



Title	Nonlinear Hydroelastic/Hydroelastoplastic Responses of Ships and Floating Structures by Two-way CFD-FEA Technique with Enhanced Performance
Author(s)	Xie, Binyang
Citation	大阪大学, 2026, 博士論文
Version Type	VoR
URL	https://doi.org/10.18910/105482
rights	
Note	

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The University of Osaka

博士学位論文

Nonlinear Hydroelastic/Hydroelastoplastic
Responses of Ships and Floating Structures
by Two-way CFD-FEA Technique with
Enhanced Performance

(高性能化 CFD-FEA 双方向連成手法による船
舶および浮体構造物の非線形流力弾性/弾塑性
応答に関する研究)

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2025 年 12 月

大阪大学大学院工学研究科

Doctoral Dissertation

Nonlinear Hydroelastic/Hydroelastoplastic
Responses of Ships and Floating Structures
by Two-way CFD-FEA Technique with
Enhanced Performance

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December 2025

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Acknowledgement

First and foremost, I would like to express my deepest gratitude to my supervisor, **Prof. Iijima Kazuhiro**. His inspiring guidance, patience, and encouragement have been the pillars of my support throughout my time at **the University of Osaka** .

I would also like to thank my best friend during my stay in Osaka, **Dr. Sumit Kumar Pal**, for his kind help and support in both my studies and daily life.

I extend my special thanks to **Prof. Osawa Naoki**, who reviewed my doctoral thesis with **great meticulousness and provided invaluable comments**.

I also wish to acknowledge all the **staff** and **students** in Laboratory 4 for their help. I want to express my sincere gratitude to **Assoc. Prof. Tatsumi Akira**. I highly appreciate the consistent assistance from the laboratory secretary, **Ms. Chayama**.

A special acknowledgement goes to the members of the **Slamming and Sloshing Group**: Mr. Yamamoto Masafumi, Ms. Sawada Nanako, Mr. Nishimura Yuta, Mr. Li Gyoshu, Mr. Yamada Yusuke, and Mr. Kitadai Hidenori. It has been a true honor to study and work alongside you.

I would like to specially acknowledge Prof. Dr. He Guanghua, Dr. Jin Penglin and Dr. Luan Zhengxiao from Harbin Institute of Technology, for the help and support of my numerical simulations.

I would also like to acknowledge all the professors in my department who provided instruction during my coursework. Without the knowledge I acquired from them, the completion of this thesis would have been impossible.

Financial support from the Japanese Government (MONBUKAGAKUSHO: **MEXT Scholarship**) is gratefully acknowledged. I also appreciate the technical support provided by **GYT Data Technology Co., Ltd.**

Last but not least, I would like to thank my parents and my friends for their unwavering support.

Abstract

This thesis develops and systematically validates high-fidelity, strongly coupled Computational Fluid Dynamics-Finite Element Analysis (CFD-FEA) frameworks for predicting linear and nonlinear hydroelastic and hydroelastoplastic responses of marine structures, including springing, slamming, and hydroelastic wave propagation. Particular emphasis is placed on improving the efficiency and robustness of fluid – structure coupling. The background, motivation, and problem statement of the problem are discussed in Chapter 1.

In Chapter 2, the methodology underlying the proposed coupled CFD-FEA frameworks is presented. Within a strong partitioned interaction – resolution scheme, the fluid and structural fields are iteratively coupled until interface convergence is achieved, yielding solutions equivalent to those obtained from monolithic formulations. Techniques to accelerate convergence and enhance numerical stability, such as a predictor-based coupling strategy, are introduced, which constitutes the core methodological contribution of the thesis.

In Chapter 3, a standard coupled CFD-FEA framework is first applied to the hydroelastic response of a segmented Barge(A) model and a segmented KVLCC2 model in regular waves. Towing-tank experiments are conducted over a range of wave conditions, and the response amplitude operators (RAOs) for motions obtained from the CFD–FEA simulations are shown to agree well with measurements. The results confirm that, in the investigated conditions, the ship structures respond predominantly in the 2-node vertical bending mode.

In Chapter 4, higher-order springing, in which the encounter frequency or its higher harmonics excite structural natural frequencies, is then investigated using a segmented Barge(B) model using the proposed coupled CFD-FEA framework with a predictor. Combined tank tests and two-way coupled simulations demonstrate that the

predictor-enhanced framework accurately reproduces the measured heave, pitch, and flexible vertical bending responses, and resolves springing vibrations up to the 14th harmonic of the wave load— a phenomenon rarely documented in the literature. This provides new physical insight into the mechanisms and spectral characteristics of high-order springing.

In Chapter 5, slamming-induced hydroelastic and hydroelastoplastic responses are studied for a wedge section model based on a standard coupled CFD-FEA framework. Comparisons between elastic and elastoplastic simulations reveal that material plasticity can significantly modify the transient hydrodynamic and structural response, indicating that hydro-elastoplastic effects should be carefully considered at the design stage.

In Chapter 6, the interaction between nonlinear waves and a highly flexible VLFS-like plate is investigated using a coupled CFD-FEA framework with a Laplace matrix, which stabilizes the coupling at the Fluid-Structure Interaction(FSI) interface when the added-mass effect is significant. The simulations reproduce key nonlinear features of the wave profile, including crest flattening, trough sharpening, and subsequent profile inversion during propagation. The degree of nonlinearity, quantified by the ratio of positive to negative peak amplitudes, is in good agreement with analytical predictions. These results demonstrate that the CFD–FEA framework with Laplace matrix provides a unified and reliable tool for studying a wide range of hydroelastic phenomena in large flexible floating structures, and they form the basis for recommendations on future developments in numerical modelling and experimental validation.

Finally, the main findings are summarized, and suggestions for future work are presented in Chapter 7.

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Chapter 1

Introduction

1.1 Background

Hydroelasticity, arising from the interaction between free-surface waves and deformable marine structures such as ships and very large floating structures (VLFS), plays a critical role in the design and safety assessment of modern marine structures. The increase in ship size has made it challenging to evaluate the safety of ship structures using traditional analysis methods, particularly when ships encounter complex sea conditions. The enlargement of a ship's size can result in a decrease in its natural frequency, making it more vulnerable to whipping and springing [1,2]. When ships with a low natural frequency interact with incoming waves, resonance is more likely to occur, which can cause local deformation or even structural failure. The hydroelastic response at a global level can apparently affect both the extreme structural response and the fatigue life of certain structural details [3]. For instance, the maritime accident involving the MOL Comfort (see Figure 1-1) has two conceivable causes [4]: The structural response of the ship increases as a result of the combination of global and local loads, as well as hydroelastic vibration.



Figure 1-1 Condition of the MOL-Comfort at the time of the accident (photo by Mitsui O.S.K. Lines, Ltd)

Therefore, when predicting the safety and motion response of large marine and floating structures, it is essential to consider Fluid-Structure Interaction (FSI) and the elasticity of the structure, also known as hydroelasticity. The analysis considering hydroelasticity offers several advantages [5]:

(1) Compared with traditional seakeeping analysis, information from the fluid field can be made the best of. Thus, when predicting rigid body motion, the results of deformation, strain, stress, bending moment, and shear force of the structure can be provided.

(2) A precise phase relationship between the steady wave-induced load and the transient slamming load of the ship can be given when sailing in waves.

(3) Ship structure response from global and local hydrodynamic loads can be integrated and investigated naturally.

(4) When sailing in waves, the structural strength, dynamic stability, and fatigue

performance of the ship hull can be unified and studied together. This can lay a foundation for direct ship-design methods.

In the 1970th, Bishop et al. developed a two-dimensional linear hydroelasticity theory [6,7] to address the interaction problem between elastic beams and fluids. This theory laid the foundation for the branch of hydroelasticity. Later, the emergence and popularisation of high-performance computers led to the development of three-dimensional hydrodynamic methods for solving hydroelasticity problems. Wu [8] proposed a three-dimensional linear hydroelasticity theory that considers generalized fluid-structure coupling boundary conditions. The theory can estimate the motion response of any elastic body to waves. Price [9] introduced an approximate correction to Wu's theory for fluid viscous resistance, enabling the analysis of its influence. Wu's theory was validated through a series of examples and further improved by Bishop et al. [10]. However, linear hydroelasticity theory assumes small disturbances of the fluid and minimal displacement and deformation of the structure. This assumption is not always applicable. Therefore, nonlinear hydroelasticity theory was further developed [11–14]. Consideration has been given to factors such as the instantaneous change of structure immersion depth, anharmonic oscillation, and the slamming effect of waves. However, analytical solutions alone may not be sufficient for complex hydroelastic problems. There are two primary methods for the experimental study of hydroelasticity: real-ship tests and model tests. Results from real-ship tests are precise. However, conducting such tests is generally expensive and challenging. Due to the scale effect, there is a discrepancy between the results obtained from the model test and the actual results. The model test can meet the basic requirements for general design and construction and is therefore widely used. Riggs et al. [15] proposed two simple ship models called Barge and Wigley, which have become benchmarks for three-dimensional hydroelasticity studies. Senjanović and Malenica [16] conducted a tank test on a segmented Barge ship to study the hydroelasticity under regular waves. Remy

et al. [17] investigated the elastic response of a Barge ship to diagonal waves, based on Malenica's experiment. Fu et al. [18] investigated the general characteristics of hydroelastic response in flexible floating interconnected structures, including their displacement and bending moments under various conditions. Iijima et al. [19] and Zhu et al. [20,21] designed scaled ship models based on real ships. The study focused on the wave frequency response, as well as the bending and torsional vibrations, of the segmented ship model in both regular and irregular waves. Chen et al. [22] investigated the vibration characteristics of a segmented ship model at varying wave angles. Fonseca and Guedes Soares [23] conducted a model test on an S-175 container ship. The results reveal the non-linear response of the structure. A self-propelled test of a segmented ship was conducted by Jiao et al. [24]. Kim et al. [25] investigated fatigue damage in a 9400 Transmission Extension Unit (TEU) container ship based on a real ship test. Yang et al. [26] conducted a large-scale combined beam model experiment on a 20,000TEU container ship. Jiao et al. [27] investigated hydroelastic responses through a model experiment in harsh irregular seaways.

Numerical simulation has become the most effective and economical solution for studying hydroelasticity in recent years. Guo and Zou [28] investigated ship maneuvering with four degrees of freedom (DOF) in still water using the ONR roll ship model. Iijima et al. [29] proposed a consistent structural analysis procedure to estimate the global and local load effects considering symmetric and antisymmetric hydroelastic vibrations in high waves. Kim et al. [30] investigated torsion, bending and warping distortion of a one-dimensional beam using a hybrid Boundary Element Method (BEM)-FEM scheme. He and Kashiwagi [31] investigated the hydroelastic response of an elastic vertical plate under solitary waves using a BEM-FEM coupled method. Takami et al. [32,33] developed an in-house code and conducted one-way and two-way CFD-FEA coupled simulations for hydroelastic response using an Inverse Distance Weighting (IDW) mapping algorithm [34]. This in-house code can couple any CFD

and FEA code, providing a new option for investigating hydroelasticity. Jiao et al. [35] and Huang et al. [36] estimated ship wave loads and hydroelastic responses using fully coupled computational fluid dynamics (CFD) and finite element analysis (FEA) solvers in a two-way manner, with Star-CCM+ and Abaqus, respectively. Pal and Iijima [37] attempted to investigate higher-order springing phenomenon using direct coupling of CFD-FEM analysis for a barge-type structure. Lu et al. [38] investigated the hull girder collapse behavior in the case of a 5250 TEU containership using a coupling between CFD and FEA. [39] analyzed the structural collapse characteristics of a three-cabin structure using the CFD-FEA coupled method. A fully coupled CFD-Discrete Module Beam (DMB) method is also developed and used for the hydroelasticity of ships [40] and Very Large Floating Structures (VLFS) [41] recently.

1.2 Springing

Springing occurs when the encounter frequency directly excites the natural frequency of the structure, or the excitation frequency could be a different integer multiple of the encounter frequency [1,42,43]. These are known, respectively, as linear and non-linear springing. Although the difference in excitation sources between linear and nonlinear springing is not clearly discussed in the literature, the basic understanding of hydroelasticity theory suggests it is an unsteady wave pressure field on the hull [44]. In the presence of a simple harmonic wave load, with an excitation frequency much higher than the natural frequency of the structure, it is not expected to have any resonance effect. However, when it comes to the complex fluid-structure interaction of floating bodies, the incoming waves are always a mixture of different wave components, namely, radiated or diffracted waves generated by the same body or other floating objects. Thus, different components of the waves indeed contribute to the springing phenomenon. Most traditional potential theory-based rigid body hydrodynamic solvers perform best for linear problems with a mean wetted surface. Later, different levels of non-linearities are included separately to better represent the problem of interest. While

it is accurate for the general seakeeping problem, when it comes to highly nonlinear phenomena, it is challenging to formulate and simulate each phenomenon separately.

Springing responses in the linear frequency domain (2D strip theory and 2D Timoshenko beam theory) as well as nonlinear time domain analyses (3D hydro and 2D beam) are compared with model test results by Lee et al. [45] as a part of the WILS project. From the comparison of the two-dimensional hydro-elastic method and model test results, it can be observed that there is a high bending moment response in the HF region for both cases; however, significant differences in the magnitude of the vibrations are also noted. Wave-induced vertical bending moment time series amidships are compared between the nonlinear hydro-elastic method and model test for linear, second-order, and third-order springing. It was reported that there were dissimilarities between the nonlinear code and the model test. Vidic-Perunovic [44] used modified second-order strip theory to account for the additional second-order springing excitation coming from interacting directional waves. Kim et al. [46] used the time domain Rankine panel method -FEM based hydro-elastic solver to study the springing response of a flexible floating barge with a zero-forward speed under head and oblique waves. Senjanović et al. [47] performed vibration analysis of a 7800 TEU containership using a coupled 1D FEM and 3D frequency domain potential theory-based solver. Their method pointed out that the transfer functions for the hull sectional forces in the case of resonant vibration were much higher than the resonant ship motion. Shao and Faltinsen [48] presented a time-domain Higher Order Boundary Element Method (HOBEM) with a formulation of the boundary value problem in a body-fixed coordinate system for the numerical study of the second-order springing on a modified Wigley hull. Kim and Park [49] analyzed the second-order effects of the springing response of a very large ore carrier using higher-order spectral analysis. The vertical bending moment response spectra were quite prominent, although the power of wave elevation at the corresponding frequency regime is negligibly small, which proved the significance of

the second-order effect. Wang et al. [50] addressed the wave-induced vibrations in an ultra-large ore carrier under fully loaded and ballast conditions and compared them with 3D frequency domain hydro-elastic theory. Increasing the draft to reduce the springing responses was also discussed. Tilander et al. [51] used a potential flow theory-based frequency domain hydro-elastic seakeeping model for springing analysis of a passenger ship in waves. Riesner and El Moctar [52] used the Cummins approach together with the beam element (for strong bending and torsion) to predict the higher-order springing of a post-Panamax containership. Non-linearities like Froude-Krylov and hydrostatic forces are added separately. Duan et al. [53] used the time-domain Taylor Expansion Boundary Element Method (TEBEM) for springing responses of a containership with forward speed in regular waves. Despite these studies, a higher-fidelity method or experiment is necessary in order to understand and reproduce the springing phenomenon, regardless of the limitations of the numerical solver [54].

