison of fixed hydrofoil model.

	Mesh size Δx (m)	Numbers of cells
1	0.0156	448×448
2	0.0078	896×896
3	0.0039	1792×1792

in Fig.4.6. The convergence analysis was conducted following the ITTC recommended procedures. By examining the results across three different grid resolutions, the convergence ratio R was calculated between successive resolutions. The R values consistently ranged between 0 and 1, confirming that the simulation results exhibit monotonic convergence. The period between 20 and 30 seconds was specifically chosen for analysis because, during this interval, the wake reaches full development. In the fixed case, the numerical scheme becomes a direct forcing method, and achieves expected results. The consistently positive C_d indicates a constant, though periodically fluctuating, drag force acting on the hydrofoil, with no thrust generation observed.

4.4.2 Active flapping hydrofoil

This subsection examines the active flapping hydrofoils in uniform flows, highlighting the transition from drag to thrust as demonstrated in previous studies (19, 31, 32, 33). The experimental findings of Read et al. (19) serve as benchmarks for this analysis. They conducted experiments with a towed NACA0012 hydrofoil undergoing harmonic (sinusoidal) oscillation in heave and pitch, measuring lift, thrust forces, and pitching torque across various angles of attack and ϕ .

The towing speed was 0.4 m/s, and the hydrofoil chord length was 0.1 m, resulting in a Reynolds number of $Re=4 \times 10^4$. This setup is replicated in a 2D numerical tank sized $0.5 \text{ m} \times 0.6 \text{ m}$, similar to the configuration shown in Figure 4.5(a). The numerical simulation adopts a uniform flow at 0.4 m/s, with the hydrofoil's chord length also at 0.1 m and matching the fluid density. Heave and pitch displacements are induced using Eq.4.8 with $h_0=0.075 \text{ m}$ and $\phi=-90^\circ$. This configuration produced a trajectory that allows

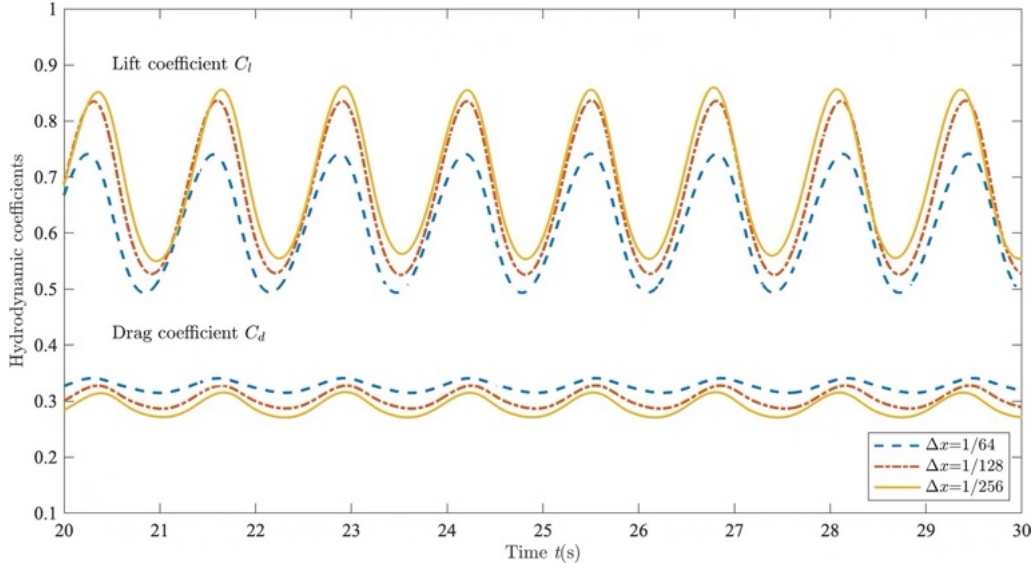


Figure 4.6: Validation of convergence of fixed hydrofoil facing uniform flow.

for a symmetric evolution of $\alpha_{eff}(t)$, closely following the experimental design. The pivot point is set at a third of the chord length from the leading edge, with an initial angle of attack of -15° . The simulations and experiments span Strouhal numbers St_A from 0.08 to 0.44 and frequency f ranges from 0.21 to 1.17 Hz, utilising RANS with zero eddy viscosity to resolve the fluid dynamics.

Numerical results, compared with experimental data in Fig.4.7, show the thrust coefficients C_t plotted against the amplitude-based Strouhal number St_A . The findings indicate that C_t rises with St_A . However, across all St_A values, the calculated C_t consistently falls below the experimental C_t , except at $St_A=0.44$. One of the primary reasons for this discrepancy is the inherent limitations of 2D simulations. As discussed in Mittal and Balachandar's study (34), 2D simulations often fail to account for three-dimensional effects such as spanwise flow variations and tip vortices, which significantly influence thrust generation. The 2D model may not accurately represent the growth and separation of the boundary layer along the span of the foil, resulting in differences in the thrust coefficient. Additionally, the experimental setup likely includes end effects from the finite span of the foil, which are absent in 2D simulations and can alter the flow pattern and pressure distribution around the foil, contributing to differences in the thrust measurement.

The Strouhal number St_A , as defined, reflects the ratio of the hydrofoil's trailing edge speed to the flow velocity. Increasing the hydrofoil's amplitude enhances motion-induced thrust, as shown by higher C_t values for larger St_A , as shown in Fig.4.7. This relationship is further illustrated by analyzing the wake and thrust development for varying St_A values, with visual examples in Fig.4.8. The figures capture the fluctuating forces on the hydrofoil across a motion cycle, oscillating between thrust and drag. Significantly, the forces on the hydrofoil oscillate between thrust and drag within a motion cycle. At specific times (t_0 , $t_0 + T$, and $t_0 + 1/2T$) (Fig.4.8 columns (a), (c) and (e)), when the hydrofoil returns to its initial position, implying a zero heave velocity and maximum AoA, thrust is produced, except at $St_A = 0.08$ time $t_0 + T$. Conversely, at times (t_0 , $t_0 + 1/4T$, and $t_0 + 3/4T$) (Fig.4.8 columns (b) and (d)), the heave acceleration is zero, and the AoA in the pitch direction reaches its minimum ($=0^\circ$), consistently generating drag.

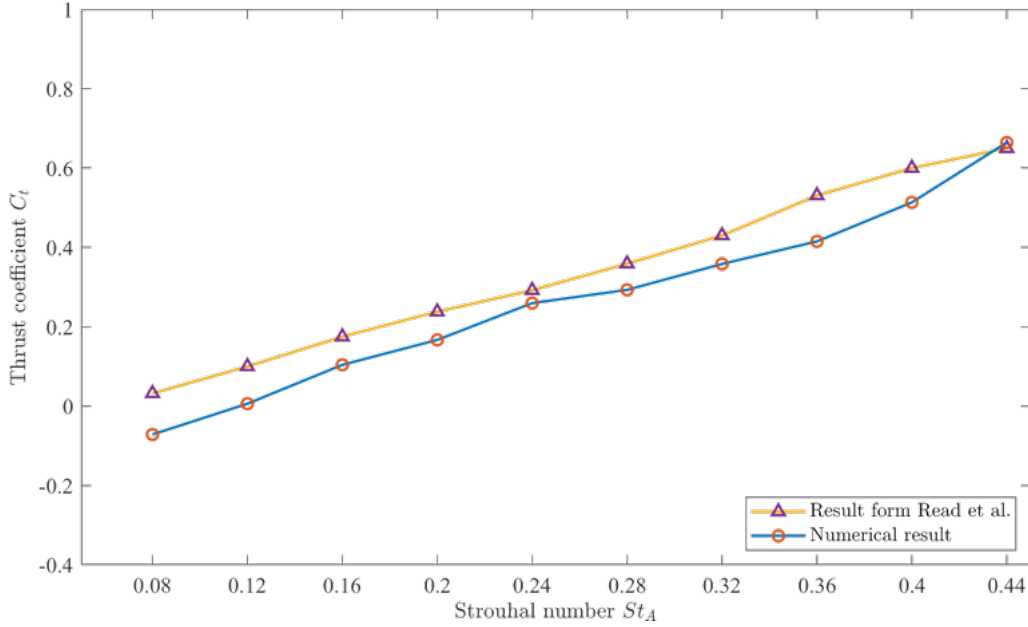


Figure 4.7: The comparison of the thrust coefficient C_t under different St_A between the experiment (the upper line) and active hydrofoil simulation ($\Delta x = \frac{1}{1280}$ m).

4.5 Hydrofoil in waves: Passive flapping hydrofoil

4.5.1 The effect of heading wave

This section explores the interaction between waves and hydrofoils under regular wave conditions with varying spring stiffness, comparing the results with Isshiki et al.'s physical model tests (20, 21). The setup for numerical simulations is illustrated in Fig.4.1, highlighting that computational domain sizes adjust according to the wavelengths discussed in Tab. 4.2. Shorter wavelengths in group I use a 10 m by 1.3 m domain, while longer waves in group II require a 20 m by 1.3 m domain, optimizing calculation efficiency. The simulation space includes air, water, and hydrofoil phases. A NACA0015 hydrofoil, with a chord length of 0.4 m, is positioned 0.077 m below the air-water interface, mirroring the experimental setup. The simulations employ restoring forces and moments to simulate the experimental springs (Eq.4.9), controlling heave and pitch movements with stiffness values of $k_h=10388$ N/m and $k_p=129.24$ N·m/rad, matching the physical tests.

In Fig.4.9, the time-averaged advance speeds of a passive hydrofoil are plotted across a range of wavelengths, comparing numerical simulation results with experimental and calculated data from Isshiki et al. (20, 21). The advancing speed, U , is defined as the average speed of the foil after it has reached a steady motion following several wave cycles. This speed is determined by measuring the distance the foil travels in the x-direction over a specific period and then dividing that distance by the elapsed time. The graph demonstrates a consistent trend across all three data sets, with the numerical results slightly lower than the experimental findings but closely following the calculated results. This close alignment suggests that the numerical simulations are capturing the key physical phenomena with a high degree of accuracy, despite the inherent simplifications of 2D modeling. The convergence between the numerical and experimental data, especially at

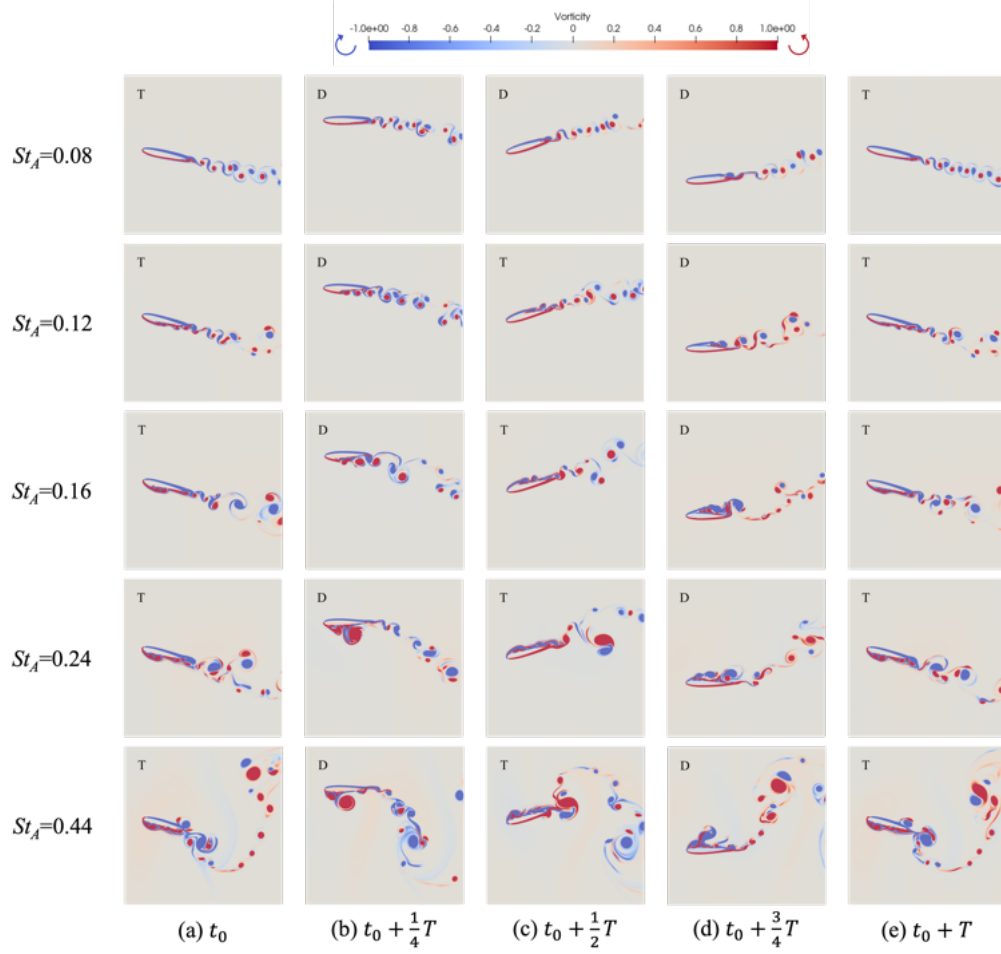


Figure 4.8: Snapshot series of the motion of active hydrofoil and its wake in one cycle ($St_A=0.08$ (1st row), 0.12(2nd row), 0.16(3rd row), 0.24(4th row), 0.44(5th row), $\Delta x = \frac{1}{1280}$ m. The force status is marked in the top left corner, where D is for drag and T is for thrust.).

Table 4.2: Wave conditions in the semi-active flapping hydrofoil simulations.

Group	No.	Wavelength λ (m)	Wave height $2a$ (m)	Wave steepness
I	1	2.00	0.096	0.048
	2	2.37	0.108	0.045
	3	2.92	0.123	0.042
	4	3.75	0.143	0.038
II	5	4.75	0.159	0.034
	6	5.92	0.170	0.029
	7	6.88	0.174	0.025
	8	7.70	0.175	0.023

longer wavelengths, indicates that the simulations are particularly effective at capturing the hydrofoil's interactions with such wave conditions. The persistent lower thrust predictions in the simulations are likely due to the two-dimensional approach's limitations in fully representing the intricate three-dimensional flows and interactions observed in

actual environments.