Several example cases of springing vibrations in different full-scale models were shown by van Gunsteren [55]. After a few years, Slocum and Troesch [56] reported a detailed study on the importance of non-linear springing excitation using experiments. They concluded that the non-linear excitations are dependent on the ship speed, encounter frequency, and wave height. Storhaug and Moan [57] also conducted the model test and compared it with the ocean-going model. Hirdaris et al. [58] also correlated the two-dimensional hydroelastic results with the full-scale measurement data available for the Great Lakes bulk carrier. The inclusion of springing and whipping generated double-peak response spectra in good agreement with measurements; however, the sea state information was not sufficient to carry out additional correlation of existing data. Derbanne et al. [59] used coupled 3D FEM- 3D BEM in the frequency and time domains in order to account for springing and whipping, which was validated using model test results. Hong and Wan Kim [60] conducted a series of model tests on a 10,000 TEU containership with an H- and U-shaped backbone, focusing on higher-

order springing and whipping. Higher-order vertical bending springing was observed up to the 5th order, and for torsion springing, it was up to the 4th order. Non-linear vertical bending springing was also investigated by the MOERI (Maritime and Ocean Engineering Research Institute) report in 2009 for the model test [61]. Yu and Hong [62] performed segmented model tests and numerical analysis based on a non-linear seakeeping code, considering nonlinear hydrostatic and Froud-Krylov force with quadratic pressure terms, for investigating wave-induced extreme and springing loads. In this paper, the HF vibrations were observed during a model test of a segmented blunt barge model. Later, the authors attempted to validate those responses.

The CFD solver was found to be useful to capture such nonlinear phenomena in hydroelasticity by combining with an FEA solver. Seng et al. [63] developed a hydroelastic model that combines OpenFOAM and a flexible body motion solver, where the body's linear elastic deformation is described by modal superposition. Oberhagemann [64] presented a coupled Reynolds Averaged Navier-Stokes (RANS) equation and beam FE model to account for complex resonant excitation of the hull girder in waves, including impulsive phenomena. Lakshminarayanan and Temarel [65] employed commercial STAR-CCM+ and ABAQUS-based co-simulation to achieve RANS (CFD)-FEM coupling, thereby enhancing the understanding of wave-induced loads on a containership. Using the same procedure, Jiao et al. [35] investigated the hydroelastic responses of the S175 hull; however, wave-induced vibrations contained both springing and whipping terms, and the simulation results were not strongly supported by experiments.

Although the CFD-FEM coupled technique has been widely applied in hydroelasticity research with the growing ease of computational resources, it is important to note that the field of hydroelasticity encompasses a wide range of problems (e.g., slamming induced whipping, springing, etc.), as highlighted in Bishop and Price

[7], and therefore needs specific attention to each part separately for better understanding of the phenomenon. While previous studies based on the CFD-FEA coupled technique have focused on hydroelastic responses [32,35,36,40,41,66,67] such as a combination of slamming and whipping, the research in Chapter 4 addresses only springing, a specific type of hydroelastic phenomenon that has not been thoroughly investigated through CFD-FEM coupling.

1.3 Slamming

Slamming involves a powerful impact between a ship's hull and waves. The hydrodynamic force caused by slamming may lead to springing or whipping of the ship [68]. For container ships, the ongoing drive to increase container carrying capacity has led to a rapid increase in ship size, making them susceptible to whipping and springing [69,70]. Predicting slamming impact pressures is crucial in the structural design of marine structures, particularly high-speed craft. The time interval of water entry is extremely short. Thus, large impulsive pressure loads may cause local deformation or failure of the ship structure. In a rough sea, the slamming cannot be avoided even at the stern or midship of a ship. Slamming impacts occurring in the bow flare region of large container ships can directly lead to plastic deformation or fatigue accumulation in local structures such as stiffened panels, making accurate load evaluation and response understanding essential for both design and operation. Recent reviews have highlighted the importance of accounting for the interaction between impact pressure and structural dynamic characteristics (hydroelasticity) in evaluations [71]. Furthermore, practical approaches for obtaining design loads have also been organized, and frameworks for estimating extremes and risks are being systematized [72].

In the 1930s, Von Karman [73] and Wagner [74] developed theoretical formulas for the maximum pressures at the bottom of the seaplane float during landing, which

agreed well with the experimental results. Zhao et al. [75] developed two different theoretical methods for predicting slamming loads on two-dimensional sections, which were validated by comparing them with experimental results from drop tests of ship cross-sections. Tveitnes et al. [76] developed a test rig that drives a wedge section, with end plates and down guides, to enter the water vertically at nearly constant velocity, thereby measuring the entry force. Wang et al. [77] experimentally and numerically investigated the water entry of a free-fall wedge with a focus on the evolution of the pressure on the impact sides and the top side. Luo [78] presented an uncoupled method that combines the Wagner theory and the finite element method to analyze the slamming problem for a 3D structure. Mohammad et al. [79] addressed the water entry experiments of compliant slender bodies. Aquelet et al. [80] investigated the water entry problem using the Euler–Lagrange coupling algorithm. Zhu et al. [81] investigated the fully nonlinear free-surface deformations of initially calm water resulting from the entry and exit of a horizontal circular cylinder, considering both forced and free vertical motions, using the CIP (constrained interpolation profile) method. Wang et al. [82] carried out a free-fall slamming experiment on a steel model of a hull section. The slamming pressure and structural elastic response stress on both sides of the steel model were measured.

The study of slamming-induced hydroelastic/hydroelastoplastic responses has been conducted in recent years [83]. Hosseinzadeh et al. [84–87] conducted both experiments and numerical simulations (ALE, CFD-FEA), and the effects of water impact velocity, deadrise angle, mass of the wedge, and bending stiffness on the slamming pressures and structural responses are discussed in detail. Park et al. [88] investigated the plastic deformation of a flat plate under slamming impact by coupled Eulerian–Lagrangian (CEL) method. Yu et al. [89,90] proposed an analysis model coupling the hydrodynamic forces and the plastic response of rectangular beams and one-way stiffened panels. The proposed model has been further validated by both

experiment and numerical simulations.

1.4 Nonlinear Hydroelastic Waves in Deep Water

Hydroelasticity, also commonly observed in the response of VLFS, floating solar and ice sheets, occurs when waves interact with a floating, deformable body, inducing deformations within the material's elastic limit. Hydroelasticity has long been a primary focus of research related to VLFS [91,92]. Linear hydroelastic behavior has been investigated initially, where the deflection and wave were assumed to be small, and only linear terms were considered. In the early days of research, researchers delved themselves into reducing the computational burdens that arise due to large-scale models. Based on a domain decomposition method, Ohmatsu [93] described the hydroelastic behavior of a pontoon-type VLFS. To enhance computational efficiency, Kashiwagi [94] integrated a B-spline approach with a pressure-distribution technique and an eigenfunction-expansion method. A comparative study of three codes for the linear hydroelastic response of a pontoon-type VLFS was conducted by Riggs [95]. While two of the codes adopted potential flow theory, the third one was based on the Green–Naghdi theory for long waves. When applied to the pontoon-type VLFS under a broad spectrum of wave periods and incident angles, all three codes predicted similar results.

Meanwhile, the nonlinear behavior of VLFS is also a concern. Xia and Shen [96] presented a shallow water nonlinear analysis of the interaction between a water wave and a floating ice cover in river channels by applying the Korteweg-De Vries (KdV) type equation. A weakly nonlinear approach for VLFS analysis was presented by Belibassakis and Athanassoulis [97]. The nonlinear influence of variable bathymetric regions was discussed. Vanden-Broeck and Părău [98] developed a theoretical model for nonlinear hydroelastic waves beneath ice sheets, treating the ice sheet as an elastic, flexible plate and neglecting inertial effects. Iijima and Ma [99] proposed a second-

order theoretical framework, analogous to Stokes wave theory, to derive analytical solutions for periodic wave propagation along a thin elastic plate. Pambela et al. [100] further compared the analytical solutions with experimental results of deep-water nonlinear wave propagating beneath a thin elastic plate and discussed the instability frequency. More recently, by using a time-domain nonlinear potential flow framework, Liang et al. [101] investigated the interaction between nonlinear waves and a three-dimensional floating elastic plate, incorporating axial-force effects into the plate's structural response. The resulting boundary-value problem was then solved using the higher-order boundary element method (HOBEM). A general numerical model that can replace tank tests needs to be developed. Among the previous works, most of the analytical or potential-flow frameworks could not consider varying types of relevant nonlinearities at the same time, including free-surface, sea bottom interaction, membrane (axial load-vertical deflection coupling), large deflection, material nonlinearity, etc. A high-fidelity and broadly applicable framework is therefore desirable.

As general-purpose framework, the coupled approaches have been prevalent for the investigation of hydroelastic problems recently with the advances in computational capabilities. Ma et al. [102] developed a coupled Smoothed Particle Hydrodynamics (SPH) and Finite Element Method (FEM) model, which can capture the hydroelastic behavior of a thin elastic plate floating on the water surface. It was observed that the second-order wave component reverses at the instability frequency. Huang et al. [36] investigated the global motion and vertical bending moment (VBM) of a single module VLFS under freak wave conditions. A commercial co-simulation suite was adopted to realize the coupling between the CFD software Star-CCM+ and the FEA software Abaqus. The presence of a freak wave causes a multi-fold amplification in the global motion of the VLFS, leading to a substantial rise in the instantaneous maximum VBM. Wei et al. [41] developed a time-domain hydroelastic-moored model, which integrates

a quasi-static mooring module with a fully coupled CFD-Discrete Module-Beam (DMB) approach, to evaluate the coupled effects between the mooring system and structural hydroelasticity by varying key design parameters like structural and mooring stiffness. While coupled frameworks have demonstrated potential for hydroelastic problems, questions remain regarding the fidelity of their predictions; for VLFS, validation for these frameworks is still limited. Meanwhile, for highly flexible FSI problems, it is always challenging to reach a good convergence at the FSI interface for coupled approaches [103].

1.5 Objectives of the thesis

This thesis develops and systematically validates high-fidelity, strongly coupled CFD-FEA frameworks for predicting linear and nonlinear hydroelastic and hydroelastoplastic responses of marine structures, including springing, slamming, and hydroelastic wave propagation. Particular emphasis is placed on improving the efficiency and robustness of fluid – structure coupling. The objectives of the thesis are summarised as follows:

- Development and adoption of Enhanced Two-Way Coupled Frameworks

Two distinct coupling frameworks are adopted to address different numerical challenges in FSI. Framework A incorporates an interface Laplace matrix and a stabilization matrix. This approach effectively stabilized the coupling at the FSI interface, particularly when the added-mass effect was significant, without requiring modification to the solvers for predictors. To accelerate convergence, a rigid body motion (RBM)-deformation decomposed predictor is proposed and implemented in the framework B. This method significantly reduced the number of sub-iterations required for convergence in partitioned coupling schemes, enhancing computational efficiency for problems where RBM contributes significantly to the total motion.

- Hydroelastic Behavior of Segmented Ships

Using Framework A, the hydroelastic responses of segmented Barge (A) and KVLCC2 models will be investigated in regular waves.

- Higher-Order Springing Analysis

The springing phenomenon will be studied using the predictor-enhanced Framework B on a segmented Barge (B) model.

- Slamming-Induced Hydroelastic and Hydroelastoplastic Responses

Framework A will be applied to predict slamming loads on a wedge section. The framework will be validated against experimental data within the elastic range. Furthermore, the study will extend to full-scale models to investigate hydroelastoplastic responses.

- Nonlinear Hydroelastic Waves in Deep Water

The propagation of nonlinear waves along a highly flexible, thin elastic plate will be investigated using Framework A with the Laplace matrix scheme.

Chapter 2

Methodology

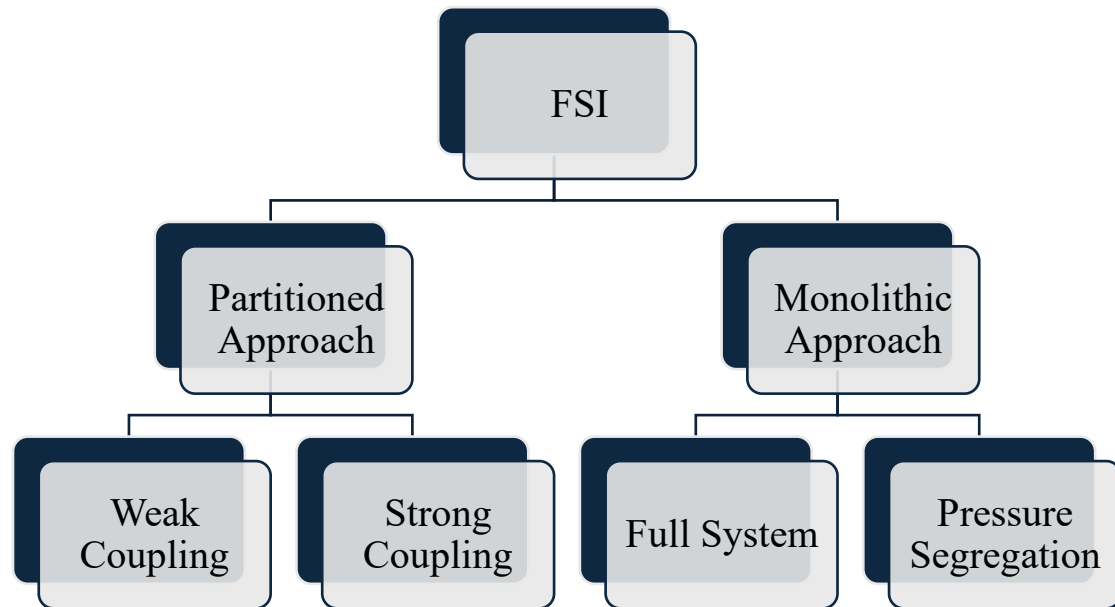


Figure 2-1 Categories of FSI approach.

In this chapter, the methodology underlying the proposed coupled CFD-FEA frameworks is presented. Typically, methods for solving FSI problems could be classified into two categories: the monolithic approach [104] and the partitioned approach [105,106] (see Figure 2-1). Within a strong partitioned interaction–resolution scheme, the fluid and structural fields are iteratively coupled until interface convergence is achieved, yielding solutions equivalent to those obtained from monolithic formulations. Techniques to accelerate convergence and enhance numerical stability, such as a predictor-based coupling strategy, are introduced, which constitutes the core methodological contribution of the thesis (see Table 2-1). Framework A is included in commercial software LS-DYNA, while framework B is further developed

based on Takami et al.'s work [107,108].

Table 2-1 Two-way coupled CFD-FEA frameworks applied in this thesis

ID	CFD solver	FEA solver	Predictor	Stabilizer
A	ICFD	LS-DYNA	No	Yes
B	STAR-CCM+	LS-DYNA	Yes	No

2.1 CFD Solver

This section details the CFD solvers, STAR-CCM+ and LS-DYNA ICFD, which are used for the coupling.

2.1.1 STAR-CCM+

The finite volume method (FVM) implemented in a commercial CFD solver, STAR-CCM+ , is applied in the CFD calculation phase of framework B.

Table 2-2 Main characteristics of the STAR-CCM+ solver

Multiphase	VOF
Physical model of fluid	Incompressible
Solution	Unsteady implicit analysis (SIMPLE)
Turbulence model	Laminar/SST K- ω

The integral conservation equations of mass and momentum in STAR-CCM+ are solved sequentially, which is also known as a segregated flow solver. In the segregated flow solver, pressure and velocity are coupled through a pressure–velocity algorithm in which mass conservation is enforced by solving a pressure-correction equation. This equation is derived from the continuity and momentum equations and is used to obtain

an intermediate, or predicted, velocity field, which is subsequently corrected to ensure that the final velocity satisfies the continuity constraint. The overall scheme corresponds to a predictor–corrector strategy, with the pressure field determined implicitly by solving the pressure-correction equation. In FVM, the Navier-Stokes equations could be written in terms of a generic transport equation with an integral form:

$$\frac{\partial}{\partial t} \left(\int \rho \phi \, dV \right) + \int \rho \mathbf{u} \phi \, d\mathbf{a} = \int \Gamma \nabla \phi \, d\mathbf{a} + \int S_\phi \, dV \quad (2-1)$$

where ϕ represents the transport of a scalar property; $d\mathbf{a}$ denotes the surface vector; $\mathbf{u}$ is the velocity vector; S_ϕ is the source term; Γ is the diffusion coefficient. The transient term, $\frac{\partial}{\partial t} \left(\int \rho \phi \, dV \right)$, represents the rate of change of the scalar property ϕ within the control volume. The convective flux, $\int \rho \mathbf{u} \phi \, d\mathbf{a}$, represents the net rate of decrease of the scalar property ϕ across the control-volume boundaries due to convection. The diffusive flux, $\int \Gamma \nabla \phi \, d\mathbf{a}$, corresponds to the net rate of increase of the scalar property ϕ across the control-volume boundaries due to diffusion. The source term, S_ϕ , represents the creation of the scalar property ϕ within the control volume.

The main characteristic of the STAR-CCM+ solver is shown in Table 2-2. The Semi-Implicit Pressure Linked Equations (SIMPLE) algorithm is used to control the overall solution. SIMPLE can be summarized as follows:

1. Impose boundary conditions for velocity and pressure.
2. Reconstruct velocity and pressure gradients (for second-order spatial accuracy).
3. Evaluate velocity and pressure gradients required by the discretized operators.
4. Solve the discretized momentum equations to obtain an intermediate (uncorrected) velocity field, denoted by u^* .
5. Compute the uncorrected face mass fluxes ϕ_f^* from u^* .
6. Assemble and solve the pressure-correction equation to obtain the cell-centered pressure correction p' .

7. Update the pressure field using under-relaxation:

$$p^{\text{new}} = p^{\text{old}} + \alpha_p p' \quad (2-2)$$

where α_p is the pressure under-relaxation factor.

8. Update boundary pressure corrections p'_b to ensure consistency with the corrected pressure field and boundary constraints.
9. Correct the face mass fluxes using the pressure correction:

$$\phi_f^{\text{new}} = \phi_f^* + \phi_f' \quad (2-3)$$

10. Correct the cell-centered velocities:

$$u^{\text{new}} = u^* - \frac{V}{a_p} \nabla p' \quad (2-4)$$

where $\nabla p'$ is the gradient of the pressure correction, a_p is the vector (or diagonal) of central coefficients of the discretized momentum equation, and V is the cell volume.

11. Update the density due to pressure changes.

An interface-capturing method, the Volume of Fluid (VOF) multiphase model, is used in STAR-CCM+ to predict the distribution and motion of the interface between immiscible phases, thereby capturing the free surface. The distribution of the phases and the interface location are represented using the phase volume fraction α_i . The volume fraction of phase i is defined as:

$$\alpha_i = \frac{V_i}{V} \quad (2-5)$$

where V_i is the volume of phase i in the cell. The volume fractions of all phases within a computational cell must satisfy the unity constraint:

$$\sum_{i=1}^N \alpha_i = 1 \quad (2-6)$$

where N is the total number of phases.

STAR-CCM+ calculates the volume fractions of phases as follows:

- For a two-phase VOF formulation, the volume-fraction transport equation is solved only for the primary phase. The volume fraction of the secondary phase is then obtained in each cell by enforcing the unity constraint.
- When three or more VOF phases are present, the volume-fraction transport equations are solved for all phases. The resulting volume fractions are then normalized in each cell so that their sum equals unity.

2.1.2 LS-DYNA ICFD

The LS-DYNA ICFD solver is a FEM implemented commercial incompressible CFD solver, which is applied in the CFD calculation phase of framework A. The main characteristic of the ICFD solver is shown in Table 2-3.

Table 2-3 Main characteristic of the ICFD solver

Multiphase	Level set
Physical model of fluid	Incompressible
Solution	Unsteady implicit analysis
Turbulence model	Variational multiscale approach (VMS)

The incompressible Navier-Stokes equations could be written as:

$$\frac{\partial u}{\partial t} + \nabla \cdot (uu) = -\frac{1}{\rho} \nabla p + \nabla \cdot [\nu \nabla u + \nu (\nabla u)^T] \quad (2-7)$$

$$\nabla \cdot u = 0 \quad (2-8)$$

where ρ is the density of the fluid; u is the velocity of the fluid; p is the pressure; ν is the viscosity coefficient. Equation (2-7) and (2-8) represent the momentum equation

and continuity equation, respectively. A three-step fractional projection method is employed for the Navier-Stokes equations [104]:

Step 1:

$$u_i^* - \mu \frac{dt}{\rho} \frac{\partial}{\partial x_j} \left(\frac{\partial u_i^*}{\partial x_j} \right) + u_j^* dt \frac{\partial \bar{u}_i^{n+1}}{\partial x_j} = u_i^n - \kappa \frac{dt}{\rho} \frac{\partial p^n}{\partial x_j} \delta_{ij} \quad (2-9)$$

Step 2:

$$\frac{\partial}{\partial x_i} (-u_i^*) = \frac{dt}{\rho} \frac{\partial}{\partial x_i} \frac{\partial}{\partial x_j} (-p^{n+1} + \kappa p^n) \delta_{ij} \quad (2-10)$$

Step 3:

$$u_i^{n+1} = u_i^* + \frac{dt}{\rho} \frac{\partial}{\partial x_j} (-p^{n+1} + \kappa p^n) \delta_{ij} \quad (2-11)$$

where u^* and u^{n+1} represent the predicted velocity and the final step divergence free velocity, respectively; dt is the time step size; κ is a scalar value that varies from 0 to 1, determining the amount of pressure splitting; δ_{ij} is the unit tensor; $\bar{u}^{n+1}$ is the estimation of u^{n+1} , which could be written as:

$$\bar{u}_i^{n+1} = 2u_i^n - u_i^{n-1} \quad (2-12)$$

In the first step, a predicted velocity u^* is computed. This velocity usually does not satisfy the incompressibility constrain given by Equation (5). Hence, in the second step, the velocity u^* is projected into a space of divergence free vector field and a Poisson equation of pressure is obtained. This pressure will correct u^* to get a final step divergence free velocity u^{n+1} . The final step consists of moving the FE nodes to their new time step position x^{n+1} . If this is done iteratively then convergence is achieved when the FE nodes always move to the same final position.

As shown in Figure 2-2, the CFD solver uses a level set method [109], a fast and reliable technique for accurately tracking and representing moving free surfaces. An implicit function ϕ whose zero isocontour ($\phi=0$) represents the interface is introduced,

which could be defined as a distance function:

$$\varphi(x) = \begin{cases} \min(|x - x_{\text{interface}}|) & \text{on } \Omega^+ \\ 0 & \text{on } \partial\Omega \\ -\min(|x - x_{\text{interface}}|) & \text{on } \Omega^- \end{cases} \quad (2-13)$$

where x and $x_{\text{interface}}$ represent the coordinate of each CFD grid in the simulation domain and on the interface, respectively; Ω is the simulation domain. The fluid domain is defined by $\varphi > 0$ while the vacuum is defined by $\varphi < 0$. The motion of the interface where $\varphi = 0$ could be described by the simple convection equation:

$$\frac{\partial \varphi}{\partial t} + v \nabla \varphi = 0 \quad (2-14)$$

where v represents the velocity of each CFD grids; t is the time. It is an Eulerian formulation of the interface evolution since the interface is captured by the implicit function φ .

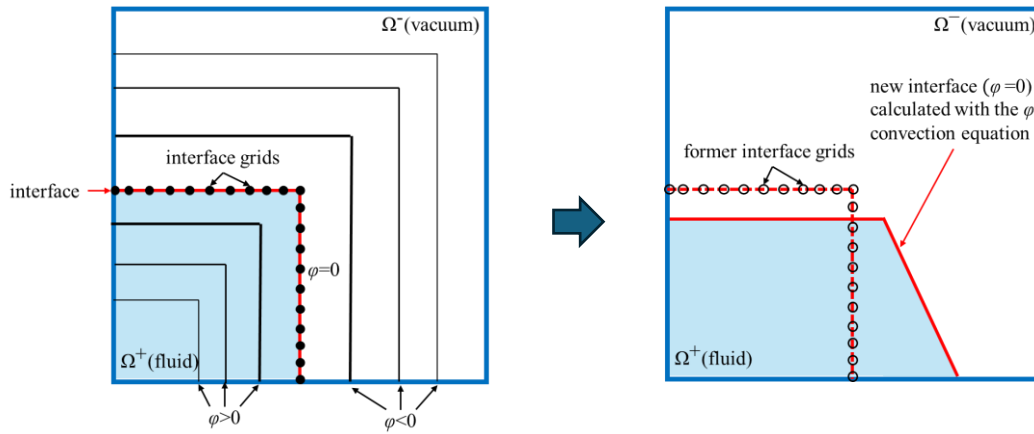


Figure 2-2 Level set interface tracking method

ALE approach for mesh movement is adopted in the CFD solver. The node of CFD cell moves by solving a linear elasticity problem using the element sizes as stiffness. Large deformation of the fluid mesh can occur in the FSI simulation. To maintain the

quality of the fluid mesh, an automatic remeshing is introduced where the solver will use an a-posteriori error estimator to compute a new mesh size to satisfy a maximum perceptual error [110].

2.2 FEA Solver

LS-DYNA mechanical solver is adopted as the FEA solver in the coupling. An implicit solution is adopted in the FEA calculation phase. For the dynamic problem, the linearized equilibrium equations can be written by:

$$M\ddot{U} + C\dot{U} + KU = R \quad (2-15)$$

where M , C and K represent the mass matrix, damping matrix, and stiffness matrix, respectively; R is the external load vector; U , $\dot{U}$ and $\ddot{U}$ respectively represent displacement, velocity and acceleration vector of FEA nodes.

As for time integration, the Newmark- β method [111] is applied in the FEA calculation phase, which can be written as:

$$\begin{cases} \ddot{U}^{t+1} = \frac{\Delta U}{\beta \Delta t^2} - \frac{\dot{U}^t}{\beta \Delta t} - \frac{1}{\beta} \left(\frac{1}{2} - \beta \right) \ddot{U}^t \\ \dot{U}^{t+1} = \dot{U}^t + \Delta t (1 - \gamma) \ddot{U}^t + \gamma \Delta t \ddot{U}^{t+1} \end{cases} \quad (2-16)$$

where the superscript t and $t+1$ represent the value at time step t and $t+1$; Δt is the time step; β and γ are time integral constants. Since the Newmark method lacks the robustness required for simulating long-term dynamic implicit problems, a three-point backward Euler step is introduced between two Newmark steps, which can be written as:

$$\begin{cases} \ddot{U}^{t+1} = \frac{(1+\alpha)}{\Delta t} (\dot{U}^{t+1} - \dot{U}^t) - \frac{\alpha}{\Delta t_n} (\dot{U}^t - \dot{U}^{t-1}) \\ \dot{U}^{t+1} = \frac{(1+\alpha)}{\Delta t} \Delta U - \frac{\alpha}{\Delta t_n} (U^t - U^{t-1}) \end{cases} \quad (2-17)$$

where Δt_n represents time step at the pre-Newmark time step; α is the time integral

constant. For these time integral constants, the Bath scheme (see Table 2-4) is reported to preserve energy and momentum to a reasonable degree [112]. If necessary, the time integral constant should be adjusted based on the Bath scheme. Usually, increasing the value of β and γ can bring additional numerical damping, which will increase the stability of the simulation.

Table 2-4 Bath scheme

Parameter	Bath scheme
α	0.5
β	0.25
γ	0.5

Rayleigh-type damping defined below was applied to the FE model:

$$[C] = f_m [M] + f_k [K] \quad (2-18)$$

where $[C]$, $[M]$ and $[K]$ represent the damping, mass and stiffness matrices, respectively; f_m and f_k are the mass and stiffness proportional damping constants of a particular mode. f_m and f_k have a relationship with the damping ratio ζ , which could be written as follows:

$$\zeta = \frac{f_m}{2\omega} + \frac{\omega f_k}{2} \quad (2-19)$$

where ω is the target angular frequency.

2.3 CFD-FEA Coupling

As a partitioned approach, fluid and structure solvers are integrated to solve the FSI problem, which requires the convergence of the fluid and solid values at the interface and provides the same results as the monolithic approach after iteration-

resolution progress (see Figure 2-3). The procedure could be summarized as follows:

- i. Advance the fluid/structure solver from time step t_n to t_{n+1} and obtain the updated values $\mathcal{F}_{t_{n+1}}^i/\mathcal{S}_{t_{n+1}}^i$ at the FSI interface. The updated values $\mathcal{F}_{t_{n+1}}^i/\mathcal{S}_{t_{n+1}}^i$ at the FSI interface are applied to the structure/fluid solver.
- ii. Advance the structure/fluid solver from time step t_n to t_{n+1} and obtain the updated values $\mathcal{S}_{t_{n+1}}^i/\mathcal{F}_{t_{n+1}}^i$ at the FSI interface. The updated values $\mathcal{S}_{t_{n+1}}^i/\mathcal{F}_{t_{n+1}}^i$ at the FSI interface are applied to the fluid/structure solver.
- iii. Advance the fluid/structure solver from time step t_n to t_{n+1} and obtain the updated values $\mathcal{F}_{t_{n+1}}^{i+1}/\mathcal{S}_{t_{n+1}}^{i+1}$ at the FSI interface.
- iv. Check the convergence in terms of the i th and $i+1$ th values. If convergence is reached, proceed to the next time step; otherwise, return to step i .

It should be noted that the solver to be advanced first depends on a specific physical phenomenon, for instance, a ship sailing in the waves. Most of the ship's motion in such a situation can be considered a response to the wave propagating. Then, the fluid solver should take the lead, so-called fluid-leading coupling.