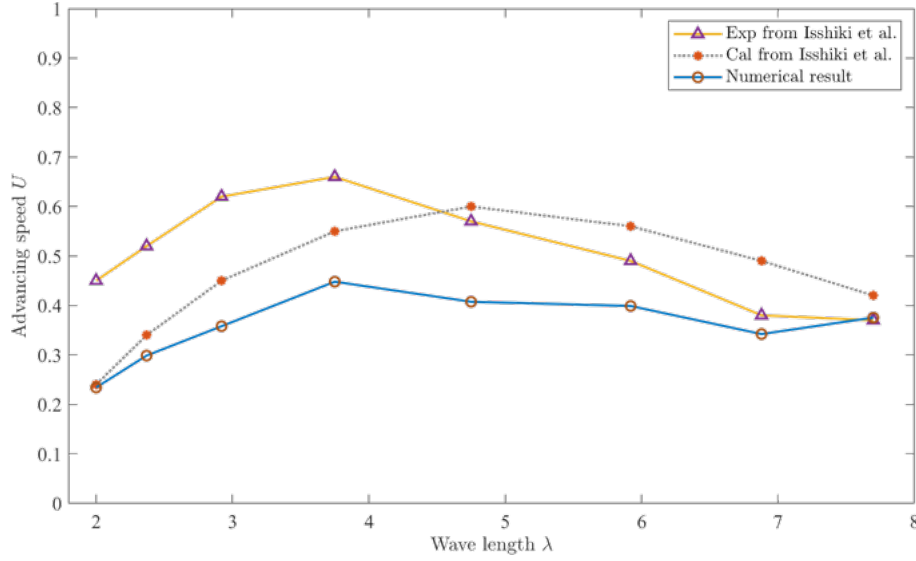


Figure 4.9: The comparison between the experiment, calculation and simulation for passive hydrofoil ($\Delta x = \frac{1}{160}$ m.)

Fig.4.10 captures the hydrofoil's interaction with regular waves, specifically with a wavelength of 3.75 m where it achieves its highest advance speed within the studied range. The snapshots cover the moment the hydrofoil encounters a head-on wave and its subsequent motion over six wave periods. Fig.4.11 illustrates the thrust variation corresponding to Fig.4.10, with the horizontal axis intuitively representing the states from t_0 to t_5 as snapshots. The red dots in the figure indicate the thrust at each moment. Here, the thrust in the direction of the foil's motion is considered positive. Comparing the snapshots, it can be observed that during the progression from t_0 to t_5 , the force is predominantly positive but fluctuates, demonstrating the transition and generation of thrust and drag. At t_4 , a neutral vortex street forms, leading to mild propulsion. After t_5 , an RvK vortex street develops, visibly boosting the hydrofoil's self-propelled motion against the incoming waves. Overall, thrust dominates over drag, resulting in noticeable forward motion.

Fig.4.12 shows the hydrofoil's motion and wake over one wave period, where the chord line orientation and vortex street evolution are presented. As the hydrofoil tilts under the wave crest (Fig.4.12(a)), a negative AoA generates an RvK vortex street, propelling it against the wave. This force fluctuates with the wave's motion, transitioning between thrust and drag, ultimately resulting in net positive thrust. By the end of one cycle, the pattern repeats with the next wave crest (Fig.4.12(e)), indicating a consistent thrust-generation mechanism, as confirmed by studies like those of Isshiki et al.(20, 21).

Fig.4.13 illustrates how the hydrofoil's non-dimensional displacement varies with wavelength over time, with the advance velocities indicated next to each line and also shown in Fig.4.9. The displacement initially rises mildly due to the hydrofoil's interaction with the first waves, then peaks and falls sharply within each period, reflecting the dynamic motion between wave crests and troughs. This cyclic motion results in net positive displacement.

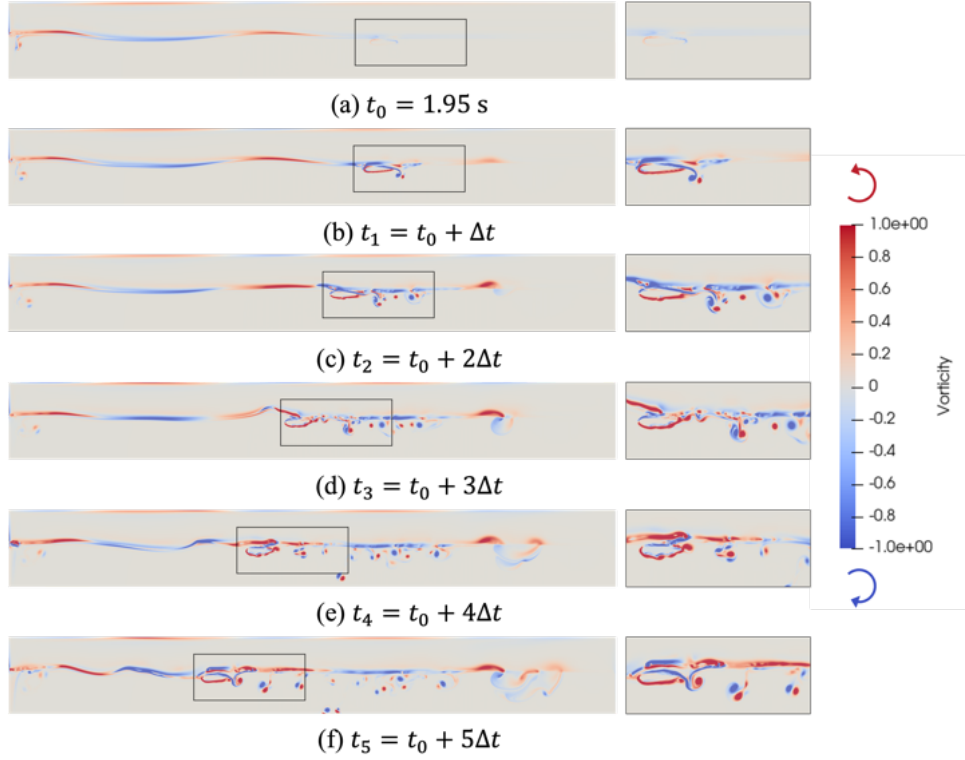


Figure 4.10: The position of the passive hydrofoil in wave (wavelength=3.75 m, time interval $\Delta t = 1.70$ sec is the wave period). The series of snapshots on the right are the zoomed-in parts.

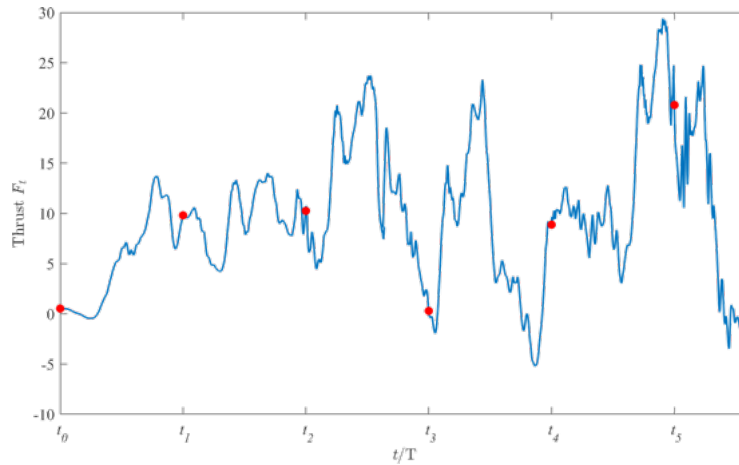


Figure 4.11: Thrust variation of the passive hydrofoil in wave corresponding to foil snapshots.

4.5.2 The effect of the pitch stiffness

This section examines how pitch stiffness influences thrust generation, affecting the hydrofoil's pitch amplitude and its propulsion performance. Originally, the pitch spring stiffness (k_p) was 129.24 N·m/rad in Section 4.5. To deepen the investigation, simulations with k_p values between 0 and 12900 N·m/rad were conducted. When the foil encounters

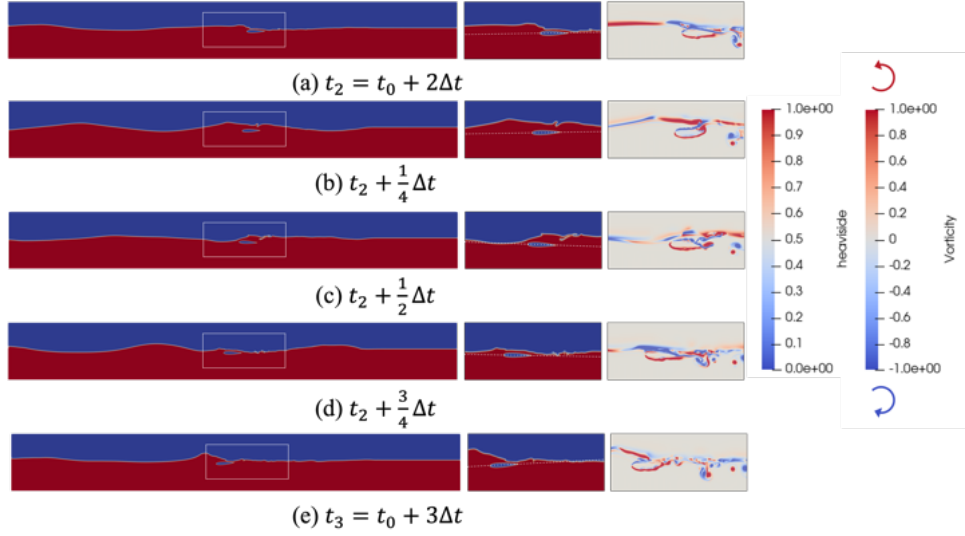


Figure 4.12: The position of the passive hydrofoil in one wave period (wavelength=3.75 m, $t_0 = 1.95$ sec, time interval $\Delta t = 1.70$ sec is the wave period). The snapshots on the left show the Heaviside function, while those on the right show the zoomed-in vortex fields. The dash lines are the chord lines.

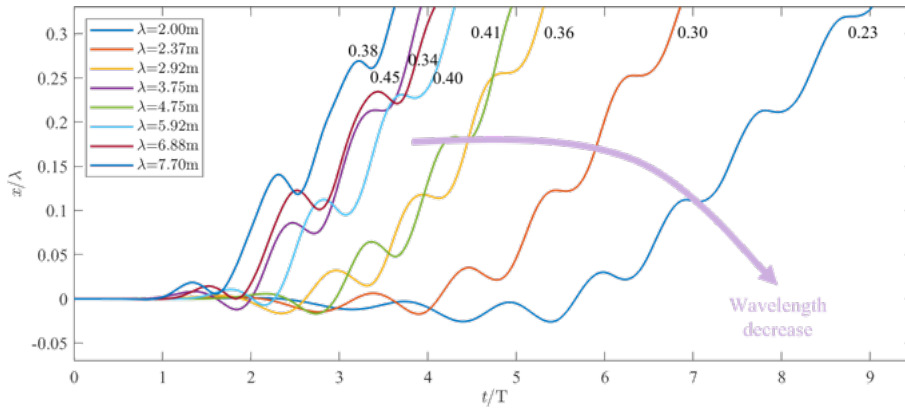


Figure 4.13: The non-dimensional displacement x/λ of the passive hydrofoil with time over wave period t/T facing different waves ($\Delta x = \frac{1}{160}$ m).

waves, it begins to flap with the wave motion. The presence of pitch stiffness causes the foil's pitch angle to deviate from zero, and the spring attempts to restore it to its equilibrium position. As a result, the flapping frequency of the foil does not synchronize exactly with the wave frequency. The resulting thrust coefficients (C_t) are plotted against k_p in Figure 4.14.

Fig.4.14 illustrates the influence of pitch stiffness k_p on thrust generation C_t , revealing two distinct response regions. Initially, at low stiffness, thrust is negligible. As stiffness increases, the thrust coefficient rises sharply (region (I)), peaking at a stiffness of 129.24 N·m/rad. Beyond this peak, further increases in stiffness result in a decrease in thrust (region (II)). The relationship between thrust generation and stiffness is linked to the Strouhal number's optimal range for effective propulsion. As highlighted earlier, optimal C_t is achieved when $0.2 \leq St_A \leq 0.4$ (19). The system's flow dynamics are influenced by vary-

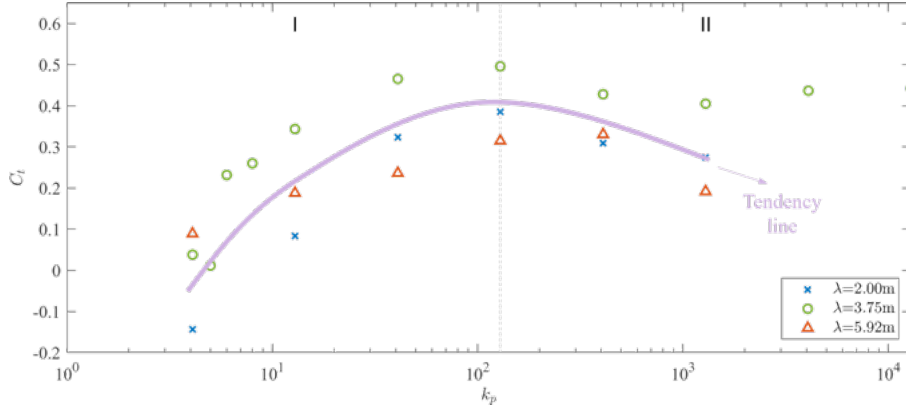


Figure 4.14: The thrust coefficient C_t of the passive hydrofoil with different pitch stiffness k_p ($\Delta x = \frac{1}{160}$ m).

ing wavelengths (from 2.00 m to 5.92 m), which affect the flow velocity and consequently the Strouhal number. At higher stiffness levels, the system becomes more rigid, leading to reduced flapping amplitude, which impacts the Strouhal number. The changes in thrust due to variations in stiffness and wavelength suggest that there is an optimal stiffness level that maximizes thrust across different wave scenarios. This behaviour is also supported by Wang et al.'s experiments (35, 36).

Fig.4.15 illustrates how the hydrofoil's position and the vorticity field vary with pitch stiffness k_p , all under wave conditions with a 3.75 m wavelength. The images captured at $t_0=8.99$ sec show the fully-developed vortex streets behind the hydrofoil. The rotational angle deviation of the hydrofoil, $\Delta\theta$, diminishes with increased pitch stiffness, leading to different hydrofoil responses to identical wave conditions. At zero pitch stiffness (Fig.4.15(a)), there's no restoring force, resulting in a hydrofoil that moves passively with the wave and only a weak vorticity field occurs at the leading and trailing edge. However, even a small stiffness value creates a regulating moment (Fig.4.15(b)), enabling a controlled pitch motion and a transition from drag to thrust, propelling the hydrofoil forward. As pitch stiffness rises, the pitch motion narrows, generating smaller vortices at higher frequencies, as observed in parts (c) and (d) of Fig.4.15. The optimal stiffness for maximum thrust suggests starting with 129.24 N·m/rad for a NACA0015 hydrofoil with a 0.4 m chord. While pitch stiffness optimization can enhance thrust, the maximum C_t remains wavelength-dependent. For those investigating the ideal pitch stiffness for maximum thrust, starting with a pitch stiffness of 129.24 N·m/rad for a 0.4 m NACA0015 hydrofoil is advisable. It is also noteworthy that although optimizing the pitch stiffness will lead to a higher thrust coefficient, the maximum threshold for C_t is mostly defined by the wavelength, see Fig.4.14.

4.6 Conclusions

This study applied CFD to examine wave-induced thrust on hydrofoils, with a special focus on the effect of pitch stiffness. The numerical approach was verified with established benchmarks, including the performance of a foil in uniform flow, to confirm its accuracy for analysing thrust generation mechanisms in varied hydrodynamic conditions.