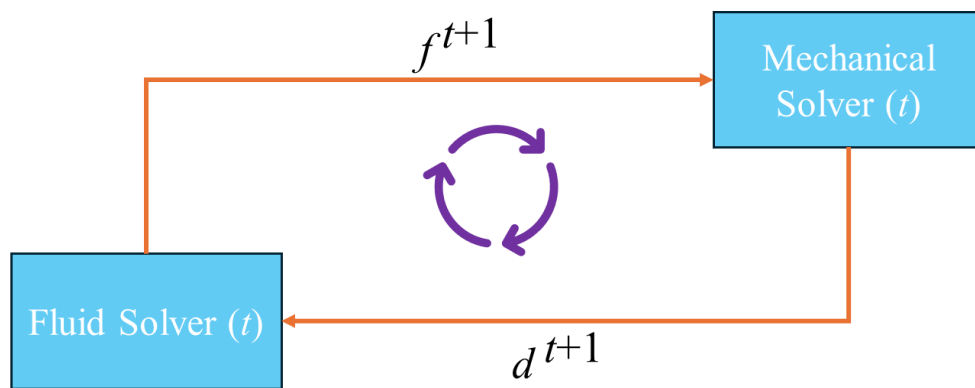


Figure 2-3 Standard partitioned approach for FSI

2.3.1 Fixed-point Equation

For a better understanding of the iteration-resolution method, this progress for the time step t_n to t_{n+1} could be simplified into a fixed-point equation (FPE) problem, which could be written as:

$$x^* = H(x^*) \quad (2-20)$$

where x^* represents the final solution for the time step t_{n+1} at the FSI interface; $H(x)$ is the FSI progress, which could be described as:

$$x_i \rightarrow S/F \rightarrow F/S \rightarrow x_{i+1} \quad (2-21)$$

where F and S represent the simulation by fluid and structure solvers, respectively; x_i and x_{i+1} are the sub-iteration values of the corresponding fixed-point iteration (FPI):

$$x_{i+1} = H(x_i) \quad (2-22)$$

where the subscript i represents the i th sub-iteration. The primary concern associated with FPI is that the progress may not converge; if it does, it may exhibit only linear convergence, which may be unacceptably slow. A straightforward method to enforce convergence is to use a suitable under-relaxation [113]:

$$x_{i+1} = (1-\omega)x_i + \omega H(x_i) \quad (2-23)$$

where ω is the under-relaxation factor. This approach has been widely used for partitioned FSI simulations and has proven its stability [86,114]. However, a constant value of under-relaxation factor ω could potentially alleviate slow convergence and even divergence in some cases as well. Adaptive under-relaxation approaches such as the Aitken method clearly exhibit better convergence [113,115,116]. Aitken under-relaxation could be defined as:

$$\omega_i = -\omega_{i-1} \frac{r_i(r_i - r_{i-1})}{\|r_i - r_{i-1}\|_2^2} \quad (2-24)$$

where $r_i (=H(x_i)-x_i)$ is the residual at the i th iteration.

2.3.2 Convergence Criteria

To decide whether to perform further sub-iterations or proceed to the next time step, convergence criteria should be used. The choice of convergence criteria should meet the following requirements:

- Be able to ensure the stability of the overall coupled simulation, where a small remaining iteration error is needed when performing time stepping.
- Could be applied to different coupling approaches. Two approaches will be compared in this research, and different criteria may lead to varying sub-iterations.

In this research, both absolute and relative convergence criteria are applied for evaluating both structure and fluid response at the FSI interface, comparing the last ($i-1$ th) iteration with the current (i th) iteration before a possible data processing (like Aitken acceleration) is applied. The absolute and relative convergence criteria could be defined as:

$$\epsilon_i^{abs} = \|r_i - r_{i-1}\|_2 \leq \epsilon_{AC} \quad (2-25)$$

	$\epsilon_i^{rel} = \frac{\ r_i - r_{i-1}\ _2}{\ r_{ini}\ _2} \leq \epsilon_{RC} \quad (2-26)$	
--	--	--

where ϵ_i is the error of FSI at the i th sub-iteration; superscripts *abs* and *rel* represent absolute and relative, respectively; r_{ini} is the initial residual, which could be calculated by $x_0 - H(x_0)$; ϵ_{AC} and ϵ_{RC} represent the absolute and relative convergence tolerance criteria, respectively. They are both set as 1.0×10^{-5} for the frameworks A and B in this research. The simulation will proceed to the next time step once the ϵ_i of the fluid and structure responses is less than the convergence tolerance criteria.

2.3.3 Data Mapping

In partitioned coupling, particularly when employing black-box solvers, the control points at the FSI interface may differ between the fluid and structure domains. Data mapping progress is usually necessary when exchanging data between the fluid

and structure domains at the FSI interface.

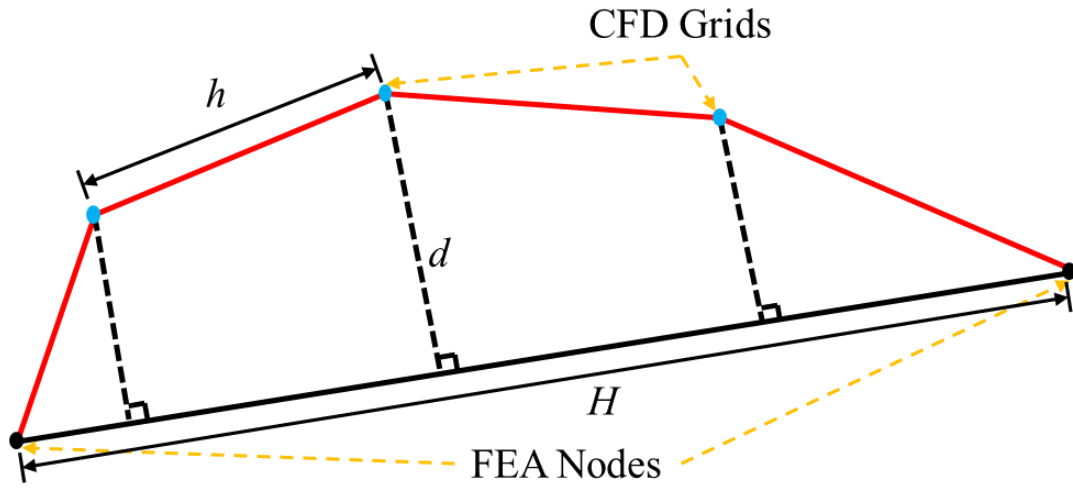


Figure 2-4 Detection of the FSI interface.

As shown in Figure 2-4, the fluid and solid surfaces that will interact in FSI simulation is detected automatically by a criterion, which is the distance d between a CFD grid/FEA node and a FEA element/CFD cell:

$$d \leq IDC \times \min(h, H) \quad (2-27)$$

where h is the size of CFD cell; H is the size of FEA element; IDC is a detection coefficient criterion. In this study, IDC is set to 0.3.

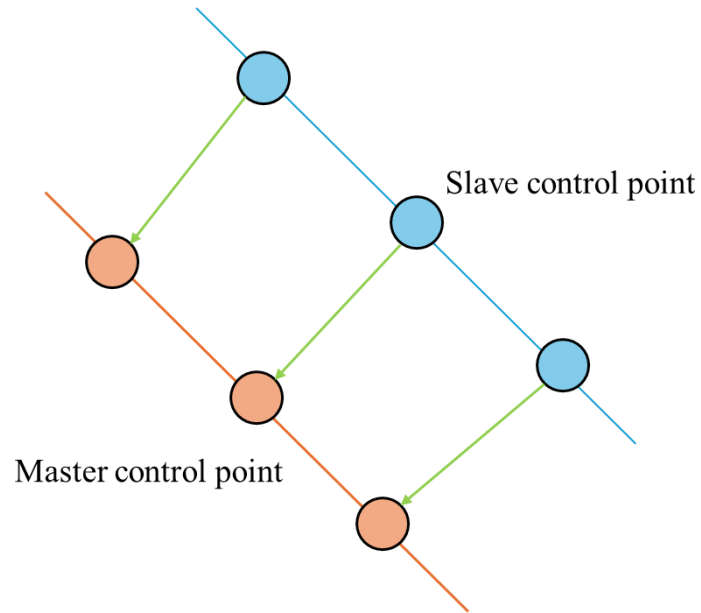


Figure 2-5 Nearest Neighbor data mapping at the FSI interface.

In the framework A, the Nearest Neighbor (NN) algorithm is used for data mapping (see Figure 2-5), which is fast, easy to use, but, of course, has numerical deficiencies. This algorithm requires the mesh of CFD and FEA to have the same level of size.

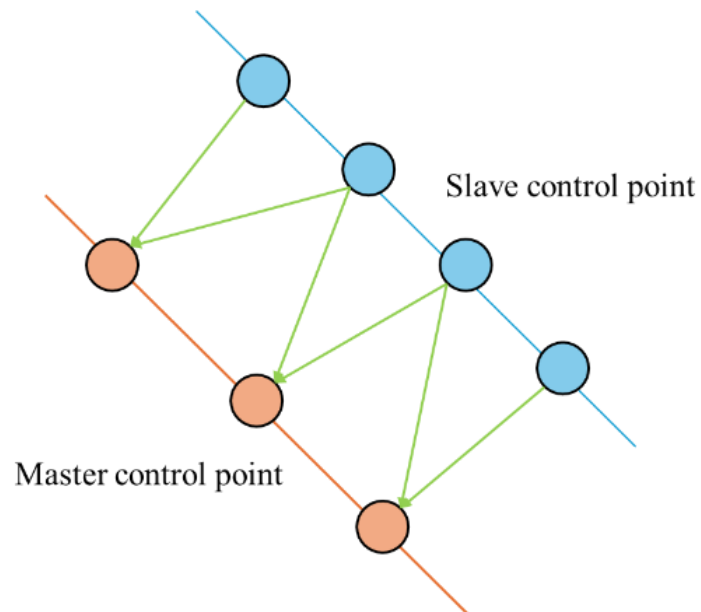


Figure 2-6 IDW data mapping at the FSI interface ($N=2$).

In the framework B, the Inverse Distance Weighting (IDW) algorithm (see Figure 2-6), proven effective and stable [32], is used for data mapping, which could be described as follows (CFD to FEA):

$$\begin{cases} P(x, y, z, t) = \frac{\sum_{i=1}^{N_{\text{gird}}} w_i(x, y, z) F(x_i, y_i, z_i, t)}{\sum_{i=1}^{N_{\text{gird}}} w_i(x, y, z)} \\ w_i(x, y, z) = \frac{1}{d(x, y, z, x_i, y_i, z_i)^k} \end{cases} \quad (2-28)$$

where $P(x, y, z, t)$ is the external pressure at target FE node; $F(x_i, y_i, z_i, t)$ is the hydrodynamic load at neighbor CFD grids; N_{gird} is the number of neighbor CFD grids; $w_i(x, y, z)$ is the distance weighting factor; $d(x, y, z, x_i, y_i, z_i)$ is the distance between each neighbor CFD grid to the target FE node; k is the power value. Six CFD grids are employed to execute mapping analyses. With an increase in the power value k , the weights for distant points decrease rapidly. If the power value k is very high, only the immediate surrounding points will influence the prediction. In this research, k is set to 2. The IDW weighting factor will be updated at the beginning of every time step, considering the potential changes at the FSI interface due to the structure response. It should also be noted that IDW mapping requires the target FSI interface to be smooth and continuous. When the mapping comes to FEA to CFD, the displacement field will be mapped to CFD by the IDW algorithm, and the fluid mesh will be updated via the mesh morphing process. Figure 2-7 shows an example of the morphed displacement along the ship hull.

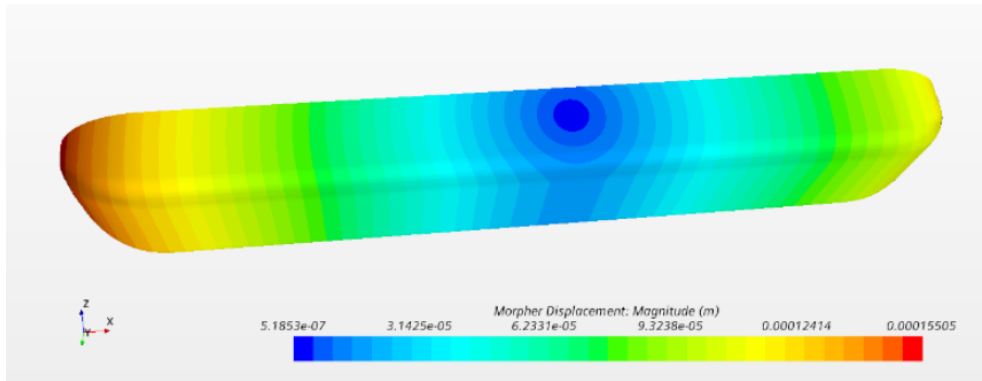


Figure 2-7 Example of morphed displacement magnitude.

2.3.4 Framework A (Stabilizer)

Conducting a coupled simulation can be particularly challenging when the added mass effect is significant. In fluid mechanics, added mass refers to the extra inertia added to a system when an accelerating or decelerating body displaces the physical volume of the surrounding fluid. In the literature, however, the term ‘added-mass effect’ is also used to describe numerical instabilities that occur in simulations where the fluid and structure have similar densities [117,118]. Other causes for such instabilities include elasticity coefficients or an excessively small time step size. In the standard partitioned scheme, the structure will go through the FSI interface without considering any change of fluid resistance and then be pulled back in the next sub-iteration. This progress is similar to the penalty method [119], which has been widely applied to Arbitrary Euler–Lagrange (ALE) simulations [68]. Therefore, there is always a lag between the CFD and FEA calculations, where inertia terms (such as the added mass from the fluid) are not updated in a timely manner. Numerical oscillation may occur during this process, especially when the density of the fluid and the structure are similar or the structure is highly flexible.

To avoid the instability caused by added mass effects in FSI simulation, one standard solution for partitioned schemes is to adopt a predictor or accelerator as

discussed in section 2.3.1 and 2.3.5 to force a convergence of the FSI simulation, which does not require any modification to the CFD or FEA code. Instead of using a predictor or accelerator, pressure calculated in the CFD simulation phase is specially handled by an FSI interface stabilizer (Laplace matrix and stabilization matrix) in the framework A. This approach was proposed by Idelsohn et al. [120] and included in commercial software LS-DYNA to realize the coupling between the LS-DYNA mechanical solver and the ICFD solver.

To clarify how the stabilizer works, the discussion should start with the basic idea of the monolithic approach. In a monolithic approach, the global FSI problem could be written in the form:

$$\begin{bmatrix} (\frac{M_p}{\Delta t} + K) & -B \\ -B^T & S \end{bmatrix} \begin{bmatrix} V \\ P \end{bmatrix} = \begin{bmatrix} F + \frac{M_p}{\Delta t} V^n \\ 0 \end{bmatrix} \quad (2-29)$$

where V is the velocity field; P is the pressure field; M_p is the mass matrix, which is a function of the fluid/structure density and shape functions; K is the stiffness matrix; Matrix B is defined as:

$$B^T = [B_1^T, B_2^T, B_3^T] \quad (2-30)$$

where $B_i^T = \int N_p \frac{\partial N^T}{\partial x_i} dV$; N is the shape function. Matrix B has non-zero terms only in the degrees of freedom related to the fluid, including the FSI interface, which means that it affects the incompressible part of the domain. S is the stabilization matrix, which will be defined in the latter part of this section. By applying the pressure segregation, the system of equations to be solved (Equation (2-29)) becomes:

$$\begin{bmatrix} (\frac{M_p}{\Delta t} + K) & -B \\ -B^T & S \end{bmatrix} \begin{bmatrix} V \\ \delta P \end{bmatrix} = \begin{bmatrix} F + \frac{M_p}{\Delta t} V^n + P_0 \\ -SP_0 \end{bmatrix} \quad (2-31)$$

where $\delta P = P - P_0$ is the change of variable; P_0 is any arbitrary vector. Pressure field in Equation (2-31) could be segregated from the velocity field by implementing the

static condensation by following:

$$\tilde{V} = \left(\frac{M_\rho}{\Delta t} + K \right)^{-1} \left(F + \frac{M_\rho}{\Delta t} V^n + BP_0 \right) \quad (2-32)$$

$$\left[-B^T \left(\frac{M_\rho}{\Delta t} + K \right)^{-1} B + S \right] \delta P = B^T \tilde{V} - SP_0 \quad (2-33)$$

$$\left(\frac{M_\rho}{\Delta t} + K \right) (V - \tilde{V}) = B \delta P \quad (2-34)$$

However, this procedure is very expensive from the viewpoint of computational cost. Since $M_\rho = \sum M^e$ and $K = \sum K^e$, where M^e and K^e represent the mass and stiffness matrices of an element, the original matrix $\left(\frac{M_\rho}{\Delta t} + K \right)^{-1}$ could be approximated:

$$\left(\frac{M_\rho}{\Delta t} + K \right)^{-1} = \sum \left(\frac{M^e}{\Delta t} + K^e \right)^{-1} \approx \sum \tau^e [M^e]_D^{-1} = M_T^{-1} \quad (2-35)$$

where M_T^{-1} is a diagonal matrix obtained from the assembly of the element contributions $\tau^e [M^e]_D^{-1}$, with:

in the solid domain:

$$\tau_s^e = \left(\frac{\rho_s}{\Delta t} + \frac{\Delta t \lambda}{Jh^2} + \frac{\Delta t G}{Jh^2} \right)^{-1} \quad (2-36)$$

in the fluid domain:

$$\tau_f^e = \left(\frac{\rho_f}{\Delta t} + \frac{\mu}{h^2} + \frac{|V^e - V_m^e|}{h} \right)^{-1} \quad (2-37)$$

By implementing the Chorin-Teman projection method [121], the term $B^T \left(\frac{M_\rho}{\Delta t} + K \right)^{-1} B$ in Equation (2-33) could be further approximated:

$$B^T \left(\frac{M_\rho}{\Delta t} + K \right)^{-1} B \approx B^T M_T^{-1} B \approx L_\tau \quad (2-38)$$

where $L_\tau = \sum (\tau^e L^e)$; $L^e = \int \frac{\partial N}{\partial x_j} \frac{\partial N^T}{\partial x_j} dV$ is the Laplace matrix for the element. Hence,

Equation (2-33) could written as:

$$[-L_\tau + S]\delta P = B^T \tilde{V} - SP_0 \quad (2-39)$$

The definition and derivation of the stabilizing matrix could be further found in Idelsohn et al. [120]. Here, only the form of S is given:

$$S = D^T M_{\tau_0}^{-1} D \quad (2-40)$$

where $D^T = [D_1^T, D_2^T, D_3^T]$ and $D_i^T = \int \frac{\partial y}{\partial x} N_p^T dV$; M_{τ_0} could be defined as:

$$M_{\tau_0} = \begin{bmatrix} \int N \frac{1}{\tau_0} N^T dV & 0 & 0 \\ 0 & \int N \frac{1}{\tau_0} N^T dV & 0 \\ 0 & 0 & \int N \frac{1}{\tau_0} N^T dV \end{bmatrix} \quad (2-41)$$

where $\tau_0 = \frac{\Delta t}{\rho}$.

Then, let's consider the classical boundary conditions at the FSI interface:

$$(V_f - V_s)^T = 0 \quad (2-42)$$

$$\sigma_f + \sigma_s = 0 \quad (2-43)$$

where V and σ represent the velocity and force, respectively; subscript f and s indicate fluid side and solid side. Equation (2-42) represents the consistency condition, which ensures that the fluid and solid meshes remain tightly coupled along the FSI interface. Equation (2-43) represents the equilibrium of forces along the FSI interface. Finally, the FSI equations of the partitioned approach could be solved as follows:

1. Predict the fluid velocity $\tilde{V}$ in the fluid domain

$$\tilde{V} = \left(\frac{M_p}{\Delta t} + K \right)^{-1} \left(F + \frac{M_p}{\Delta t} V_f^n + BP_0 \right) \quad (2-44)$$

2. Calculate the solid velocity V_s

$$V_s = \left(\frac{M_p}{\Delta t} + K \right)^{-1} (F_{\text{FSI}} + \frac{M_p}{\Delta t} V_s^n + BP_0) \quad (2-45)$$

3. Solve the poisson's equation with pressure stabilizing

$$-(L_\tau + L_{\tau_0})P = B^T(\tilde{V} + V_{\text{FSI}}) - L_\tau P_0 - SP_0 \quad (2-46)$$

4. Correct the fluid velocity

$$V_f = \tilde{V} + \frac{\Delta t}{\rho} M_D^{-1} BP \quad (2-47)$$

5. Check the convergence of V_{FSI} . If convergence is achieved, proceed to the next time step; otherwise, return to step 1.

The iterations performed from equation (2-44) to (2-47) are a standard fixed-point problem. As the pressure obtained from equation (2-24) has been stabilized by L_τ and S , which already considers the effect from the solid side, no acceleration technique is needed to improve the convergence of the coupling. Although the presented framework A has demonstrated stability in CFD-FEA coupled simulations, a shortage is also evident: the CFD code requires modification. Framework B, which does not require modification of the CFD or FEA code, is easier to apply in the engineering field.

2.3.5 Framework B (Predictor)

The in-house model fully implements the coupling of CFD and FEA. As a partitioned approach, CFD and FEA solvers are integrated to solve the FSI problem, which requires the convergence of the fluid and solid variables at the interface and provides the same results as the monolithic approach after a sub-interactive process. The coupling procedure of framework B is schematically explained in Figure 2-8, which could be summarized as follows:

- (1) $i=0$ (the predictor):

1. A predicted displacement field $u_{t_n}^{\text{pre}}$ at the next time step ($t=t_n$) is calculated by the predictor and transferred to the CFD mesh. The fluid mesh is updated ($X_{t_n}^{\text{pre}}$) via mesh morphing process (see Figure 2-8).

2. Advance the CFD solution to the next time step ($t=t_n$) where changing the fluid mesh from $X_{t_{n-1}}$ to $X_{t_n}^{\text{pre}}$ and then obtaining the predicted hydrodynamic load field, $F_{t_n}^{\text{pre}}$.

3. Convert the predicted hydrodynamic load field $F_{t_n}^{\text{pre}}$ to the external pressure field $P_{t_n}^{\text{pre}}$ on the FEA model by a data mapping progress.

4. Advance the FEA solution to the next time step ($t = t_n$) where changing the external pressure field from $P_{t_{n-1}}$ to $P_{t_n}^{\text{pre}}$ and then obtain the displacement field, $u_{t_n}^1$.

5. Check the convergence in terms of the displacement field. If the absolute and relative difference between $u_{t_n}^{\text{pre}}$ and $u_{t_n}^1$ is less than the absolute and relative tolerance, then proceed to the next time step ($t=t_{n+1}$); or, $i=i+1$ and simulation will continue to (2).

(2) $i>0$ (the corrector)

6. Transfer $u_{t_n}^i$ to the CFD mesh and update fluid mesh $X_{t_n}^i$ via mesh morphing process.

7. Advance the CFD solution to the next time step ($t = t_n$) where changing the fluid mesh from $X_{t_{n-1}}$ to $X_{t_n}^i$ and then obtaining the predicted hydrodynamic load field, $F_{t_n}^i$.

8. Convert the predicted hydrodynamic load field $F_{t_n}^i$ to the external pressure

field $P_{t_n}^i$ on the FEA model by a data mapping process.

9. Advance the FEA solution to the next time step ($t = t_n$) where changing the external pressure field from $P_{t_{n-1}}$ to $P_{t_n}^i$ and then obtain the displacement field, $u_{t_n}^{i+1}$.

10. Check the convergence. If absolute and relative tolerance are reached, then proceed to the next time step ($t=t_{n+1}$); or, $i=i+1$ and simulation will go back to step 6.

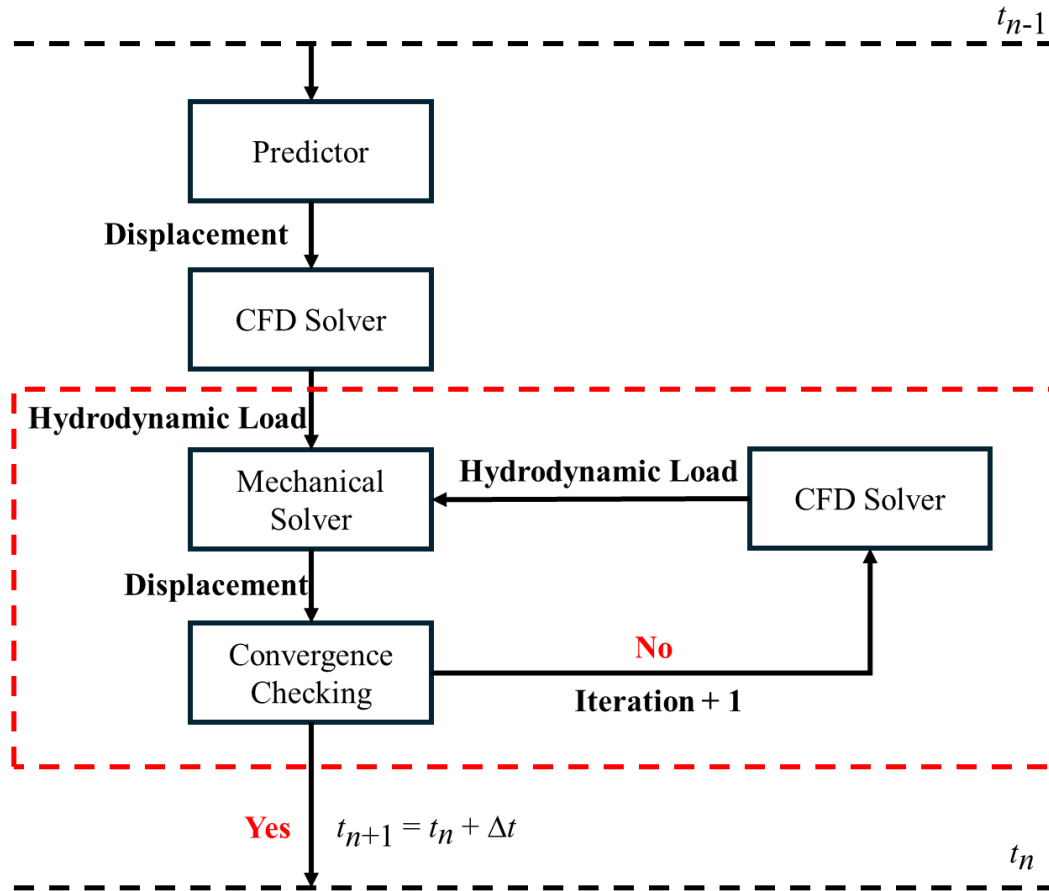


Figure 2-8 Flowchart of framework B.

To avoid impulse load at the beginning of the CFD-FEA coupled simulation, a ramp factor is applied to the external pressure field $P_{t_n}^i$ on the FEA model. The ramp factor could be defined as:

$$f(t) = \begin{cases} \left(\frac{t}{t_{\text{ramp}}}\right)^2, & t < t_{\text{ramp}} \\ 1.0, & t \geq t_{\text{ramp}} \end{cases} \quad (2-48)$$

where, $f(t)$ is the ramp factor; t is the current simulation time; t_{ramp} is the time period during which $f(t)$ will be applied to the external pressure field.

In step 1, the displacement field $u_{t_n}^{\text{pre}}$ is predicted by the predictor. As mentioned, to enforce the convergence of FPI, one straightforward idea is to employ an acceleration technique, such as the Aitken acceleration. Such accelerators could effectively reduce the total number of sub-iterations required for convergence. However, the performance of these acceleration methods is sometimes highly case-dependent [103]. For complex physical problems, a significant number of sub-iterations may still be necessary to achieve convergence.

The following discussion will revisit the FPE and FPI described in Equations (2-20) and (2-22). It is not difficult to notice that the final solution x^* , the input value x' , and the residual r have the following relationship:

$$\lim_{x' \rightarrow x^*} r = \lim_{x' \rightarrow x^*} [x' - H(x')] = 0 \quad (2-49)$$

which indicates that using a good starting value as input for the FPI can result in a notable decrease in the total number of sub-iterations. However, directly predicting the behavior between elastic structures and complex fluid fields is usually challenging, especially when the so-called added-mass effect [117,122,123] is significant. One idea is to use the information from previous time steps to predict the phenomenon at the current time step [103,124]. For example, the displacement field at the FSI interface could be predicted by the 3-step Adams-Bashforth method [125], which could be written as:

$$d_{t_n}^{\text{pre}} = d_{t_{n-1}} + \Delta t \left(\frac{23}{12} v_{t_{n-1}} - \frac{4}{3} v_{t_{n-2}} + \frac{5}{12} v_{t_{n-3}} \right) \quad (2-50)$$

where $d_{t_{n-1}}$ is the known displacement filed from the last time step t_{n-1} ; $v_{t_{n-1}}$, $v_{t_{n-2}}$ and $v_{t_{n-3}}$ represent the velocity filed at time step t_{n-1} , t_{n-2} and t_{n-3} respectively; Δt is the time step size. However, this kind of predictor does not consider the FSI phenomena, such as the added-mass effect.

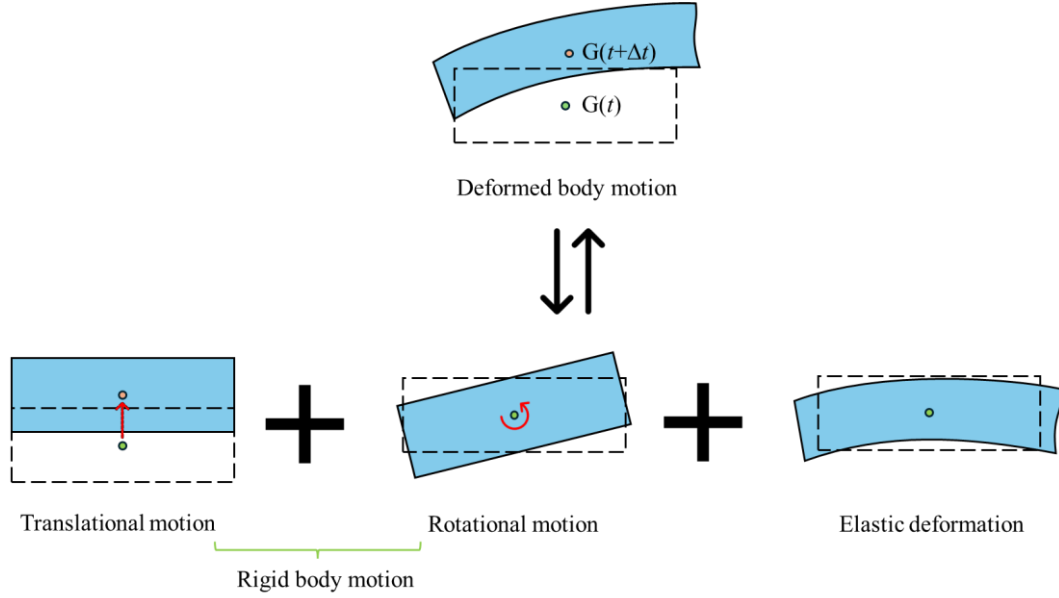


Figure 2-9 Decomposition of a deformed body.