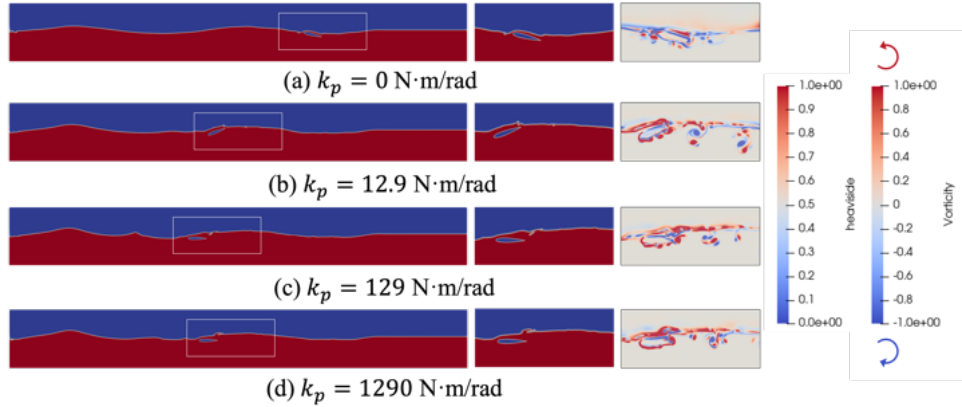


Figure 4.15: The snapshots of the passive hydrofoil under different pitching spring stiffness k_p (wavelength=3.75 m, $t_0 = 8.99$ sec). The series of snapshots on the left shows the Heaviside fields, while those on the right show the zoomed-in Heaviside and vortex fields.

Investigations into the foil in uniform flow highlighted how alternating vortex streets—RvK and vK—correspond with shifts in thrust and drag. These dynamics are essential for propelling the hydrofoil against waves and are crucial for performance optimization. The study also assessed the hydrofoil’s response to heading regular waves, examining a range of pitch stiffness up to 12900 N·m/rad and wavelengths up to 7.70 m. The findings indicate that:

- Hydrofoil advance speeds vary with wavelength, reaching peak efficiency at 3.75 m wavelength.
- A nonzero pitch stiffness is necessary for the hydrofoil to counteract incoming waves.
- There are two distinct responses to pitch stiffness: an initial increase in thrust coefficient with increasing stiffness, followed by a decrease as the hydrofoil becomes overly stiffer.
- A pitch stiffness of 129.24 N·m/rad marks the transition between the two responses, recommended as an optimal starting point for maximizing thrust in regular waves.

These conclusions provide a pathway for further refinement of hydrofoil applications in wave-dominated environments.

4.7 Acknowledgement

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Chapter 5

Stabilisation of Floating Wind Turbine Platforms with Wave Energy: Lab Evidence

5.1 Abstract

Wave Devouring Propulsion (WDP) is a green propulsion method that uses submerged foils to convert wave energy into thrust, serving both as an auxiliary propulsion system and as a stabiliser for maritime structures. This study highlights WDP's effectiveness in improving stability for semi-submersible offshore wind turbine platforms. Experiments on a 1:100 scaled model in regular head wave conditions were conducted in both free-floating conditions, and with a mooring system to validate WDP's effectiveness. The research investigates the effects of foil quantity and placement, including the pitch angle of front foils and the positioning of offset foils. Results show that the integrated foil design reduced mooring line tension by 41.07% compared to the design without foils in specific scenarios, suggesting a promising avenue for future research and application.

Keywords: Flapping foil; Wave-induced propulsion; Semi-submersible platform; offshore wind.

5.2 Introduction

Currently, the Levelized Cost of Energy (LCOE) for floating offshore wind significantly exceeds that of fixed offshore wind by more than threefold (1). This discrepancy is attributed primarily to the challenges posed by the excessive and violent motions experienced by floating wind platforms. These motions, triggered by severe operational and/or survival storm conditions, exert considerable strain on the mooring systems and induce large motions and vibrations in the turbines. To mitigate these issues, this study proposes the application of Wave Devouring Propulsion (WDP) techniques for platform stabilisation. To enhance the stability of Floating Offshore Wind Turbines (FOWTs), three principal design philosophies are generally employed: buoyancy stability (as seen in barge-type platforms), ballast stability (utilised in spar-type platforms), and mooring stability

(applied in tension-leg and semi-submersible platforms) (2, 3, 4). These design concepts is illustrated in Fig.5.1. Various platform designs are paired with corresponding mooring system configurations to achieve stability, taking into account the environmental conditions of the site. The mooring systems hold particular significance for semi-submersible platforms, which are among the most prevalent forms of FOWT (5).

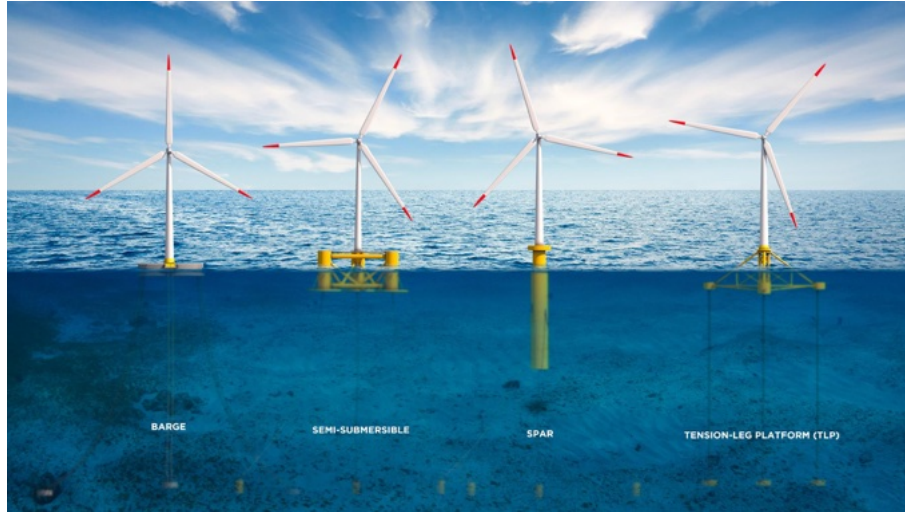


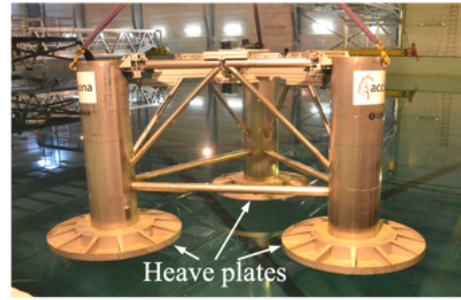
Figure 5.1: Comparative structures of offshore wind turbine foundations (3).

Existing technologies employed for stabilising offshore platforms in the industry fall into three main categories. The first category focuses on increasing the hydrodynamic damping of the floating platform. For example, barge-type floating platforms frequently incorporate a moonpool—a central opening within the hull—intended to dissipate energy and lessen the effects of wave-induced motions (2, 6). An illustration of this is the Floatgen project, a 2 MW floating wind turbine demonstrator in France, facilitated by Ideol, which employs a moonpool (depicted in Fig.5.2(a)). Similarly, semi-submersible platforms might feature heave plates affixed to the columns' bottoms, as shown in Fig.5.2(b), introducing added hydrodynamic damping forces through vortex generation at the edges (5, 7). However, both moonpools and heave plates have shown limitations in efficiently reducing platform drift caused by waves and currents, leading to increased axial tension in the mooring lines.

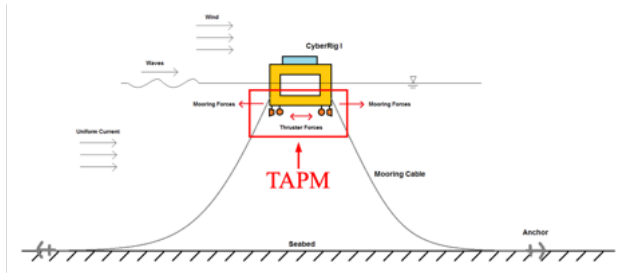
The second category to reducing mooring loads involves the use of Thruster-Assisted Position Mooring (TAPM) systems or spring-like connectors. TAPM systems, illustrated in Fig.5.2(c), are prevalent in the oil and gas sector for reducing forces on mooring lines and platform offset (8). Besides offering standard Dynamic Positioning (DP) functionalities, a TAPM system incorporates a mooring system that delivers passive restoring forces and moments, leveraging the advantages of both passive mooring and active DP systems. Despite their efficacy, the use of TAPM systems with DP thrusters is marked by substantial energy consumption, along with elevated operational and maintenance expenses. Moreover, spring-like connectors, referenced in Fig.5.2(d), designed to integrate into the mooring line, aim to mitigate peak mooring loads (9). Nonetheless, the integration of such connectors introduces a heightened risk of wear and tear on moving parts, potentially compromising system reliability over time.



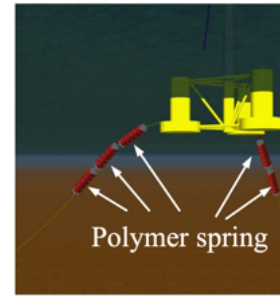
(a) Moonpool of Floatgen (10)



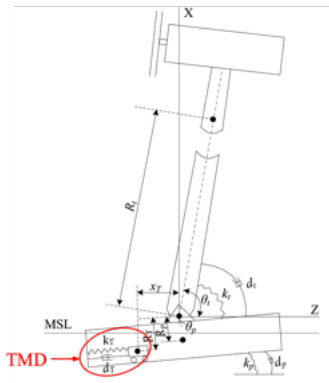
(b) Heave plates under semi-submersible platform (11)



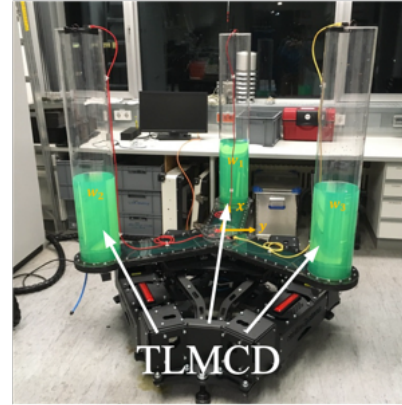
(c) TAPM systems (12)



(d) Spring-like connectors (13)



(e) TMD (14)



(f) TLMCD (15)

Figure 5.2: Different types of offshore platform stabilisation technologies.

The third category focuses on vibration damping systems designed to reduce the motion of the platform by absorbing and dissipating energy. An example of this is the use of Tuned Mass Dampers (TMDs) (14), which are systems that use a secondary mass that oscillates out of phase with the platform to counteract unwanted vibrations, as shown in Fig.5.2(e). Similarly, Tuned Liquid Multi-Column Dampers (TLMCDs) are employed in FOWTs to provide bi-directional damping. TLMCDs in Fig.5.2(f) use the movement of liquid within multiple columns to reduce the platform's response to wave and wind-induced motions (15). Both TMDs and TLMCDs offer significant potential in enhancing platform stability by targeting and mitigating specific resonant frequencies that contribute to excessive motion. However, the design and implementation of these systems require careful consideration of the platform's dynamic characteristics to achieve optimal performance.

The aforementioned methods, while innovative, exhibit certain limitations in stabilising offshore platforms. In this context, Wave Devouring Propulsion (WDP) is a novel solution that promises a more effective approach to overcoming these challenges. Inspired by the natural movements of aquatic animals, particularly whales, WDP utilises flapping foils to convert wave energy for thrust generation (16, 17, 18, 19, 20, 21, 22, 23). By integrating these foils with offshore platforms, the motion of the foils and the movement of the platform in the marine environment translates the kinetic energy of waves into stabilising forces for the platforms. Distinguished by its passive operation, WDP requires no external power sources or feedback mechanisms. This innovative method offers a compelling advantage by significantly reducing platform vibrations and mooring line loads, all while necessitating minimal maintenance.

While WDP technology has attracted considerable interest from both the scientific and commercial sectors, its application has predominantly been in the context of marine vehicles, such as Unmanned Surface Vehicles (USVs) (24, 25, 26) and ships (27, 28). Notable among these is the Suntory Mermaid II (27), a wave-powered vessel that achieved a landmark trans-Pacific voyage from Honolulu. Despite these successful implementations, the integration of WDP technology in offshore platforms remains a largely uncharted domain (29), presenting a promising avenue for future research and development in marine propulsion systems. Here, it is worth highlighting a study by Yoshida et al. (30), who also attached the WDP to the platform. However, their approach differs significantly from the present study. While both studies attach the plate to the platform's lower beam, Yoshida et al. (30) used a flat plate, whereas this research employs a NACA0030 foil shape, emphasizing the influence of the foil shape and pitch angle over the motion itself. Furthermore, Yoshida et al.'s (30) platform was not a standardized semi-submersible platform, and their experiments were conducted entirely under free-floating conditions, without considering the advantages of WDP for mooring lines. Their focus was mainly on the changes in platform position due to wave angle variations, monitored via video camera tracking, without addressing changes in heave, pitch, or drift speed.

This study innovatively explores the utilisation of Wave Devouring Propulsion (WDP) for semi-submersible platforms, employing scaled-down models within a wave tank under regular wave conditions. The research examines the optimal configuration of foils attached to the platform, considering variables such as the quantity of foils, direction, placement, angular orientation, and wave conditions. Through pilot testing and free-floating tests, the most suitable configuration design is explored. Subsequently, moored tests are conducted to further investigate and validate the effectiveness of WDP.

The paper is organised as follows: Section 5.3 provides an in-depth description of the experimental arrangement, detailing the scaled-down model and foil plates, the setup of the wave tank, and the specific wave conditions. Section 5.4 conducts exploratory research to identify the optimal number, placement, and orientation of the foil plates. Building upon these findings, Section 5.5 investigates the impact of different foil rotation angles under varying wave conditions, with the aim of identifying the most effective design and determining the most efficient wave conditions for this design. Further, in Section 5.6, the proposed design is validated by distinct the mooring line tension reduction. Finally, Section 5.7 summarises the conclusions of the study.

5.3 Experimental Setup

5.3.1 Platform Model

This study adopts semi-submersible platforms as the chosen model, a prevalent type of offshore wind platform, with its layout based on the OC5 DeepCwind floating system (31). To ensure the platform's behaviour in water closely mimics that of its real-life prototype, it was scaled down with the scale factor of 100 ($\lambda=100$).

Fig.5.3 displays the 3D CAD model of the platform along with a photograph of its placement in the tank. The wave propagates along the positive x-direction. The platform model is semi-submerged in the tank with calm water, and its draft, which is the depth of the platform below the still water level (SWL), measures 0.2 m. The platform consists of one main column at the centre and three offset columns. Each offset column is composed of a 5 mm thick acrylic cylinder with a polyethylene foam offset base. These three offset columns are arranged in the shape of an equilateral triangle and are connected to the main cylinder through aluminium hollow beams. The connection between the three offset columns and the main cylinder is facilitated by fifteen aluminium hollow beams. To be specific, the top and bottom sections each have six aluminium beams, with three beams connecting to each offset column and the remaining three beams connecting with the middle cylinder. Additionally, three cross-brace beams extend from the top of each offset column to the bottom of the main cylinder, providing further structural stability. Tab. 5.1 provide a comprehensive summary of the platform model's dimensions.

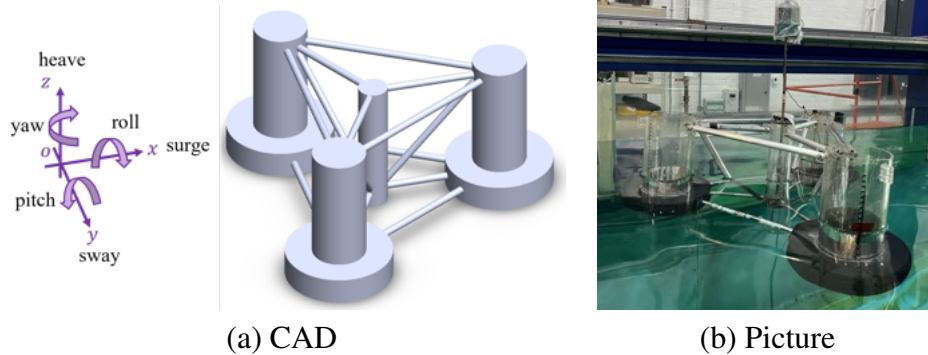


Figure 5.3: 3D CAD model and photograph of the semi-submersible platform placed in the tank.

5.3.2 Foil Models and Parameters

The study utilises NACA type foils, specifically NACA0030, as the chosen foil geometry for Wave Devouring Propulsion (WDP) applications. NACA foils are widely used in aerospace and marine engineering due to their standardized and reliable aerodynamic properties. While NACA0012 and NACA0015 are common in WDP studies (21, 29), their 12% and 15% thickness-to-chord ratios do not capture the characteristic fish-like shape with a rounded head and tapered tail. In contrast, the NACA0030 foil, with its 30% thickness-to-chord ratio, better represents this shape, offering a clear distinction from

Table 5.1: The dimensions of the platform model

Properties	Units	Scaled model	Full model	Scaling Factor (1:100)
Depth of platform base below SWL (draft)	m	0.2	20	λ
Elevation of main column below SWL	m	0.1	10	λ
Elevation of offset columns above SWL	m	0.12	12	λ
Height of offset columns	m	0.26	26	λ
Height of foam bases	m	0.06	6	λ
Depth to bottom beams below SWL	m	0.14	14	λ
Diameter of main column	m	0.065	6.5	λ
Diameter of offset columns	m	0.12	12	λ
Diameter of foam bases	m	0.24	24	λ
Diameter of beams	m	0.016	1.6	λ
Thickness of beams	m	0.0015	0.15	λ
Distance between each offset columns	m	0.5	50	λ
Platform mass including ballast	kg	12.919	12919000.00	λ^3

flat plate foils often used in WDP research (30, 32). The choice of a symmetric foil like NACA0030 is also intentional. WDP relies on symmetric foils to generate reverse Kármán vortices and produce thrust. Although asymmetric foils could be effective, they have not been thoroughly studied, and they may lead to uneven wake thrust, potentially affecting stability. Further research into asymmetric designs is warranted, but for this study, the focus was on achieving stable thrust with a well-documented symmetric fish-like foil.

To facilitate the experiments, four types of NACA0030 foil plates were prepared using 3D printing and PLA Plastic (see Fig.5.4). The foil plate configurations are as follows: 1. 100 mm chord foil with a front hole. 2. 100 mm chord foil with a rear hole. 3. 150 mm chord foil with a front hole. 4. 150 mm chord foil with a rear hole. It is noteworthy that all the foil plates share a consistent span of 230 mm. The foil plates have a density of 314.6 kg/m³. These foils exclusively serve as the 'foil' elements, preserving the platform's intended behaviour while effectively fulfilling their role in wave devouring propulsion.

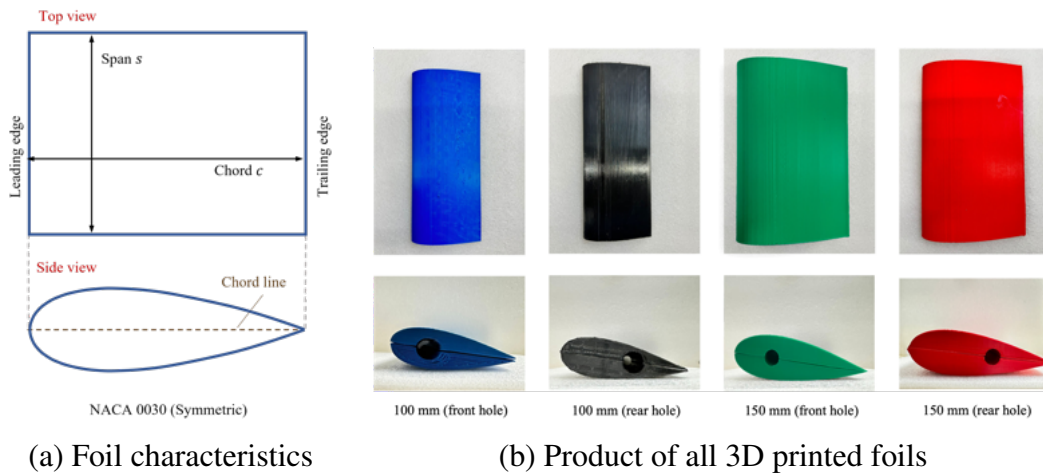


Figure 5.4: Four types of NACA0030 foil plates used for the study.

5.3.3 The assemble of platform and foil

The Cranfield University PSE lab wave tank/towing tank has three flaps as the wave generator on one end, and a metal beach as wave absorption on the other end. The wave tank frame has a size of 30 m long, 1.5 m wide and 1.5 m water depth, as shown in Figure 5.5. The centre of the platform is positioned 10.92 m away from the side of the tank where waves are generated. The test distance is the spacing between the two tank pillars, which is 2.37 m.

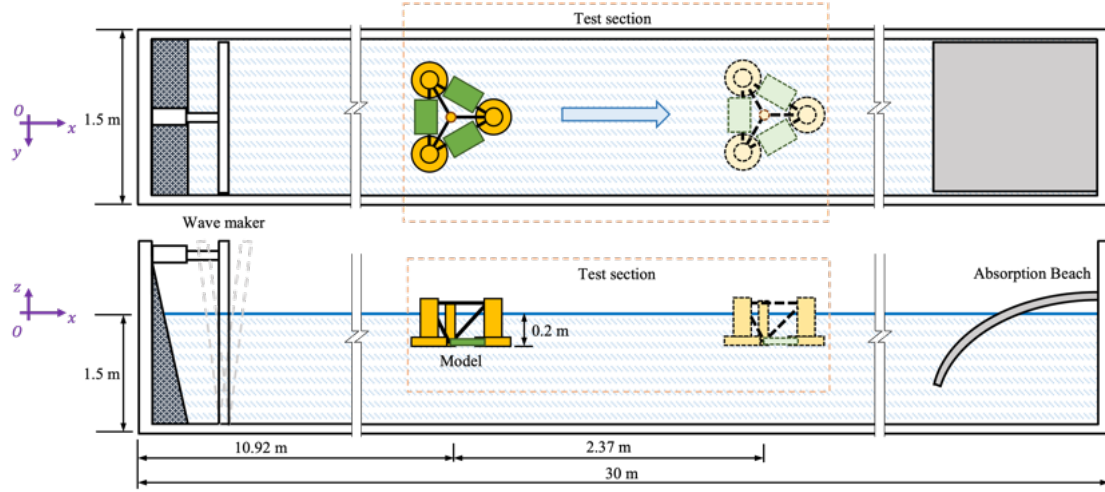


Figure 5.5: The schematic diagram of the wave tank setup (with 3-foil setup as example).

The foils are positioned on the lower beams between two foam offset bases. Figure 5.6 illustrates the appearance after installing three horizontal foil plates along with the dimensions of the model. One of the primary motivations for this placement is the understanding that the closer the foil is to the water surface, the more wave energy is available for conversion, resulting in a greater thrust (33, 29). Additionally, the beam provides a convenient point of attachment for the foil, and its surrounding space remains unobstructed, ensuring the platform's original design remains intact. Given these factors, this location was deemed more suitable than other potential sites such as the bottom of the platform. To capture heave motion data, the GoDirect Force and Acceleration Sensor is firmly situated at the platform's central screw (see in Fig.5.6(a), the grey rectangular part in the side view). The sensor measures force from ± 50 N with a resolution of 0.01 N, acceleration up to ± 16 g with a resolution of 0.01 g, and rotational motion with a resolution of 0.1° . It supports a maximum sampling rate of 200 Hz. For this experiment, data was sampled at 50 Hz to accurately capture dynamic responses. The sensor was calibrated using Go Direct software before each test.

5.3.4 Similarity Analysis

In this study, geometric similarity was applied, where the model was scaled down by a factor of 100. This approach ensures that the model and prototype maintain consistent shape and size ratios, providing a foundational similarity criterion for the research. To further analyze the motion of the foil and platform in the marine environment, a dimensional analysis was conducted using the Buckingham Pi theorem. The main parameters

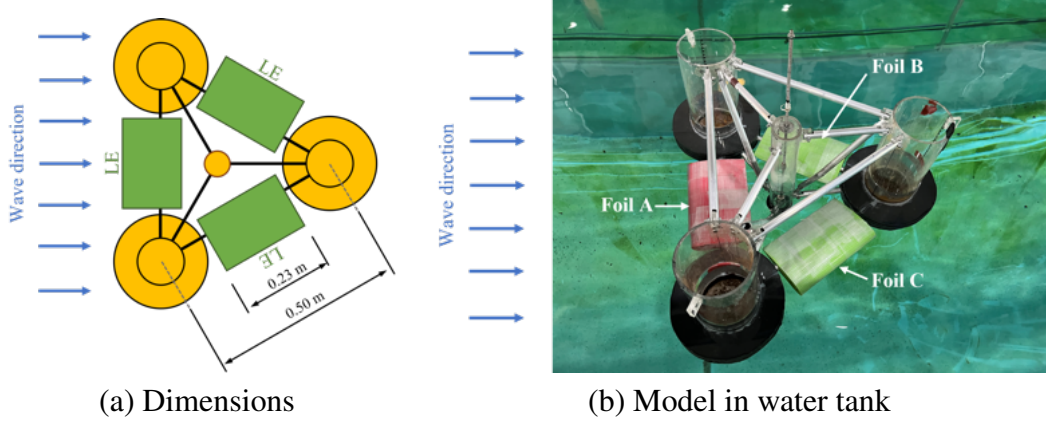


Figure 5.6: The dimensions of one assembled model (semi-submersible platform with 3 horizontal foils).

considered include the fluid velocity U , characteristic length L , fluid density ρ , fluid dynamic viscosity μ , gravitational acceleration g , and oscillation frequency f . U , c , ρ are chosen as independent variables. The resulting dimensionless numbers are the Reynolds number ($Re = \frac{\rho UL}{\mu}$), the Froude number ($Fr = \frac{U}{\sqrt{gL}}$), and the Strouhal number ($St = \frac{fL}{U}$). It is important to note that the characteristic length L should be selected appropriately depending on the specific flow under consideration, as shown in Eq.5.1.

$$Re = \frac{\rho c U_{\infty}}{\mu} \quad (5.1a)$$

$$Fr = \frac{U_{\infty}}{\sqrt{gB}} \quad (5.1b)$$

$$St_A = \frac{fA}{U_{\infty}} \quad (5.1c)$$

where c is the chord length of foil, B is the length of the platform (i.e. distance between each offset column of platform) and A is the peak-to-peak oscillating amplitude. It is assumed that the direction of U_{∞} is always aligned with the heading wave direction. The oscillating amplitude A is often taken as the width of the wake.

In offshore engineering, it is challenging to simultaneously satisfy all three similarity criteria due to their conflicting requirements, particularly in scaled models. Specifically, when the focus is on the motion of a foil underwater, Reynolds similarity is predominant; when the focus is on the motion of a semi-submersible platform in the ocean, Froude similarity is dominant; and when the focus is on the periodic motion of either the foil or the platform, Strouhal similarity becomes more relevant. Given that the primary objective of this study is to examine the effects of WDP on platform motion, it is suggested that Froude similarity should be prioritized in model tests. According to Chakrabarti (34), Froude's law or similitude is the most suitable scaling law for free floating and mooring tests concerning hydrodynamic similitude. This law dictates that the Froude number of the scaled model and the prototype should remain the same, while the dimensional relationship between the scaled model and the prototype should follow a linear proportion with the scale factor.

This study deliberately avoids using dimensionless analysis, focusing instead on pre-

senting the experimental data directly. Applying one similarity analysis from a scaled experiment to real-world scenarios can be inappropriate due to varying research perspectives. However, future studies could benefit from employing 3D numerical simulations. These simulations could validate the laboratory data at full scale. Additionally, they would help address the limitations of scaled models. By doing so, a more comprehensive application of dimensional analysis across different similarity criteria would be possible.

5.3.5 Wave Condition

Regular wave is selected as the wave condition. Table 5.2 presents the pitch angles and wave conditions used in all scenarios for this experiment. The data illustrates the range of wave scenarios analyzed to evaluate the system's performance under different conditions. The flow conditions, in the form of horizontal and vertical velocity, are calculated using linear wave theory and specifically:

$$u(x, z, t) = \frac{\pi H}{T} \frac{\cosh[k(z + d)]}{\sinh(kd)} \cos(kx - \omega_w t) \quad (5.2a)$$

$$v(x, z, t) = \frac{\pi H}{T} \frac{\sinh[k(z + d)]}{\sinh(kd)} \sin(kx - \omega_w t) \quad (5.2b)$$

where, u and v are the instantaneous horizontal and vertical particle velocities at (x, y) location, H is the wave height, T is the wave period, d is the water depth, $k = \frac{2\pi}{\lambda}$ is the wave number, $\omega_w = \frac{2\pi}{T}$ is the angular wave frequency, and λ the wavelength calculated with the linear dispersion equation, Eq.5.3.

$$\lambda = \frac{gT^2}{2\pi} \tanh(kd) \quad (5.3)$$

To assess the performance under different wave conditions, appropriate wave conditions are selected based on the St_A number, refer to Eq.5.1c. For the passive running test, the platform's motion is influenced by the waves. The model oscillates up and down in sync with the wave frequency, and the wave height H can serve as an estimate for the platform's heave amplitude. This is because, under wave influence, the vertical displacement of the platform closely follows the wave height. While other factors like inertia and damping might affect the actual amplitude, H provides a reasonable first approximation. Thus, in Eq.5.1c, the oscillating frequency f can be reasonably replaced with wave frequency f_w and the oscillating amplitude A can be approximated with the wave height H . Regarding the characteristic flow velocity, U_∞ , five options are available: wave celerity, horizontal and vertical velocities at the foil's initial submerged location ($z = -0.17$ m), or the horizontal and vertical velocities at the still water level ($z = 0$). For the designed model (a combination of WDP and platform), the foil experiences varying flow conditions due to wave-induced kinematics at different depths. The selection of the characteristic velocity should be based on the foil's motion, as the study focuses on the foil's WDP capability. Therefore, choosing the horizontal and vertical velocities at the foil's initial submerged location ($z = -0.17$ m) is more appropriate than simply using the wave celerity or velocities at the surface. Additionally, the horizontal velocity magnitude is larger than the vertical velocity and acts over the foil surface, producing desired effects such as a vortex

Table 5.2: Test matrix for all conditions.