Another potential method is to simplify the physical problem while obtaining a reasonably accurate predicted value. In this research, an RBM-Deformation decomposed predictor is proposed. As illustrated in Figure 2-9 (where G represents the center of gravity), the motion of a deformed body could be decomposed into two parts: rigid body motion and elastic deformation. Then, one straightforward idea is to simplify the FSI problem between a fluid and an elastic body to the FSI problem between a fluid and a rigid body in the prediction phase, referred to as the RBM-deformation decomposed predictor. The benefits of doing so are shown as follows:

- The techniques for describing the interaction between fluid and rigid bodies are well-developed. Numerous methods, such as those based on potential theory, have been extensively validated for fluid-rigid body interaction problems, demonstrating

their ability to predict RBM outcomes effectively.

- The interaction between fluid and rigid bodies can be readily estimated. Utilizing RBM as a predictor in partitioned fluid-structure coupling offers a computationally efficient solution.
- Easy to implement, requiring minimal modifications to the existing CFD solver, FEA solver, and coupling code.

The RBM-deformation decomposed predictor can be based on any theory or algorithm and only needs to predict a value that approximates the actual RBM. It is worth noting that this research focuses solely on the performance and applicability of the RBM-deformation decomposed predictor. Hence, a detailed introduction and comparison of various RBM predictors is beyond the scope of the current research. This research predicts the RBM by a simple implicit coupled simulation between CFD and 6-degree-of-freedom (DOF) RBM equations. The equations of free motion could be defined as:

$$\begin{cases} m \frac{dV}{dt} = F \\ J \frac{d\Omega}{dt} + \Omega \times J \Omega = M \end{cases} \quad (2-51)$$

where m is the mass of the rigid body; F is the resultant force vector acting on the body; V is the velocity vector of the center of mass; J is the tensor of the moments of inertia; Ω is the angular velocity vector of the rigid body; M is the resultant moment vector acting on the body.

It is important to note that the proposed predictor should be conditional. Intuitively, decomposing RBM and deformation is usually beneficial when RBM contributes more to the total body motion than deformation. Therefore, clarifying the conditions under which this approach is applicable is crucial to ensure its validity and facilitate its wider adoption.

As noted in the Introduction, several numerical benchmarks-such as FSI3 [126,127]-have been proposed to evaluate the performance of FSI methods. However, RBM is typically not considered in these benchmarks. Here, an example problem is presented to illustrate the proposed RMB-deformation decomposed predictor, assess its accuracy and computational efficiency, and highlight its benefits across a range of engineering problems.

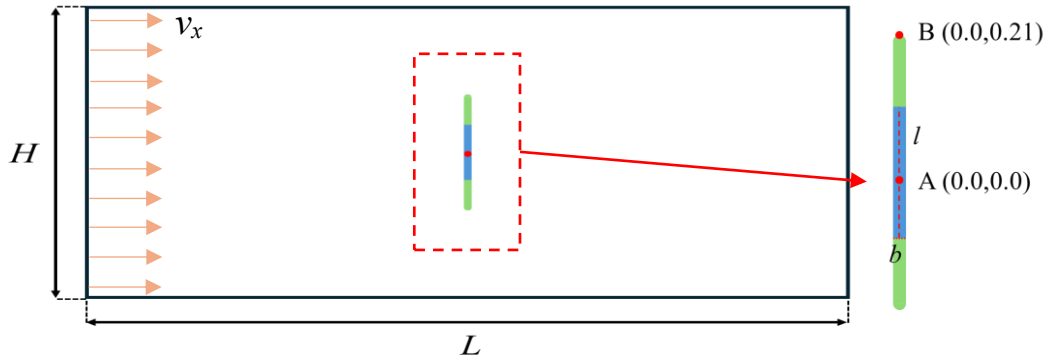


Figure 2-10 Schematic diagram of the example case.

As shown in Figure 2-10, the simulation domain is 2.0 m in length L and 1.0 m in height H . The fluid is considered to be Newtonian and incompressible. The target body consists of two parts: elastic bars (green) and an elastic base (blue). The body center is positioned at reference point A (0.0,0.0). The elastic bar is 0.1 (m) in length and 0.01 (m) in width, and its root is fully attached to the elastic base. Reference point B is positioned at (0.0,0.21). Displacement at B will be measured. The velocity distribution at the left boundary of the fluid domain is characterized as:

$$v_x = 0.5 \sin(2\pi t) \quad (2-52)$$

where v_x is the fluid velocity along the x direction; t represents the simulation time.

The choice of the material parameters is guided by the following considerations:

- The flow in a laminar regime is preferred, which implies relatively small Reynolds numbers [126].
- The flow should be capable of deforming the elastic structure.
- To clarify the conditions under which the proposed approach is applicable, not only the common engineering scenarios but also extreme cases, such as those involving highly flexible materials, should be considered.

An overview of material properties for fluid and structure is shown in Table 2-5. Glycerin is chosen as the fluid material. The base of the structure body is modeled by steel. As for the elastic bar, two cases are considered: in case I, polypropylene (rubber-like material) is chosen so that the material's stiffness is low enough to allow significant deformation; in case II, steel is used.

Table 2-5 Overview of the structure and fluid material parameters

Structure material	Density	Elastic modulus	Poisson's ratio
Steel	7,580 (kg/m ³)	2.1 (10 ¹¹ Pa)	0.3
Polypropylene	1,100 (kg/m ³)	1.6 (10 ⁶ Pa)	0.37
Fluid material	Density	Kinematic viscosity	
Glycerin	1,260 (kg/m ³)	1.42 (Pa·s)	

The calculation setup of CFD is shown in Table 2-6. The FVM implemented in a commercial CFD solver, STAR-CCM+, is applied in the CFD calculation phase. SIMPLE achieves an implicit coupling between pressure and velocity. The time step size is set as 0.01 s.

Table 2-6 Calculation setup of CFD, example case

Space model	The two-dimensional model
-------------	---------------------------

Physical model of water/air	Incompressible
Viscous Regime model	Laminar flow
Time step size	0.01 (s)
Solution	SIMPLE unsteady implicit analysis

The overall CFD domain at the cross is shown in Figure 2-11. The left boundary is set to be a velocity inlet condition, while the right boundary is set as a pressure outlet condition. The top and bottom of the fluid domain and the body surface are set to be non-slip conditions, where the fluid tangential velocity is always zero. The basic mesh size of the fluid domain is 0.01 m. The mesh size around the body is the finest throughout the entire fluid domain, allowing for accurate calculation of the FSI load at the FSI interface. The minimum mesh size on the body is 0.005 m. A total of 42,000 cells is generated in the fluid domain.

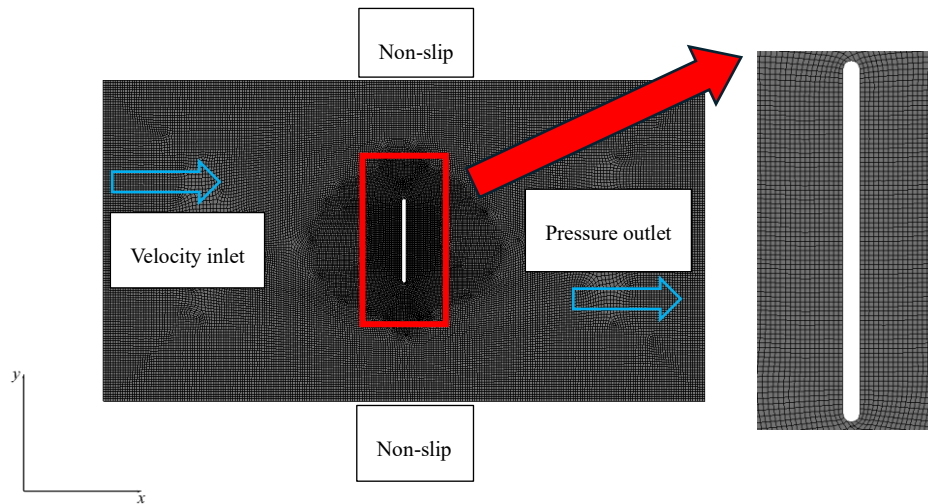


Figure 2-11 Overview of CFD Domain, example case.

A dynamic implicit commercial solver based on LS-DYNA is used to calculate the structure's response. Figure 2-12 shows the overview of the FEA model. The body is discretized with 3/4-node 2D shell element. The mesh size is 0.005 m, which is the

same as the CFD mesh size at the FSI interface. The FEA model contains 340 elements. Only the translated motion is released in the simulation for the elastic base. There is no constraint on DOF for the elastic bar.

For the time integration, the Newmark- β method is applied in the FEA calculation phase, and an additional three-point backward Euler step is introduced between two Newmark steps. For these time integral constants, the Bath scheme, which has been discussed in section 2.2, is adopted. The time step size for the FEA simulation is set as 0.01(s), which is the same as the CFD simulation to realize the strong implicit coupling.

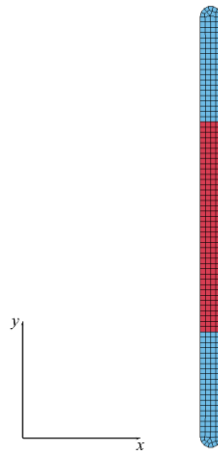


Figure 2-12 Overview of the FEA model, example case.

No predictor, 3-step Adams-Bashforth predictor, and the proposed RBM-deformation decomposed predictor yield no visible difference in the simulation results. Hence, the simulation results discussed in this section are from the RBM-deformation decomposed predictor case. It should be noted that the simulation diverged with no predictor in case II. Figure 2-13 compares the structure's Von Mises stress and mesh at time $t = 2.0$ (s) and $t = 4.5$ (s), respectively. Figure 2-14 shows the CFD pressure field and FEA mesh deformation at time $t = 4.5$ (s). Compared with case I, visible deformation could be observed from the figures. The time history of displacement at

measuring point B is shown in Figure 2-15.

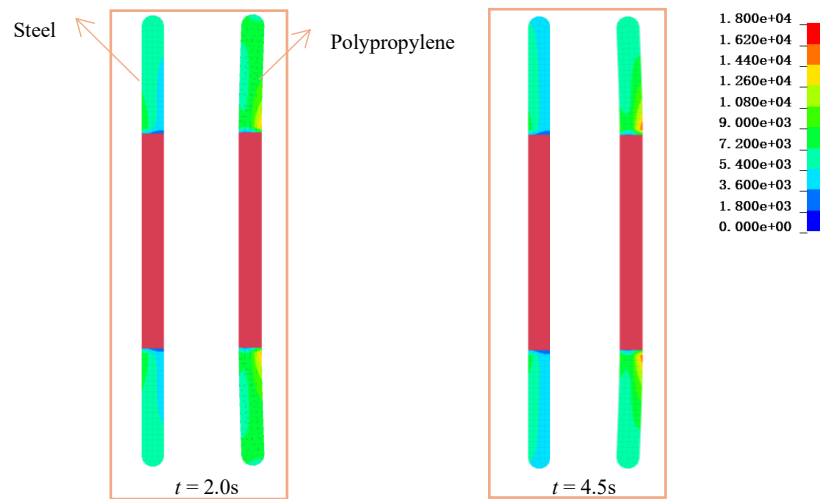


Figure 2-13 Structure response (Von Mises stress) at $t = 2.0s$ (right) and $t = 4.5s$ (left).

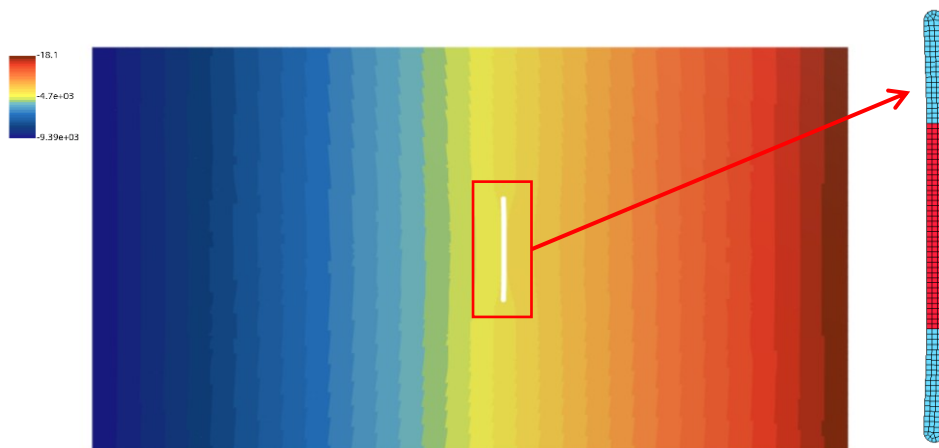


Figure 2-14 Pressure field and deformation of FEA mesh ($t = 4.5s$).

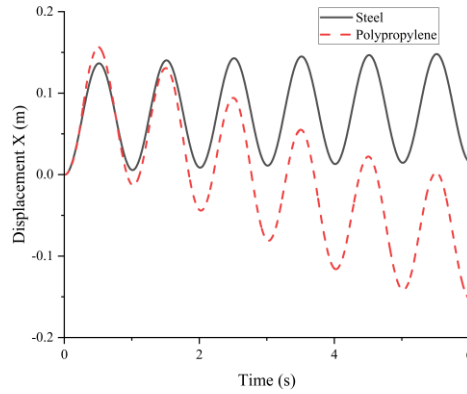
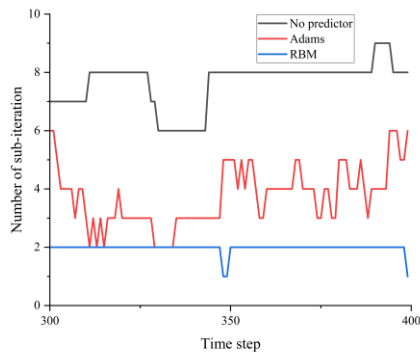
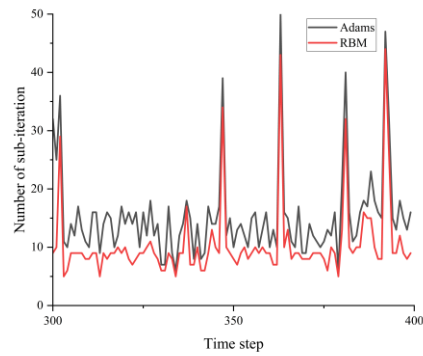


Figure 2-15 Time history of displacement at reference point B.

The resulting iteration numbers over a complete oscillation period for both methods are depicted in Figure 2-16. Table 2-7 shows the average number of sub-iterations during the whole simulation. In case I, with no predictor, the average number of sub-iterations during the whole simulation. In case I, with no predictor, the average number of sub-iterations is 7.64. The number for the RBM-deformation decomposed predictor is significantly smaller (1.97). This indicates that the RBM-deformation decomposed predictor is effective for situations where RBM contributes more to the total body motion than deformation. In case II, as the elastic bar becomes highly flexible, the predictor's effectiveness is limited.



(a) case I



(b) case II

Figure 2-16 Number of sub-iterations with different predictors.