Pitch angle (°)	Type	Water depth d (m)	Height H (m)	Frequency f (Hz)	Wave period T (s)	Wavelength λ (m)
Free-floating Tests						
180	Airy	1.50	0.15	0.65	1.54	3.65
150	Airy	1.50	0.15	0.65	1.54	3.65
90	Airy	1.50	0.15	0.65	1.54	3.65
30	Airy	1.50	0.15	0.65	1.54	3.65
15	Airy	1.50	0.15	0.65	1.54	3.65
0	Airy	1.50	0.15	0.65	1.54	3.65
-15	Airy	1.50	0.15	0.65	1.54	3.65
-30	Airy	1.50	0.15	0.65	1.54	3.65
-90	Airy	1.50	0.15	0.65	1.54	3.65
-150	Airy	1.50	0.15	0.65	1.54	3.65
-180	Airy	1.50	0.15	0.65	1.54	3.65
30	Airy	1.50	0.15	0.45	2.22	6.79
30	Airy	1.50	0.15	0.50	2.00	5.78
30	Airy	1.50	0.15	0.55	1.82	4.95
30	Airy	1.50	0.15	0.60	1.67	4.25
30	Airy	1.50	0.15	0.65	1.54	3.65
30	Airy	1.50	0.15	0.70	1.43	3.18
30	Airy	1.50	0.15	0.75	1.33	2.76
30	Airy	1.50	0.15	0.80	1.25	2.44
30	Airy	1.50	0.15	0.85	1.18	2.17
30	Airy	1.50	0.15	0.90	1.11	1.92
30	Airy	1.50	0.15	0.95	1.05	1.72
30	Airy	1.50	0.15	1.00	1.00	1.56
Moored Tests						
30	Airy	1.50	0.02	0.65	1.54	3.65
30	Airy	1.50	0.04	0.65	1.54	3.65
30	Airy	1.50	0.06	0.65	1.54	3.65
30	Airy	1.50	0.08	0.65	1.54	3.65
30	Airy	1.50	0.10	0.65	1.54	3.65
30	Airy	1.50	0.12	0.65	1.54	3.65
30	Airy	1.50	0.14	0.65	1.54	3.65
30	Airy	1.50	0.16	0.65	1.54	3.65
30	Airy	1.50	0.18	0.65	1.54	3.65
30	Airy	1.50	0.20	0.65	1.54	3.65

street. The foil's trajectory, induced by waves, is smaller than that of the wave particles at $z=-0.17\text{m}$. At this depth, the wave particle velocity represents the maximum velocity the foil can reach. Therefore, U can be expressed as the maximum horizontal velocity at the initial submerged location of the foil ($z = -0.17\text{ m}$), denoted as U_{umax} , as shown in Eq.5.4a. Finally, Eq.5.1c is presented in a wave-based form as Eq.5.4b.

$$U_u = \frac{\pi H \cosh[k(z+d)]}{T \sinh(kd)} \quad (5.4a)$$

$$St_A = \frac{f_w H}{U_{umax}} \quad (5.4b)$$

Anderson (35) reported optimum thrust generation for $0.2 < St_A < 0.5$, and this range was further supported by simulations conducted by Xing et al. (36). Read et al. (37) not only reinforced Anderson's conclusions but also broadened the range of exploration up to an St_A value of 0.6. Given this confluence of findings, our study strategically focuses on wave conditions with a frequency of 0.65 Hz and a height of 0.15 m (Tab. 5.2). These conditions correspond to an St_A value of 0.42, placing it squarely within the empirically established optimal range, thus offering a promising avenue for identifying conditions of peak efficiency.

5.4 Pilot Tests

Pilot Tests explore various foil configurations in a free-floating setup, examining foil quantity, orientation towards waves, and the significance of installation holes. This stage is aimed at narrowing down the vast array of possible configurations and identifying critical variables for subsequent, detailed tests in the next section.

5.4.1 Effect of Foil numbers

Six distinct models were developed to determine the optimal number of foils, with configurations depicted in Fig.5.7. These were based on the three beams connected to the lower foam base, encompassing models with one (1), two (2), three (3), or no foil plates (numbered as 0). All the foil plates discussed in this section are horizontally positioned using the rear hole, as detailed in Section 5.3.2. It should be noted that for Models 0, 1, 2, and 3, the leading edge (LE) of the foil is directed towards the incoming wave, implying that the model faces the heading wave. Conversely, in Models 1' and 2', the orientation of the initial platform causes the trailing edge (TE) of the foil to be directed towards the incoming wave, suggesting that the model faces the following wave. The inclusion of both TE and LE orientations serves to explore how different directions might affect energy capture efficiency and offers a more comprehensive guide for practical applications.

Under regular wave conditions, the average drifting speed of the models was recorded within the test section with the drifting distance of 2.37 m. The average speed was determined by recording the time taken for each pass through the test section. The speed was then calculated using the formula: speed = distance / time. To minimize the influence of 3D effects, only tests where the model maintained a straight-line drift in the test section without any deviation or offset were considered valid. To ensure the reliability of the findings, results were averaged from three valid trials for each configuration. The summarised results of this study can be found in Fig.5.8.

In the study presented in Fig.5.8, a comprehensive evaluation of drift speeds across different foil configurations was conducted. When the platform operated without any foils

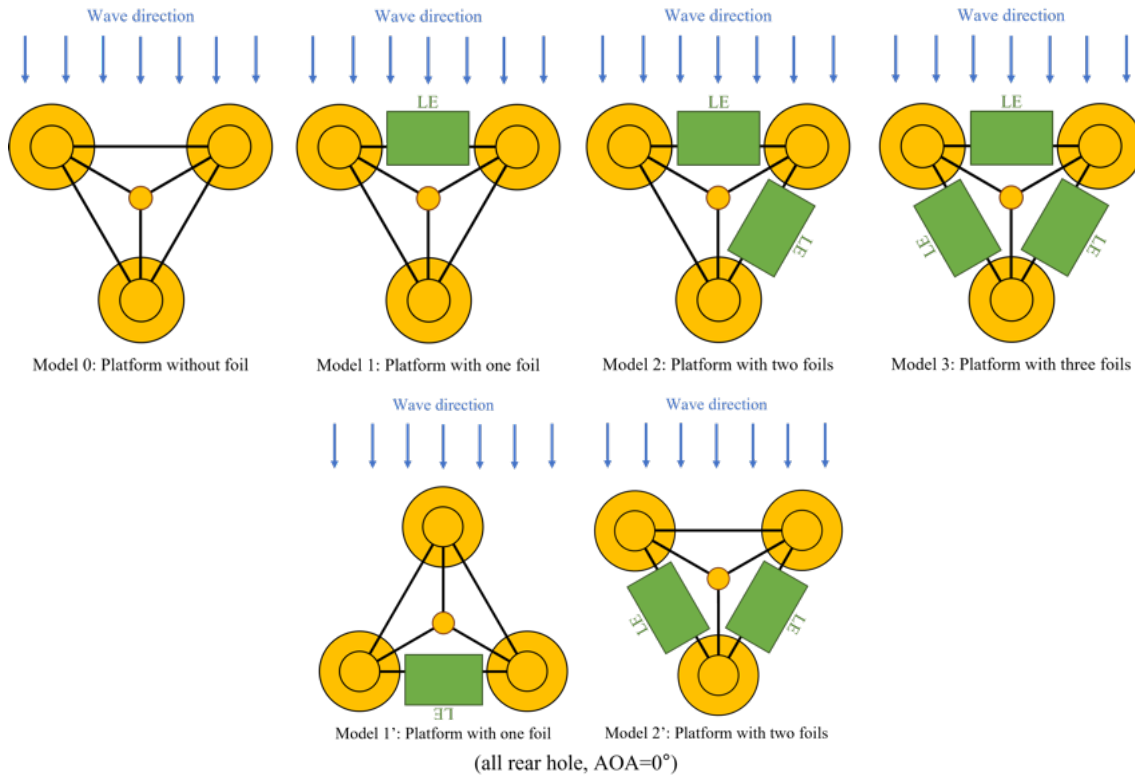


Figure 5.7: Schematic illustrations of the platform and foil(s) combinations: six model scenario.

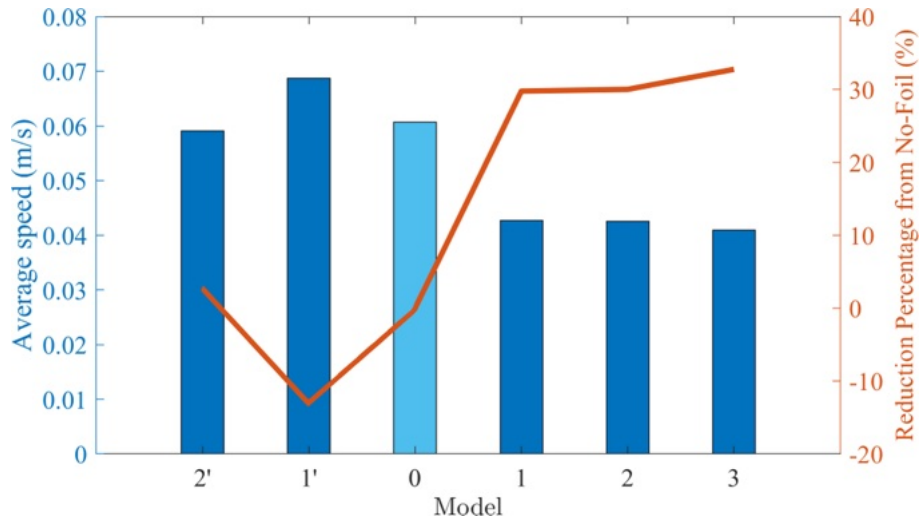


Figure 5.8: Comparison of average drifting speeds across various foil number configurations. (Blue bar represents average speeds, and orange line indicates percentage reduction compared to No-foil setup (Model 0).)

(referred to as Model 0), it exhibited a drift speed of 0.06 m/s. This baseline allowed for a clear comparison against setups that utilised foils. Notably, even with the presence of foils, models exposed to following waves (specifically Model 1' and 2') did not show a significant deceleration. In fact, in the setup where a single foil with its TE faced the

waves (Model 1'), the drift speed was marginally higher than that of the foil-less configuration. An evident observation from the gathered data highlighted the stark difference in drift speeds between scenarios with LE foil setups compared to Model 0 (the 0-foil setup). Specifically, under heading waves, platforms equipped with the foil(s) experienced a drift speed reduction of approximately 30% relative to the configurations without foils. Such a reduction is indicative of the foil's innate capability to undergo vertical oscillations in tandem with the platform's wave-induced motions. This synchronized movement adeptly channels the wave's energy into propulsive thrust for the platform, validating that WDP is predominantly effective under heading wave conditions. This conclusion aligns harmoniously with the physics mechanism of WDP (33, 29, 21).

Among the various setups tests, delving deeper into the specifics of configurations, the 3-foils configuration demonstrated superior performance. While the 1-foil and 2-foil configurations exhibited a drift speed reduction of about 30% compared to the 0-foil setup, the 3-foil setup outperformed them with a notable 32.76% decrease in speed. This setup also presented greater flexibility in terms of arrangements. For instance, one could position one foil with a specific pitch angle while the remaining two foils could be oriented horizontally or vertically. Given the significant advantages observed with the 3-foils configuration, it was deemed essential to further investigate its potential. Subsequent phases of this study aimed to explore variations in foil positioning within the 3-foil setup, with an overarching objective to determine the most optimal arrangement that would confer maximum stability to the platform.

5.4.2 Effect of Hole position

In this section, a series of free-floating tests are presented to assess the impact of varying foil hole positions on the overall model performance. The motivation behind this investigation arises from the nuances of foil installation onto the platform. Recognizing the significance of how the foil is installed on the platform, the section specifically examines whether, at a consistent foil submergence depth, the position of the hole where the foil is anchored to the platform substantially influences the overall performance. This determination will inform if such nuances warrant consideration during the design phase.

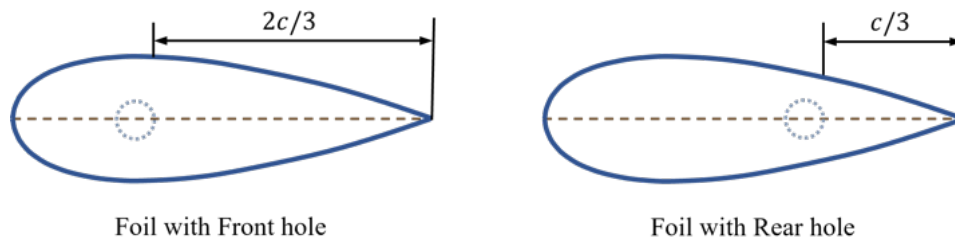


Figure 5.9: Schematic illustrations of different hole positions.

Fig.5.9 illustrates two distinct foils used in this section, each with a different hole position. Two prominent holes, termed the 'front hole' and the 'rear hole', were considered. As their names suggest, the distinction between them lies in their proximity to the LE. The diameter of the holes is designed to be identical to that of the construction beams

Table 5.3: Average drift speeds for different foil hole positions

Hole position	Ave speed (m/s)	Decrease	Difference
Front hole	0.0442	27.20%	2.15%
Rear hole	0.0433	28.77%	

to facilitate assembly and installation. A chord length of 100 mm for the NACA0030 profile was chosen instead of the 150 mm used in other sections. This choice was due to the insufficient space within the platform's crossbeam to horizontally place the 150 mm foil with the front hole in its proper orientation (with the LE facing outwards). Each foil was tested using the Model 1 configuration (Fig.5.7), and the differences in average drift speeds were compiled in Tab. 5.3. As evident from the table, the presence of the foil under heading regular waves slows down the platform's drift speed: a 27.20% decrease with the front hole and a 28.77% decrease with the rear hole. However, the extent of this deceleration is only marginally influenced by the difference in foil positions, with a speed variance of just 2.15%. Such minor variations might not only be attributed to the hole positions but also to several uncontrollable external factors. These can range from water conditions, minor water leakages, to subtle discrepancies in stopwatch or sensor calibrations.

5.5 Free-floating Tests

Following the exploratory Pilot Tests, Free-floating Tests concentrate on foil pitch angle and wave conditions influence to find the optimal design for enhancing platform stability.

5.5.1 Effect of pitch angle

In this section, the study adopts and modifies the Model 3 (3-foil configuration) as the foundational setup, adjusting the angles of the front foil plate. The primary objective is to identify the optimal design for the "WDP+platform" by altering the pitch angle of the foil oriented towards the wavefront. A comprehensive range of pitch angles was tested, spanning from -180° to $+180^\circ$, specifically at: -180° , -150° , -90° , -30° , -15° , 0° , 15° , 30° , 90° , 150° , and 180° , see Fig.5.10. This extensive range allows for an in-depth analysis of the system's response across different attack angles. Due to spatial constraints at the foil's rear upon its installation on the platform, a full 150 mm rear-hole foil flip wasn't feasible. As a result, pitch angles of $\pm 150^\circ$ and $\pm 180^\circ$ employed a front hole, while the range from -90° to 90° utilised a rear hole foil. Consistent with the conditions outlined in preceding sections, the tests were conducted with a wave height of 0.15 m and a frequency of 0.65 Hz. Under free-floating conditions, the model was tested for variations in drifting speed, heave acceleration, and pitch angle changes under head waves at different pitch angles. Each specific configuration underwent three separate tests, with results subsequently averaged to bolster reliability and robustness.