Table 2-7 Average number of sub-iterations with different predictors

Case	No predictor	Adams	RBM
I	7.64	3.74	1.97
II	DIV	15.30	10.75

This section proposes an RBM-deformation decomposed predictor for the implicit CFD-FEA coupled method. Based on a simple 2D model, the performance of the RBM-deformation decomposed predictor is compared with the no-predictor scheme and the 3-step Adams-Bashforth predictor. The following could be concluded:

- The proposed RBM-deformation decomposed predictor can potentially decrease the total number of sub-iterations in implicit CFD-FEA coupled simulations. In this research, compared with the no-predictor scheme, the RBM-deformation decomposed predictor could save about 150% of the sub-iterations in case I.
- Intuitively, decomposing RBM and deformation is usually beneficial when RBM contributes more to the total body motion than deformation. The proposed RBM-deformation decomposed predictor is effective when RBM is significant, and deformation is relatively small, which is suitable for many ocean engineering applications.

Chapter 3

Hydroelastic Behavior of Segmented Ships

3.1 Introduction

In this chapter, the hydroelasticity of two segmented ship models (Barge and KVLCC2) is investigated by experimental and numerical methods. The two-way CFD-FEA coupled framework A is adopted and further validated for its accuracy against experimental results. The current approach emphasizes the overall convergence between two solvers, maintaining a strongly coupled manner to comprehensively address the fluid-structure interaction phenomenon, including the added mass effect. A series of experiments was conducted on a segmented Barge and the KVLCC2 under various wave conditions. The motions of the models were measured, and the displacement Response Amplitude Operators (RAOs) were calculated. The CFD-FEA coupled framework A demonstrates agreement with experimental results by comparing the numerical prediction with the tank test results in terms of motion, affirming the validity and robustness. Finally, the hydroelastic response is discussed and investigated.

The objectives of the present study could be summarized as follows:

1. Tank tests were conducted on a segmented Barge (A) and KVLCC2 model under various wave conditions. Motion was measured and hydroelasticity of the models are focused.
2. Experimental analysis and two-way CFD-FEA coupled framework A in this chapter are validated by published data.

3.2 Experimental Setup

3.2.1 Ship Models

This study employed two segmented ship models, namely Barge (A) and KVLCC2, as shown in Figure 3-1. Experiment on the Barge (A) ship is based on the experiment

carried out by Senjanović and Malenica [128], while the experiment on the KVLCC2 ship is newly designed based on the model and data provided by Zhang et al. [129].



(a) Barge (A) model



(b) KVLCC2 model

Figure 3-1 Experimental ship model.

In the Barge model, there are a total of twelve segments rigidly connected to the backbone, as shown in Figure 3-2. Apart from the bow segment (refer to Figure 3-3), the remaining eleven segments are identical in shape. The gap between each segment is set at 0.015 m. Table 1 describes the principal particulars of the Barge ship.

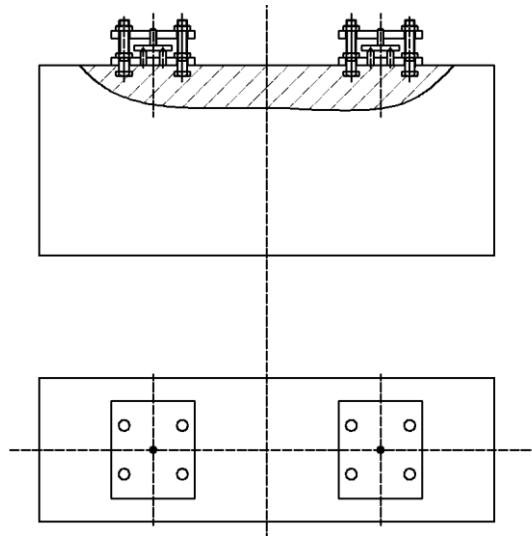


Figure 3-2 Connection between segment and backbone.

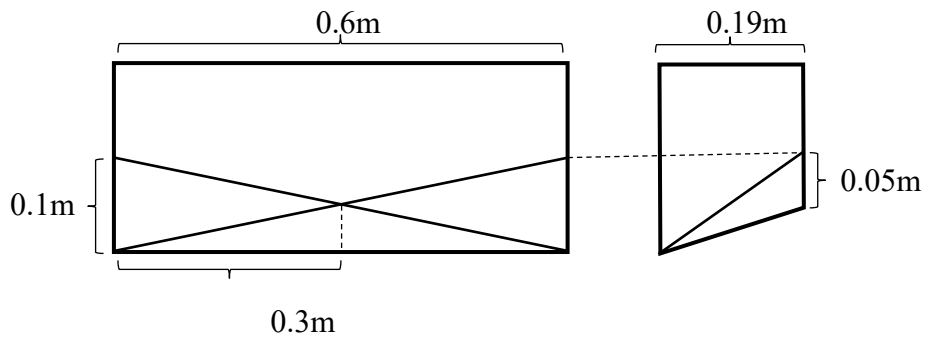


Figure 3-3 Bow segment of the Barge (A) ship

Table 3-1 Principal particulars of the Barge (A) model

	Length (m)	Width (m)	Depth / Thickness (m)	Draft (m)
Segment	0.190	0.600	0.250	0.120
Backbone	2.445	0.050	0.006	/

Regarding the KVCLL2 model, it is divided into sixteen segments due to the complexity of ship lines and the rapid changes in curvature of the ship's surface (see Figure 3-1 (b)). Compared to the original ship, the experimental model's deck has been raised to prevent green water from entering during the experiment. Similar to the Barge model, all the segments are rigidly connected to the backbone. The distance between each segment is 0.013 m. The length of F2 to F7 and F10 to F11 is 0.15 m, while the length of F8 and F9 is 0.18 m. To ensure the accuracy of the lines in the bow and stern, F1 and F16 were created by 3D printing technology (see Figure 3-5). The backbone of the model measures 2.729 m in length, 0.1 m in width, and 0.005 m in thickness. The principal particulars of the KVLCC2 ship model are listed in Table 3-2.

Figure 3-6 shows the mass distribution of the Barge and KVLCC2 models along the x -axis. In the Barge model, all segments are box-type and have a similar size, resulting in a similar weight for each segment. The KVLCC2 model's mass distribution is based on the actual ship, with the weight of each segment decreasing from the center to the two ends, as shown in Figure 3-6 (b).

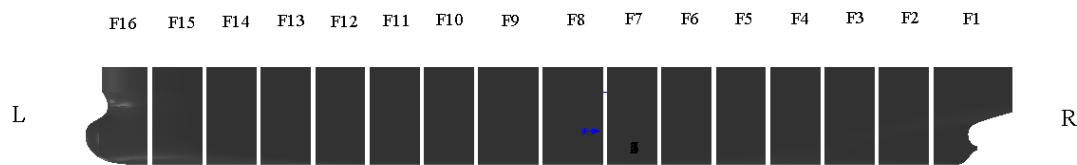
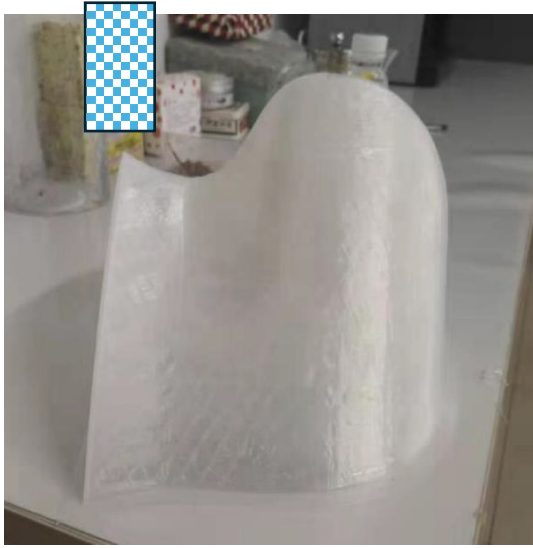


Figure 3-4 Segments of the KVLCC2 model.



(a) Bow segment

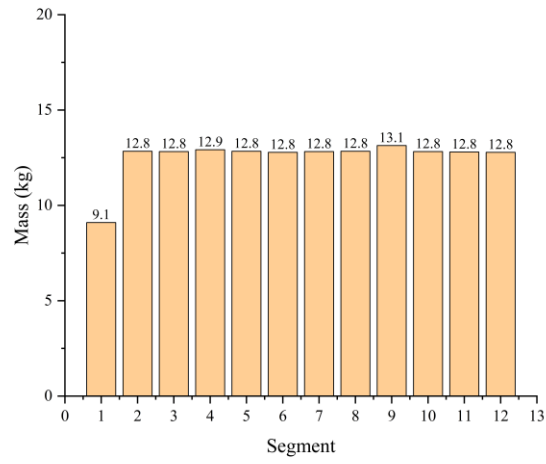


(b) Stern segment

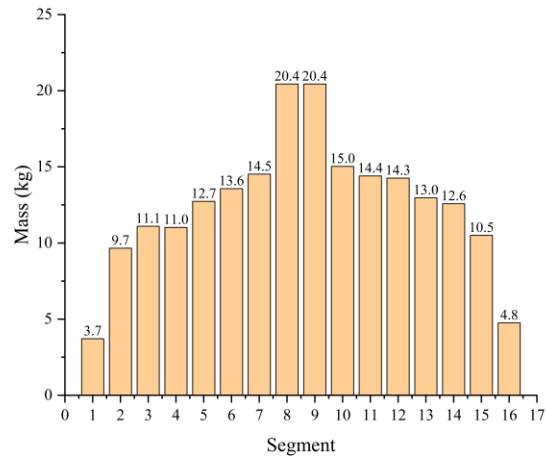
Figure 3-5 The bow and stern segments of the KVLCC2 model, made by 3D printing technology.

Table 3-2 Principal particulars of the KVLCC2 model

	Length (m)	Width (m)	Depth (m)	Draft (m)	Center of gravity (m)
Symbol	L_{pp}	B	D	d	KG
Value	2.728	0.486	0.2917	0.2117	0.155



(a) Barge (A)



(b) KVLCC2

Figure 3-6 Mass distribution of the Barge (A) and the KVLCC2 models.

3.2.2 Mooring and Measuring System

In the experiment, the ship model is constrained by springs to avoid unexpected motion (see Figure 3-7). Only heave and pitch are released.

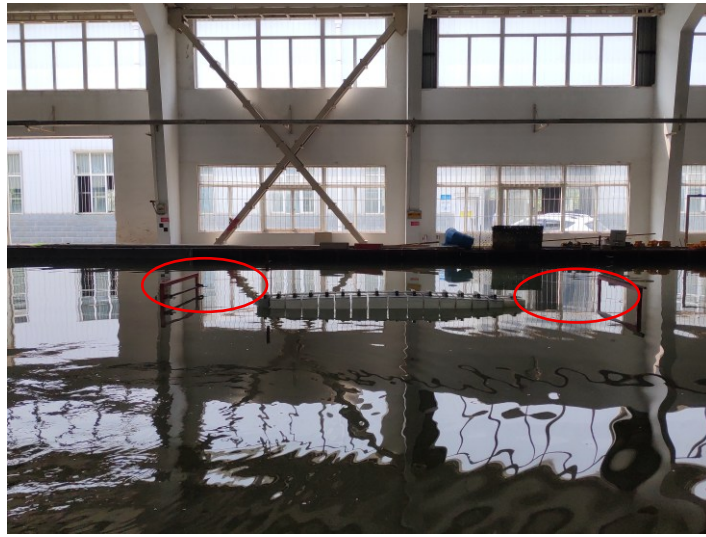
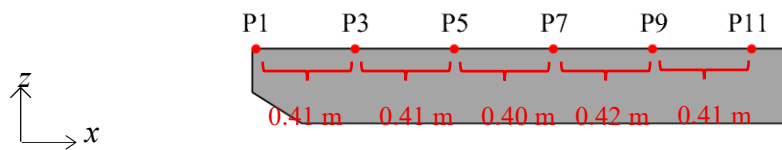
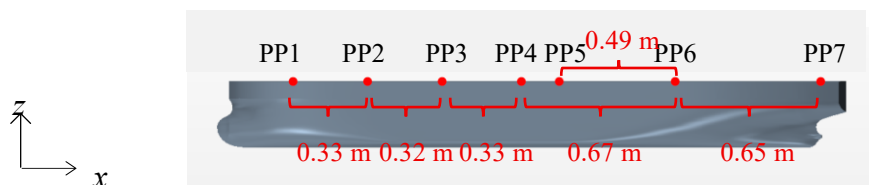


Figure 3-7 Spring mooring system.

As shown in Figure 3-9, a VISUALEYESTM 3D motion capture system, which includes a tracker and several wireless LED markers, is adopted in the experiment. The tracker captures real-time signals from multiple LED markers and computes motion with high accuracy. The location of the LED markers for Barge and KVLCC2 is shown in Figure 3-8. The motion capture system measures the displacement along the Z direction at each LED marker.

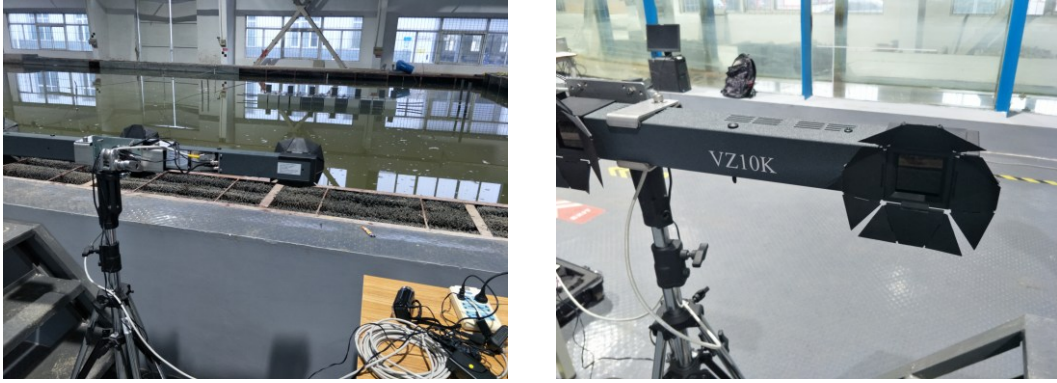


(a) Barge (A)

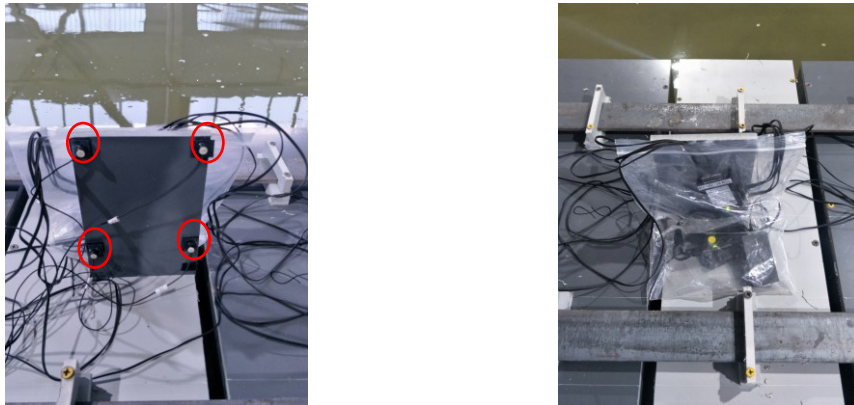


(b) KVLCC2

Figure 3-8 Location of the LED markers.