For clear delineation, the foils, placed fixed in three distinct positions, are individually named. The foil facing the head wave is designated as Foil A, illustrated in red in Fig.5.6(b). The other two offset foils are denoted as Foil B and Foil C, portrayed in green

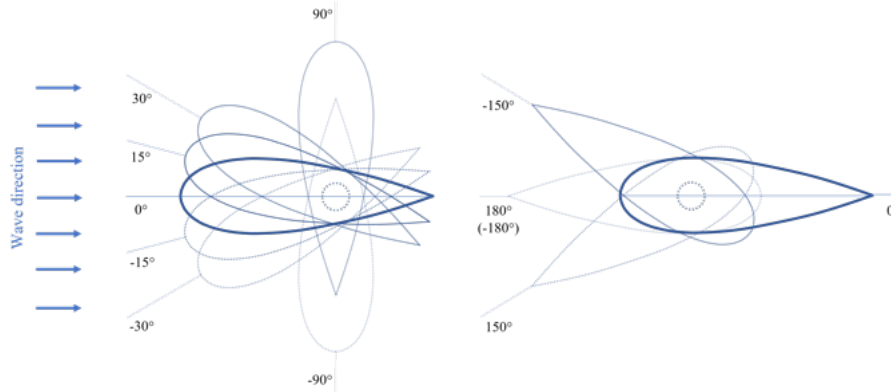


Figure 5.10: The selection of pitch angle for Foil A.

in Fig.5.6(b). Based on the orientation of Foil B and C—either horizontal or vertical—the examination of Foil A's pitch angle is divided into two subsections: Horizontal Supporting Foils and Vertical Supporting Foils.

5.5.1.1 Horizontal Supporting Foils

Within the framework of Model 3, Foil B and C were maintained in a horizontal orientation throughout the experimental tests of this section, meaning that when stationary, the offset foils were parallel to the water surface. By adjusting the pitch angles of Foil A, different drifting speed outcomes were obtained. These results, along with the percentage reduction in drifting speed compared to the platform without foils, are summarised in Fig.5.11. Compared to the no-foil baseline, all pitch angles in this setup significantly reduced the drifting speed. Notably, while Foil A's angles greater than 90° resulted in an average reduction of about 20%, acute pitch angles rendered a more pronounced stabilisation effect, averaging around 30%. The optimal performance was observed at a pitch angle of 30° , achieving a peak drifting speed reduction of 41.04%.

Further, the impact of Foil A's pitch angle on heave acceleration was also examined, as detailed in Fig.5.12. It's evident that the overall reduction isn't substantial, typically not exceeding 5%. The most significant reduction occurred at a pitch angle of 150° , where the heave acceleration decreased by 7.40%.

5.5.1.2 Vertical Supporting Foils

The drifting speeds with vertically positioned foils B and C, adjusted for varying pitch angles, are illustrated in Fig.5.13. Within the range of -30° to 30° , there's a marked reduction in drifting speed, with optimal performance observed at pitch angle= 30° , recording a speed reduction of 60.04% compared to the non-foil platform. Conversely, for obtuse angles of attack ($\pm 90^\circ$, $\pm 150^\circ$, $\pm 180^\circ$), the foil exhibits limited effectiveness in influencing the system's speed. This reveals that, under conditions where the pitch angle is obtuse, the contribution of the foil to system performance is minimal or negligible.

Fig.5.14 presents a comparative analysis of heave acceleration. Data collection began as the wave passed over the platform, during which the platform maintained a stable heave motion. Heave acceleration was recorded over 18 wave periods, capturing the accelera-

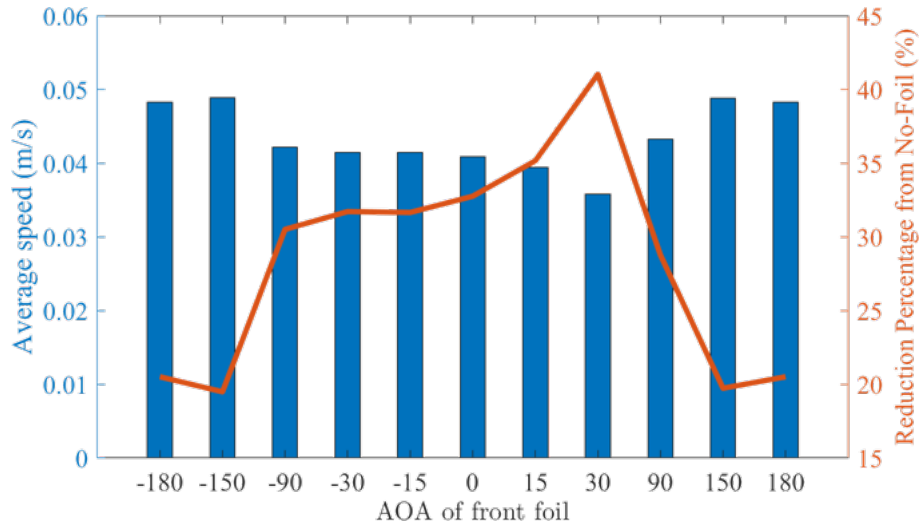


Figure 5.11: Comparison of drifting speeds: varied pitch angle of Foil A with horizontal Foil B and C. (Blue bar represents average speeds, and orange line indicates percentage reduction compared to No-foil setup (Model 0) in light blue.)

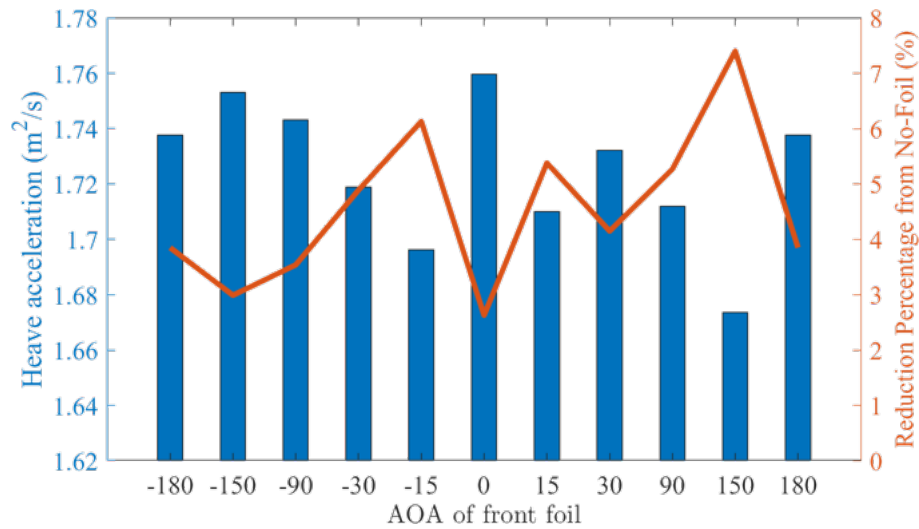


Figure 5.12: Comparison of heave acceleration: varied pitch angle of Foil A with horizontal Foil B and C. (Blue bar represents average speeds, and orange line indicates percentage reduction compared to No-foil setup (Model 0).)

tion for each cycle. The results are summarised and depicted in Fig.5.14. Overall, there is a significant reduction in heave acceleration, generally hovering between a reduction of 12% to 15%. However, an anomaly is observed at pitch angle=90°. Specifically, when all three foils are positioned vertically with their leading edges (LE) facing upwards, the decrease in heave acceleration is recorded at -3.85%. This implies that instead of mitigating the heave motion, this configuration inadvertently amplifies it. The presence of WDP generally leads to a reduction in the platform's heave amplitude, which is a positive outcome. This reduction might also be attributed to an added mass effect similar to that of a heave plate. Regardless of the pitch angle, the reduction in heave acceleration remains relatively consistent, suggesting that the benefits of reduced heave are similar across dif-

ferent pitch angle configurations. Therefore, the primary consideration in selecting the optimal design is the impact on drifting speed, which has a more significant influence.

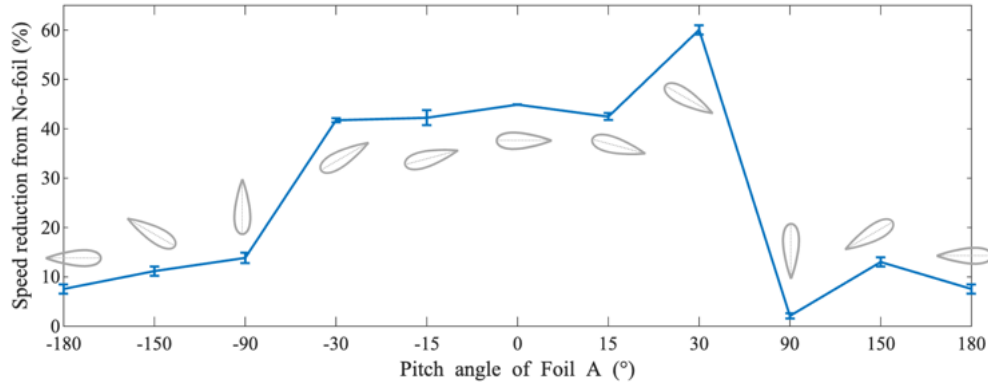


Figure 5.13: Comparison of drifting speeds: varied pitch angle of Foil A with vertical Foil B and C. (The y-axis represents the drift speed reduction of the WDP+platform model compared to the No-foil setup, expressed as a percentage, and the x-axis shows the different pitch angles. The grey foil icon illustrates the actual rotational position of Foil A).

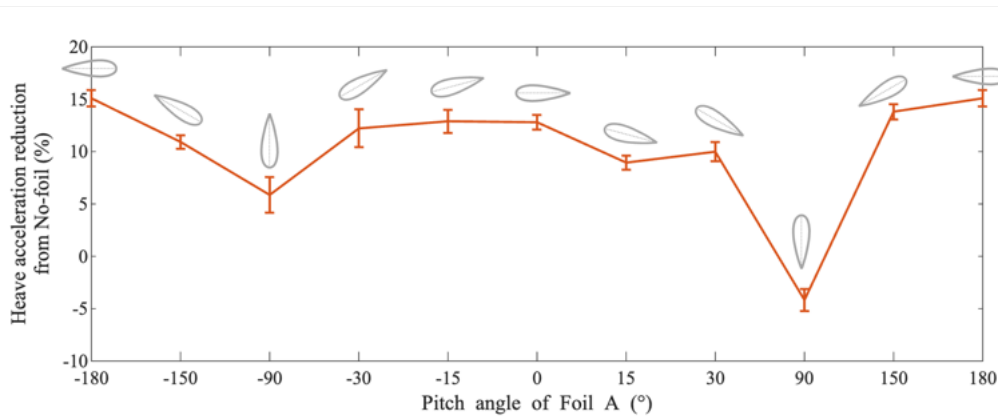


Figure 5.14: Comparison of heave acceleration: varied pitch angle of Foil A with vertical Foil B and C. (The y-axis represents the heave acceleration reduction of the WDP+platform model compared to the No-foil setup, expressed as a percentage, and the x-axis shows the different pitch angles. The grey foil icon illustrates the actual rotational position of Foil A).

5.5.1.3 Comparison

This section provides a comparative analysis of the research findings for Horizontal and Vertical Supporting Foils, aiming to determine the most optimal foil design approach. Fig.5.15 presents a comparative analysis of the drift speed of the model under both horizontal and vertical support conditions. As the pitch angle increases from 0° to 180° , both the vertical and horizontal cases exhibit a slight upward trend in drift speed. Notably, in

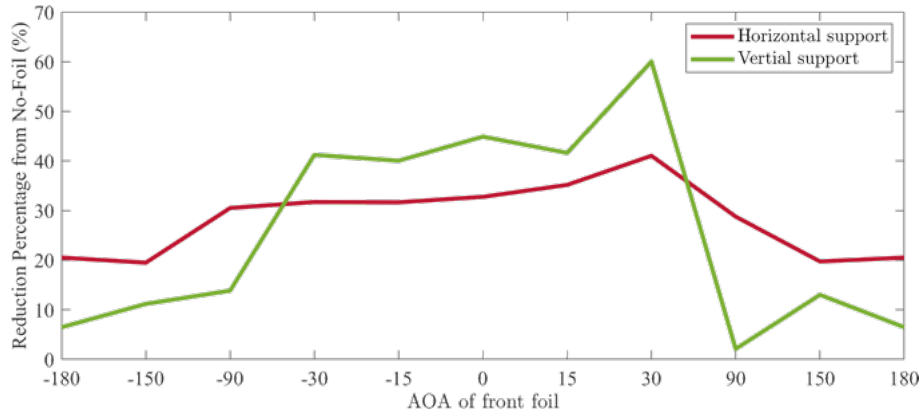


Figure 5.15: Comparison of drifting speeds: horizontal vs vertical Foil B and C.

the acute angles ranging from 0° to 90° , the average speed under the vertical support case is lower compared to its horizontal counterpart. This suggests that positioning foils B and C vertically is more effective in mitigating the influence of wave-induced drift forces on the platform within these attack angles. In particular, at a pitch angle of 30° , the drift speed of the model with vertical offset foils reaches its lowest, indicating optimal resistance against drift forces at this specific angle. This insight holds significant implications for optimizing design configurations to ensure minimal drift in wave-prone environments.

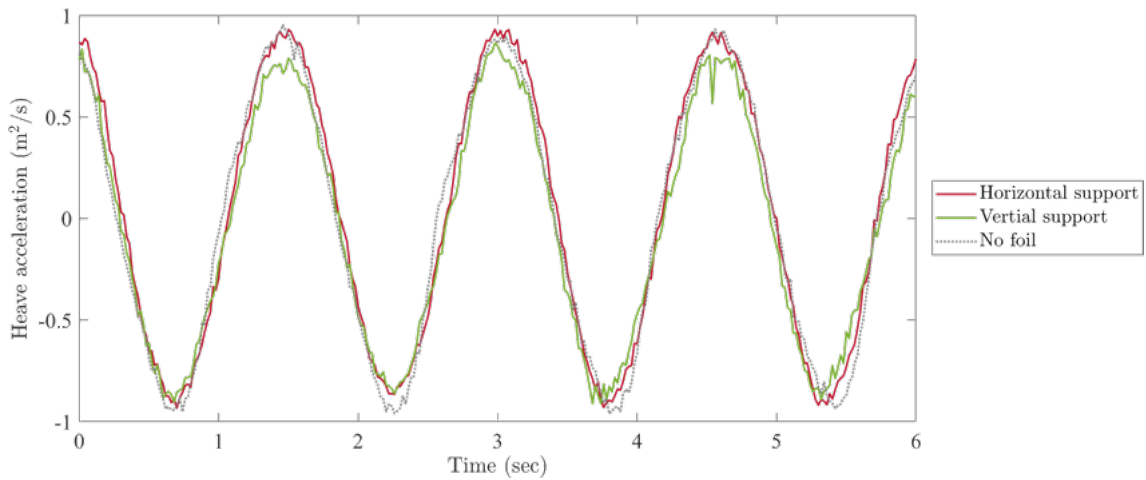


Figure 5.16: Comparison of heave acceleration: horizontal vs vertical Foil B and C with pitch angle of 30° for Foil A.

Both the Horizontal and Vertical setups show that positioning at a 30° pitch angle is optimal. Fig.5.16 compares the heave acceleration across three configurations: Horizontal, Vertical, and No foil. The Vertical setup consistently exhibits lower heave acceleration, indicating its superior ability to resist drift forces. While the Horizontal setup shows similar behaviour, its slightly higher peaks suggest it is less effective than the Vertical configuration. The No foil condition highlights the significant benefits of using a foil.

The corresponding Fast Fourier Transform (FFT) analysis shown in Fig.5.17 further illustrates the effectiveness of the Vertical foil setup. Specifically, the amplitude of the 0.65 Hz frequency component—corresponding to the wave frequency—is significantly

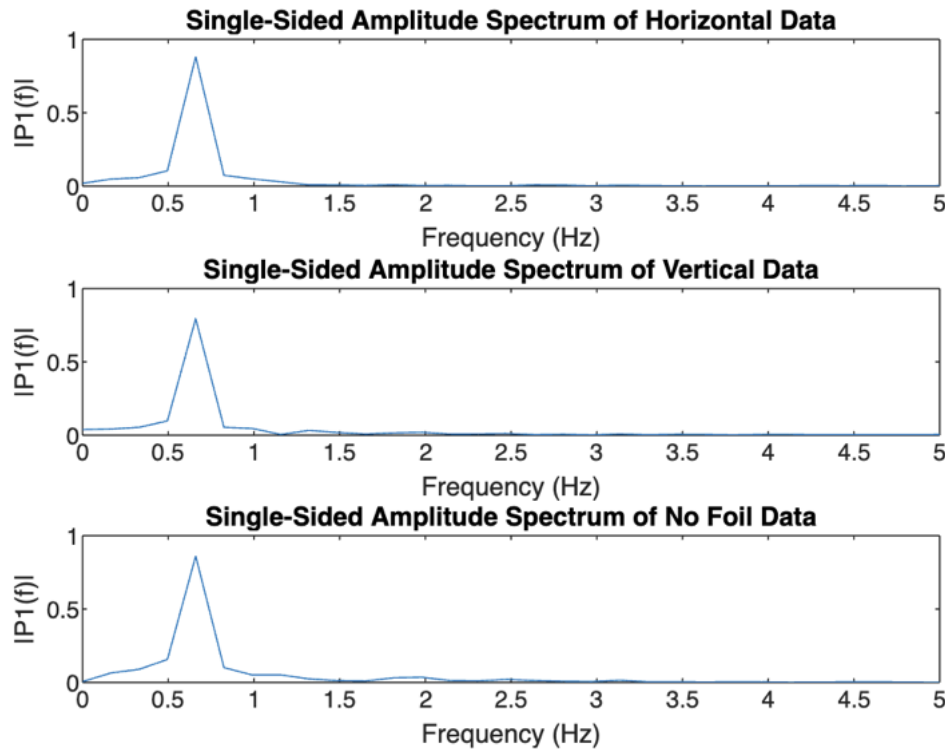


Figure 5.17: Frequency spectrum analysis of heave acceleration for different foil configurations.

lower with the Vertical foil compared to the Horizontal and No foil configurations. This suggests that the Vertical foil is more effective in damping oscillations at this frequency, absorbing or suppressing energy that would otherwise contribute to instability. This finding aligns with the time-domain results in Fig.5.16. Overall, the Vertical orientation at a 30° pitch angle for Foil A proves to be the most effective in reducing wave-induced disturbances.

5.5.2 Effect of Waves

Given the unpredictable nature of real-world marine environments, it's imperative to evaluate the platform's performance under varying wave conditions. Prior research has shown that among various configurations, the three-foil setup – with one foil set at a pitch angle of 30° (referred to as Foil A) and the other two being vertical foils (Foil B and C) – exhibits superior efficacy. This section aims to identify the precise wave conditions where this configuration demonstrates peak performance.

To ensure uniformity in the tests, a wave height of 0.15 m, as outlined in previous sections, was consistently maintained. The wave frequencies were methodically adjusted, ranging from 0.45 to 1 Hz, with increments of 0.05 Hz. To enhance measurement precision, a test distance of 1 m was chosen for determining the platform's drifting speed. Opting for a shorter test distance of 1 m, instead of the 2.37 m used in other sections, minimized inaccuracies due to wave interactions near the tank's shoreline. This is especially relevant when testing high-frequency waves which might cause backflow, sub-

sequently pushing the platform in the opposite direction and causing inconsistent and extended floating durations.

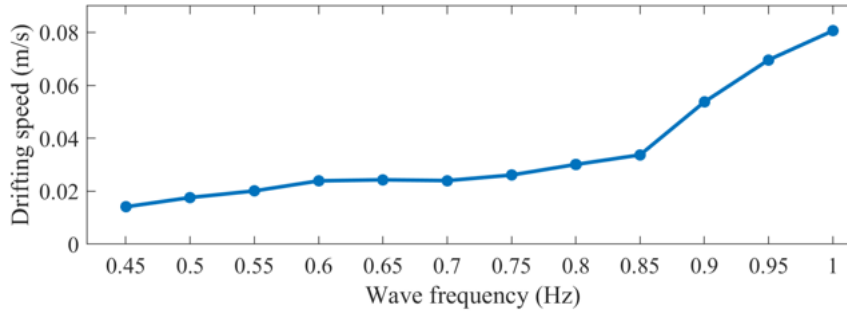


Figure 5.18: The drifting speed under different wave frequencies.

Results, as depicted in Fig.5.18, showed that as wave frequencies increase, the platform's drifting speed also escalates. At these elevated frequencies, the platform encounters more frequent propelling forces, explaining this trend. This emphasises the need to understand how different wave conditions can affect the platform's stability, especially in offshore environments where wave dynamics are constantly in flux.

5.6 Moored Tests

This section examines the effectiveness of WDP in enhancing the structural stability of offshore platforms, with a specific focus on its interaction with mooring systems. The study assesses the potential of WDP in reducing tension on mooring lines, thus contributing to overall stability.

5.6.1 Model Design

The proposed design integrates three adjustable foils between the semi-submersible columns, affixed to the interconnecting beams to ensure compatibility with the platform's structure and to generate self-propulsion, as shown in Fig.5.19. This setup used three NACA0030 foils, each with a chord length of 150 mm. The arrangement featured a frontal foil at a pitch angle of 30 degrees, complemented by two vertically positioned offset foils. This configuration is employed as the optimized foil configuration according to informed by prior findings.

5.6.2 Wave Tank and Mooring System Configuration

The wave tank experiments utilised a complex mooring arrangement, comprising five aluminium beams: two placed horizontally and three vertically. Pulleys were mounted on the vertical beams at a water depth of 0.59m to facilitate the smooth movement of the mooring lines and to adjust the initial tension of the springs located above the water surface. High-strength fishing line, 0.50 mm in diameter and capable of withstanding forces up to 210 N, was employed as the mooring line material. One end of each line

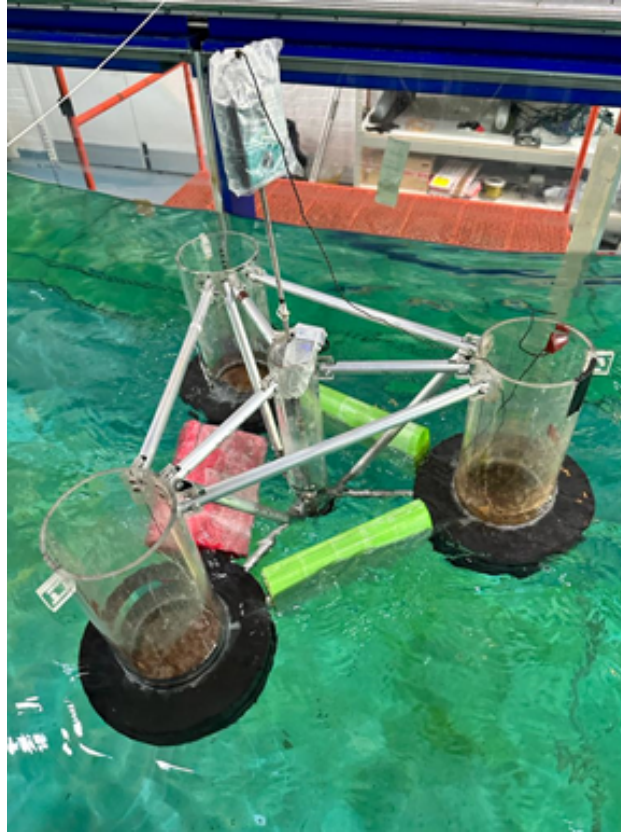


Figure 5.19: Photo of the foil-platform configuration used in the tests.

was anchored at the base of the platform's columns, while the other end, after being threaded through the pulleys, was connected to springs fixed at a consistent height above the water surface. This configuration ensured a stable orientation of the platform in calm water. The mooring system was strategically designed so that the lines were positioned at a 120° angle relative to each other, with each line forming an angle of 36.87° with the underwater horizontal plane. A Vernier force sensor was installed on one of the front vertical aluminium beams to facilitate accurate force measurements. Fig.5.20 and 5.21 present the schematic and a photograph of the wave tank with the no-foil platform in place, respectively. Fig.5.20 illustrates the detailed schematic of the wave tank and mooring system configuration, while Fig.5.21 provides a photographic view of the platform with the mooring system set up in the tank. The platform's setup and position in the tank remain consistent, whether or not the foils are installed.

The experiments were carried out with a regular head wave frequency of 0.65 Hz, varying the wave height to assess the adaptability of WDP under a range of wave conditions. The wave heights were methodically altered from 0.02 m to 0.20 m, in 0.02 m increments, to evaluate platform response and mooring line loads under varying environmental conditions, from mild to severe.

5.6.3 Discussion and result

Fig.5.22 displays the average peak-to-peak amplitude of mooring loads under various wave heights, comparing scenarios with and without the integration of foils. The peak-

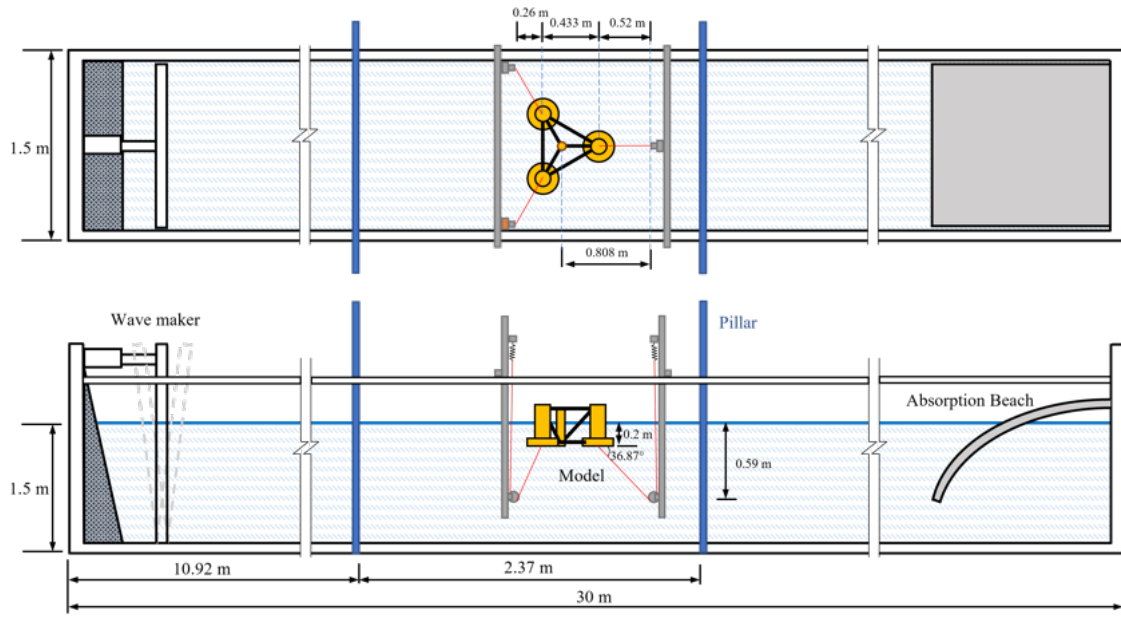


Figure 5.20: Schematic diagram of the wave tank setup with mooring system.

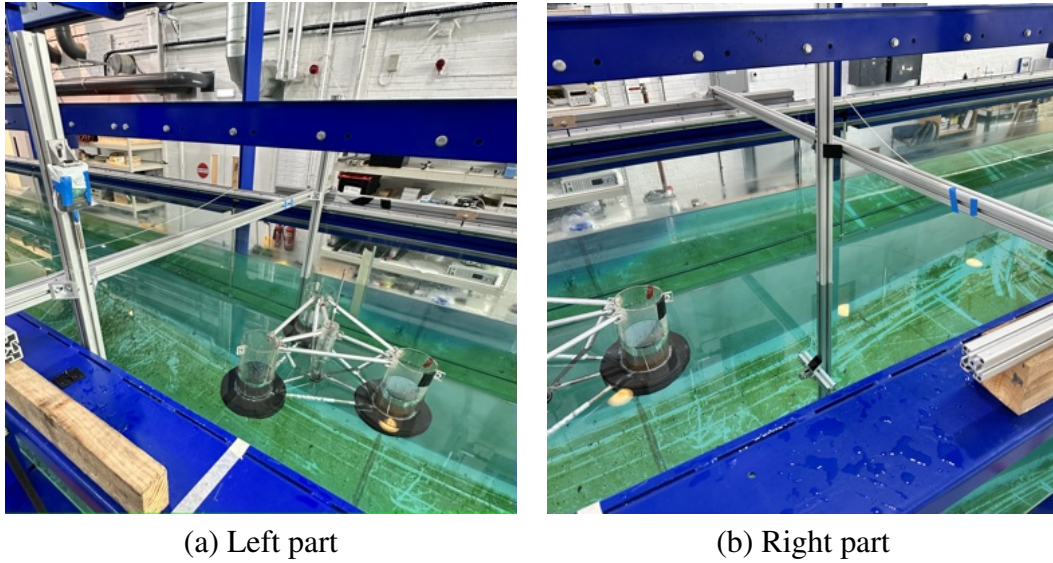


Figure 5.21: Photograph of the platform with mooring system placed in the wave tank.

to-peak tension amplitude—averaging the extremes in each motion cycle—serves as a pivotal indicator of platform stability and resilience. Consistent with theoretical expectations, an increase in mooring load correlates with escalating wave height, attributable to the augmented energy of larger waves. Moreover, the mooring loads recorded with the addition of foils are consistently lower than those without, demonstrating a significant and systematic reduction in loads when foils are employed. This discrepancy is particularly pronounced in the wave height range from 0.08 m to 0.16 m, where the difference in loads—referred to as the ‘gap’ in the figure—is more distinct. At a wave height of 0.10 m, the foils facilitate a remarkable 41.07% decrease in mooring tension. This pronounced reduction highlights the foils’ efficacy in stabilising the platform, particularly within this

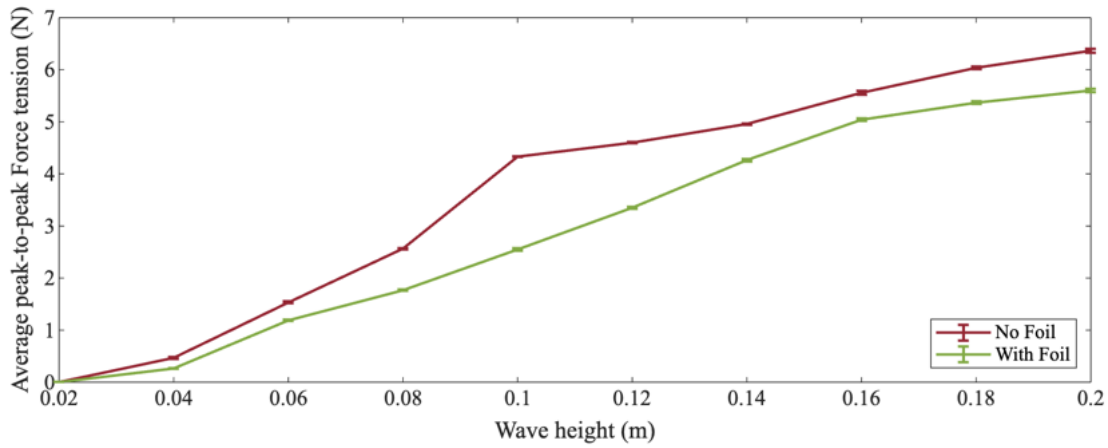


Figure 5.22: Mooring load under various wave heights: no foil vs with foil.

wave height range, providing critical insight into their operational performance and advantageous application conditions.

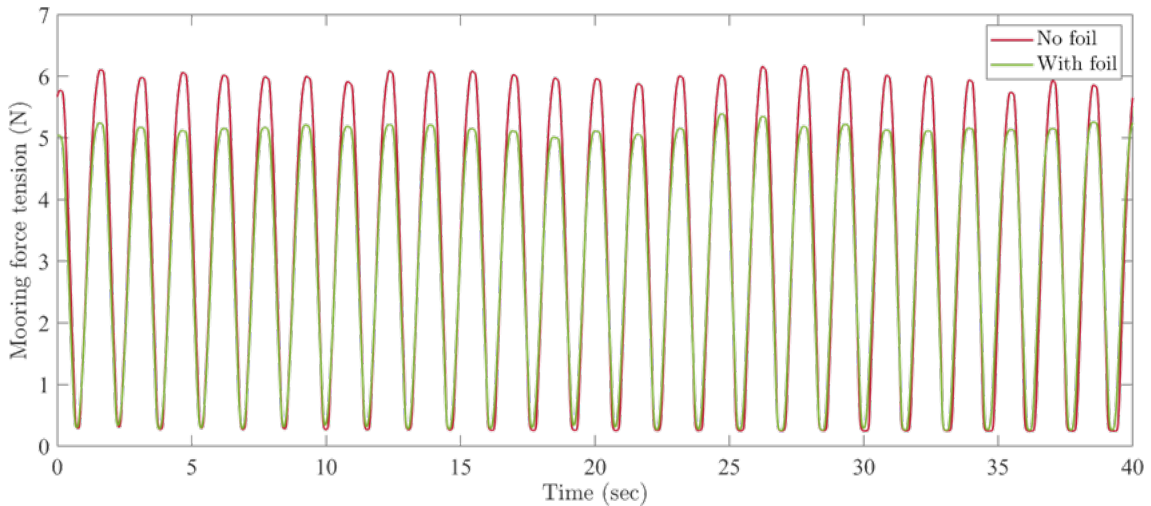


Figure 5.23: Comparison of mooring load fluctuations over time without foil (red) versus with foil (green) under a constant wave frequency of 0.65 Hz and a wave height of 0.15 m.

Fig.5.23 illustrates the temporal variations in mooring load under a wave frequency of 0.65 Hz and a wave height of 0.15 m. The comparison between scenarios with (green line) and without (red line) a foil reveals significant insights. The periodicity of the load oscillations, approximately 1.54 sec, is consistent with the wave period, which affirms the dominant role of wave motion in influencing the platform's dynamics. Additionally, the integration of the foil results in a marked 14.50% reduction in the amplitude of load oscillations, which underscores its effectiveness in dampening wave-induced forces. This attenuation of peak loads indicates that the maximum stresses on the mooring system are alleviated, potentially enhancing the system's maintenance profile and extending its operational lifespan.

5.7 Conclusion

This paper highlights a key advancement in offshore wind energy, focusing on the application of foil-induced Wave Devouring Propulsion (WDP) for semi-submersible Floating Offshore Wind Turbine (FOWT) platforms to improve stability. It investigated how foils affect drift speed, heave motion and mooring line tension in dynamic sea conditions, experimentally investigated on a free-floating/moored platform with various foil configurations. The study examined the number of foils, their orientation to waves, angles, and wave frequencies, leading to important insights for optimizing foil+platform designs.

Key findings include the superiority of using three foils over one or two, especially when foils face incoming waves with their leading edge. The study found that the position of holes for connecting foils to the platform had minimal impact, highlighting the importance of foil quantity and arrangement. Pitch angle on foils significantly improved performance, with vertical setups outperforming horizontal ones. A specific setup with two vertical and one frontal foil at 30° pitch angle showed the best performance, notably reducing heave motion when offset foils positioned vertically with a frontal pitch angle of 180°.

The optimal configuration—three foils with one frontal at 30° pitch angle and two verticals—demonstrated excellent resistance to wave-induced drift and a 41.07% reduction in mooring tension at a wave height of 0.10 m and frequency of 0.65 Hz, providing valuable insights for future designs.

Future research should explore flexible foils, active and passive elements, and responses under irregular wave conditions. These areas will enhance understanding of WDP adaptability and performance in marine environments.

5.8 Acknowledgement

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Chapter 6

Discussion and conclusion

6.1 Conclusions

This PhD thesis aimed at enhancing the stability of FOWT platforms using biomimetic-inspired WDP technology. Through a comprehensive literature review, this study confirmed the potential of WDP to improve the stability of FOWT platforms. It delved into the mechanics of WDP through numerical simulations to establish a scientific basis for its potential. Subsequently, the research applied WDP in a scaled-down water tank experiment, step by step, developing a design integrating WDP with semi-submersible platforms and validating the effectiveness of this design. Detailed conclusions are summarised with regard to the intended objectives of this study:

Objective I: (Chapter 2) To conduct a comprehensive literature review on the proposed innovative technology for FOWTs, summarising its potential advantages and foundational concepts.

This chapter delivers a thorough overview of Wave Devouring Propulsion (WDP), covering theoretical, experimental, and numerical studies, as well as its practical applications. WDP has captured the marine industry's interest due to its innovative approach to propulsion and stabilisation, particularly in challenging sea conditions. Experimental evidence suggests that foils mounted at the ship's bow enhance this effect. Numerical studies emphasise the significance of foil submergence, where shallower depths translate to improved energy conversion, albeit with a rise in wave resistance. The influence of the free surface effect is vital for simulation while head waves have been shown to augment WDP's propulsion and stability benefits. WDP's practicality is evidenced by its use in manned ships like the trans-Pacific journey of the Suntory Mermaid II and in USVs like the Wave Glider and AutoNaut. Despite these strides, the adoption of WDP technology for offshore platforms remains in its infancy, presenting an opportunity for substantial innovation in marine stability.

Objective II: (Chapter 3) To develop a simulation tool for Fluid-Structure Interaction (FSI) analyses, tailored to evaluate the dynamics of the innovative technology under various marine conditions.

This chapter introduces a numerical model designed to simulate the multi-phase flow interactions between multi-phase flows and rigid structures. The model applies a 'one-fluid' approach and incorporates a three-phase Level Set Method for precise depiction of fluid-structure dynamics. It was benchmarked against simple harmonic motion cases and

validated through physical experiments, confirming its ability to capture the complexities of FSI under various conditions. This model provides a solid numerical foundation for future mechanistic studies.

Objective III: (Chapter 4) To investigate the mechanisms of the proposed technology, focusing on the dynamics of sub-structures (e.g., foils) within a viscous numerical wave tank, to understand their interaction with wave forces.

This chapter utilised CFD to explore the influence of wave-induced thrust on hydrofoils, particularly examining the role of pitch stiffness. Through a rigorous validation process against established benchmarks, the research confirmed the accuracy of its numerical approach in studying thrust generation mechanisms under various hydrodynamic conditions. Key findings revealed the transition between alternating vortex streets (RvK and vK) and their impact on thrust and drag, underscoring their significance in hydrofoil propulsion in wave environments. Further investigations into the hydrofoil's performance in regular waves, with pitch stiffness values up to 12900 N·m/rad and wavelengths up to 7.70 m, demonstrated that hydrofoil efficiency peaks at a 3.75 m wavelength. The study identified a nonlinear response to pitch stiffness, with an optimal point at 129.24 N·m/rad, beyond which increased stiffness reduces thrust efficiency. These insights offer a foundation for optimizing hydrofoil design and operation in wave-dominant conditions, suggesting avenues for future research and application enhancements.

Objective IV: (Chapter 5) To assess the practical applicability of incorporating the innovative sub-structure for FOWT stabilisation, with an aim to provide a feasibility study and optimization strategies.

This chapter advances offshore wind energy by exploring foil-induced Wave Devouring Propulsion (WDP) on semi-submersible Floating Offshore Wind Turbine (FOWT) platforms, focusing on stability improvements. Through rigorous experimentation on both free-floating and moored platforms with varied foil configurations, it reveals how foil quantity, orientation, pitch angles, and wave frequencies impact drift speed, heave motion, and mooring line tension under dynamic sea conditions. Notably, configurations with three foils, especially those oriented with their leading edge facing incoming waves, significantly outperformed others. The research underscores the minimal influence of foil connection points while emphasizing the critical role of foil arrangement and pitch angles, where vertical orientations excelled. The optimal setup, featuring two vertical and one frontal foil at a 30° pitch angle, markedly reduced heave motion and mooring tension by 41.7% at specific wave conditions, offering practical guidelines for enhancing FOWT platform stability. This study represents a promising avenue for future research and development in marine propulsion systems, marking the beginning of future applications of WDP in FOWT platforms.

6.2 Contribution to knowledge

The intellectual contributions of this PhD research to knowledge in the field are articulated through the findings of each chapter, as listed below:

This study provides a comprehensive overview of research on wave-induced flapping foils, outlining the evolution of research and applications in this field. It summarises the geometric configurations and operational parameters critical to flapping foil effectiveness. Furthermore, it identifies and proposes potential applications for WDP and outlines

directions for future research in the area.

A novel three-phase numerical model is introduced using Level Sets to capture multiple interfaces. The model incorporates spring forces in the Fluid-Structure Interaction algorithm within the one-fluid formulation. This model also pioneers the implementation of hybrid wave-current boundary conditions to better mimic marine environments. This PhD research provides a numerical platform for multi-phase CFD simulations, establishing a reliable and robust simulation framework for fluid-structure interactions in marine environments.

The first CFD study on the WDP system is achieved focused on passive foil movement induced by wave interactions. The study investigates the mechanism of thrust generation through flapping foil dynamics, providing clarity on the transition from drag to thrust. Moreover, by identifying the optimal pitch stiffness required for peak thrust generation, this research provides a robust quantitative foundation for the design and operational parameters of foils.

This study makes a novel contribution to marine engineering by exploring the application of WDP technology on semi-submersible platforms. It presents the first experimental validation of integrating WDP to enhance offshore platform stability, marking a significant advancement in the practical application of this technology. Furthermore, the study develops comprehensive design guidelines for foil implementation on floating platforms, concentrating on optimizing the quantity and pitch angle to maximize efficiency. Most notably, it proposes a validated ‘foil+platform’ design that demonstrates a remarkable reduction in mooring tension by 41.07%, establishing new benchmarks for efficiency in marine engineering.

6.3 Future research

This research represents a substantial advancement in the application of Wave Devouring Propulsion (WDP) technology for Floating Offshore Wind Turbines (FOWTs), setting the stage for future investigations and improvements. The study opens up several key areas for further academic exploration and technological development, as listed below:

1. Hydrodynamics and System Interaction:

- Explore reversed von Kármán vortex generation for enhanced propulsion.
- Analyse foil-platform-wave interactions through numerical and experimental methods.
- Conduct large-scale tests, full-scale tests, on-site tests and hydrodynamic modelling to understand platform motion impacts.

2. Comprehensive Design Optimization:

- Optimize foil number, placement, size, and submergence for stability and efficiency.
- Investigate active and passive foil control mechanisms for dynamic environments.

- Explore material flexibility and various foil models (e.g., NACA 0012, NACA 0015) to enhance performance.

3. Economic Assessment and Operational Feasibility:

- Perform viability studies and quantify benefits, including maintenance cost savings from reduced mooring loads.

4. Expanding Applications and Integration:

- Design WDP solutions for diverse platforms, and integrate with other station-keeping techniques, e.g. DP, for improved performance in harsh conditions.
- Explore WDP's potential in supporting floating PV systems, large vessels, and O&M scenarios for marine operations.

By focusing on these strategic areas, the objective is to advance the Technology Readiness Level (TRL) of WDP with offshore platforms from TRL 3 to TRL 6. This advancement involves transitioning from conceptual validation and small-scale laboratory tests (current level) to the creation of larger-scale prototypes for offshore environments, moving towards commercial application. Such progress requires not only technological innovations but also a holistic approach that encompasses the operational, environmental, and economic aspects of offshore platform integration.

Enhancing WDP technology for offshore platforms holds the promise of significantly improving the sustainability and efficiency of maritime operations. It positions itself as a cornerstone in the maritime sector's efforts towards global net zero emissions. By innovatively adapting WDP systems for offshore platforms, focusing on strategic design integration and extensive application, the potential to revolutionize marine propulsion and stabilisation is substantial. Such advancements are key to the future progress of the maritime and offshore industries, underscoring the vital importance of WDP technology in driving sustainable offshore development.

Appendix A

Ethical Approval Letter

An ethical approval was obtained for all the data collected and used as part of the Thesis via the Cranfield University Research Ethics System (CURES). The letters of approval and the reference numbers for this PhD research project are shown below in Fig.A.1.



26 July 2024

Dear Miss Xing ,

Reference: CURES/23177/2024

Project ID: 13256

Title: Enhancing Stationkeeping and Motion Reduction of Floating Offshore Wind Turbines using Wave Devouring Propulsion Technology

Thank you for your application to the Cranfield University Research Ethics System (CURES).

We are pleased to inform you your project title change request, reference CURES/23177/2024 has been approved.

If you have any queries, please contact CURES Support.

Regards,

CURES Team

Figure A.1: Letter of approval for the project from the CURES system with reference number CURES/23177/2024.