(a) Tracker



(b) Wireless LED marker

Figure 3-9 Motion measurement system

3.2.3 Wave Conditions

The tank test was carried out in the wave tank at the Tianjin Research Institute for Water Transport Engineering (TIWTE). The wave tank is 40m in length, 12m in width and 0.9m in depth. A piston wave maker is adopted, and plenty of wave eliminators are settled at the side of the tank, behind the wave maker, and at the end of the tank to avoid wave reflection (see Figure 3-10).



Figure 3-10 Wave maker and eliminator

In this experiment, the ships are investigated under different regular waves without forward speed. The wave conditions of the experiment are shown in Table 3-3. The wave period for all test conditions is determined by the sea conditions in the northern hemisphere [130].

Table 3-3 Wave conditions of the Barge (A) and KVLCC2 experiments

Wave Condition	Wave height h_0 (m)	Wave period t_0 (s)	Wavelength λ (m)
EX-A1	0.100	1.6	3.996
EX-A2	0.100	1.4	3.063
EX-A3	0.100	1.2	2.248
EX-A4	0.100	1.0	1.561
EX-C3	0.100	1.100-1.850	1.889-5.342
EX-C4	0.070	1.100-1.850	1.889-5.342
EX-C5	0.045	1.100-1.850	1.889-5.342
EX-C6	0.030	1.100-1.850	1.889-5.342

3.3 Numerical Model

3.3.1 FEA Model

The dynamic implicit commercial solver based on LS-DYNA is adopted to calculate the structure response. Rayleigh-type damping is applied to the FE model. Both mass damping and stiffness damping are considered in the simulation. The mass damping and stiffness damping coefficients are set to 0.01 and 0.001, respectively. Since the stiffness proportional damping constant largely affects the high-frequency mode of structure [131], the unstable calculation can be caused by taking the stiffness damping into account. The time step size should be forced to a significantly small value in this kind of case. Therefore, the time step size for the FEA solver is set to 10^{-5} s in this chapter. In this chapter, the settings of time integral constants are shown in Table 3-4.

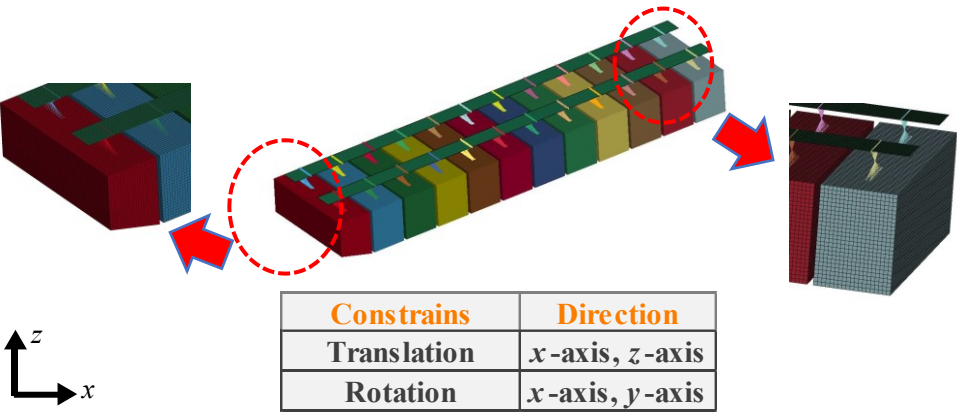
Table 3-4 FEA time integration setting, Barge (A) and KVLCC2

Parameter	In this chapter	Bath scheme
α	0.5	0.5
β	0.4	0.25
γ	0.6	0.5
time step	10^{-5} s	/

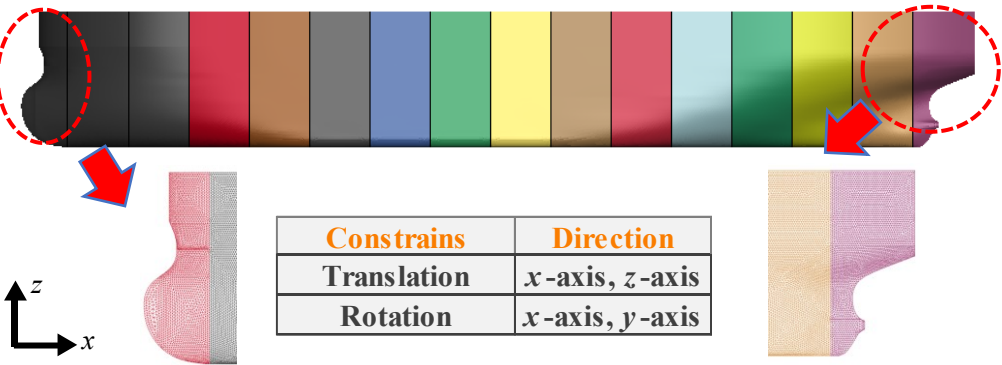
The FE models of the Barge (A) and KVLCC are shown in Figure 3-11. The FE model of the Barge (A) is the same as the experimental model, which is segmented. It should be noted that the FE model of the KVLCC2 ship is continuous and does not have a backbone. In the experiment of the KVLCC2 ship, contact between two neighboring segments was observed. Accounting for this kind of contact in the current two-way coupled methods is time-consuming. Furthermore, due to the narrow gap between neighbor segments and the complex geometry of the ship, it is necessary to ensure high

mesh quality in the CFD simulation and use a smaller time step for the two-way coupled simulation. Therefore, balancing accuracy and efficiency in the simulation, a continuous FE model has been adopted for the KVLCC2 ship.

Young’s modulus of the ship is tuned to reduce the stiffness of the FE model to a reasonable level. The material characteristics of the backbone are shown in Table 3-5. the numerical models are divided into 12 (Barge ship) and 16 parts (KVLCC2 ship), respectively, to ensure that the mass distribution is the same as the experiment models. Translation along the x and z axes, as well as rotation along the x and y axes, is constrained at the center of the ship's mass. Only the heave and pitch of the ship are released in the simulation.



(a) Barge (A)



(b)KVLCC2

Figure 3-11 FE models of the Barge (A) and KVLCC2.

Table 3-5 Material characteristics of the numerical Barge (A) and KVLCC2 models.

Parameter	Barge	KVLCC2
Material properties	Elastic	Elastic
Young's modulus	6.0×10^{12}	1.4×10^7
Poisson's ratio	0.3	0.3

3.3.2 CFD Model

LS-DYNA ICFD is adopted as the CFD solver. Overall CFD domain at cross-section $Y = 0$ is shown in Figure 3-12. CFD-FEA coupled simulation can be time-consuming, so it is important to carefully consider the size of the fluid domain. The numerical tank is $4.5L$ in length, $1.2L$ in width, and 3 m in depth. Note that the size of the fluid domain is insufficient to be considered a far-field condition. To decrease reflection from the sides and outlet boundaries, damping zones are created perpendicular to these boundaries. The damping is achieved by adding a source term S_d to the momentum equations:

$$S_d = \omega_d(f_1 + f_2|u|)u \quad (3-1)$$

where f_1 and f_2 are the damping factors; u is the velocity; ω_d is the weight function:

$$\omega_d = \frac{e^\varepsilon - 1}{e - 1} \quad (3-2)$$

with ε , the blending function which allows a smooth insertion of the source term in the damping layer:

$$\varepsilon = \left(\frac{x - x_{\text{end}}}{L_d} \right)^n \quad (3-3)$$

where x and x_{end} represent the CFD grid coordinates and end coordinates of the damping zone; L_d is the length of the damping zone; n is the scaling factor. In this research, f_1 and f_2 are both set to be 10 while n is set to be 5. The wave inlet boundary of the fluid domain is set to be a velocity inlet condition, while the wave outlet boundary is set to be a pressure outlet condition. The wave model at the inlet is based on the second-order Stokes wave [132]. Bottom and top as well as side boundaries of the fluid domain are set to be free-slip condition. The ship hull is set as a no-slip wall where the fluid tangential velocity is always zero.

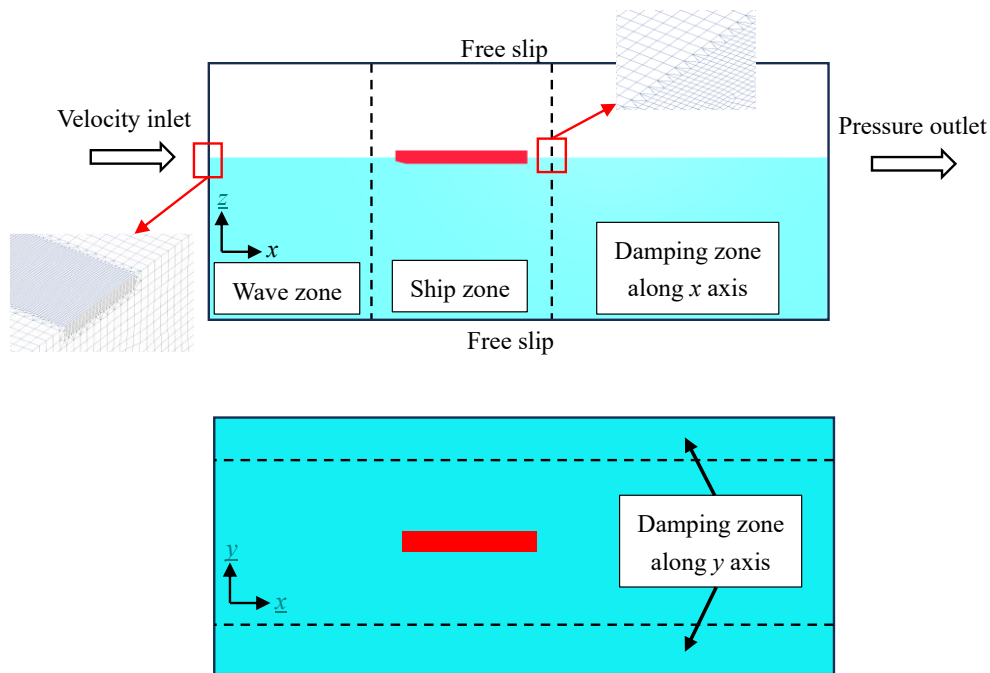


Figure 3-12 Overview of CFD domain and mesh.

The mesh size at free surface and around the ship hull is refined to capture wave and FSI load properly. A number of 80 cells per wavelength and 20 cells per wave height is recommended by ITTC [133]. In this research tetrahedral mesh is used for CFD

simulation. However, controlling the tetrahedral mesh size along a specific direction can be challenging. So, there are three criteria for mesh size: $\lambda/80$, $B/20$, $h_0/15$, and the smallest of these is used as the mesh size at the free surface zone. To calculate the FSI load on the hull with high accuracy, the mesh size around the ship hull is the finest over the whole fluid domain. The minimum size of mesh on the ship hull is $\lambda/240$. A total of 11.3 million tetrahedral cells are generated in the fluid domain.

3.4 Numerical Hammering Test

The natural frequency of a ship structure under dry and wet conditions will be different due to the added mass effect. The added mass effect could be defined as:

$$\omega_{\text{wet}} = \sqrt{\frac{1}{1+f_{\text{added mass}}}} \omega_{\text{dry}} \quad (3-4)$$

where ω_{dry} and ω_{wet} represent the dry and wet natural frequency respectively; $f_{\text{added mass}}$ is the added-mass factor.

Numerical hammering tests were conducted to determine the natural frequency of the Barge and KVCLL2 structure under dry and wet conditions. An external impulse force, equivalent to the gravitational force of the entire ship, was applied suddenly to the middle of the ship structure (see Figure 3-13). Simultaneously, the vertical bending moment (VBM) was measured at the midship.

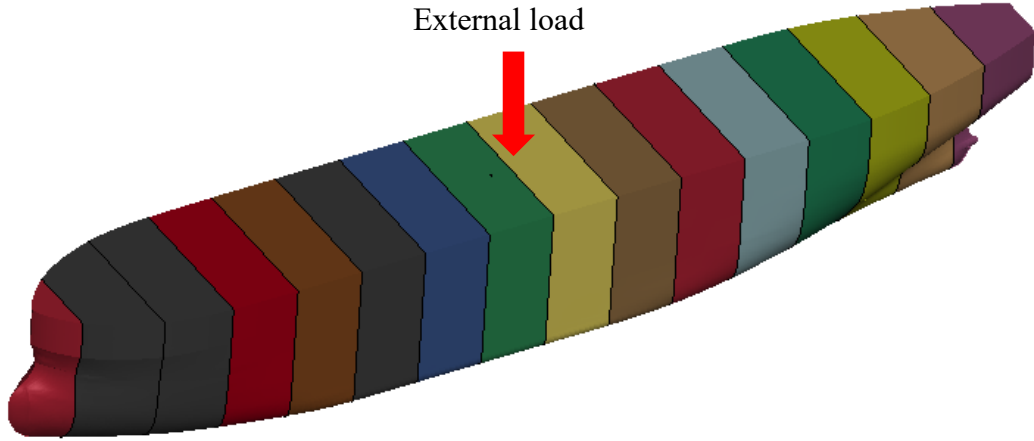


Figure 3-13 External load applied at the midship section of the hull girder.

3.4.1 Damping Ratio

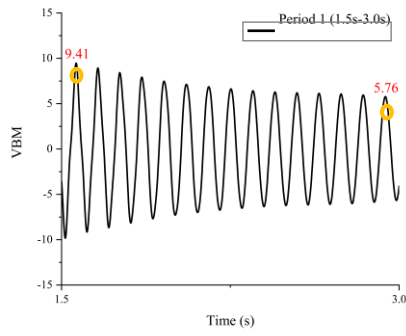
As discussed in section 2.2, if the time integration constants used in FEA simulation were set atypically, the stability of CFD-FEA coupled simulation could be enhanced. However, this unusual setting may result in unexpected numerical damping. The numerical damping in FEA simulation should be validated by examining the decay of the structural response. The damping ratio is introduced as a criterion, which can be written as:

$$\zeta = \frac{1}{\sqrt{1 + \left(\frac{2\pi}{\sigma}\right)^2}} \quad (3-5)$$

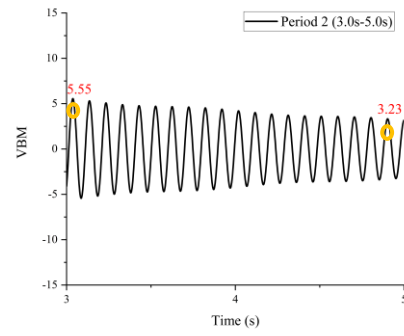
where ζ is the damping ratio; σ is the logarithmic decrement, the natural log of the ratio of the amplitudes of any two successive peaks, which can be expressed as:

$$\sigma = \frac{1}{n} \ln \frac{x_i}{x_{i+n}} \quad (3-6)$$

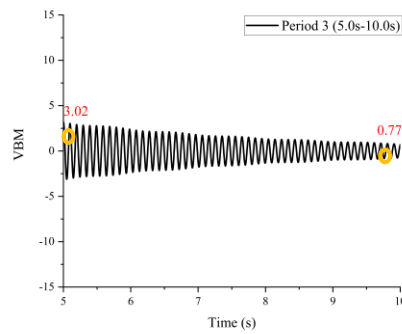
where x_i is the i -th positive peak; x_{i+n} is the $(i+n)$ -th positive peak.



(a) Time period I (1.5s - 3.0s)



(b) Time period II (3.0s - 5.0s)



(c) Period III (5.0s - 10.0s)

Figure 3-14 Decay of VBM in different time periods.

As shown in Figure 3-14, three different time periods of the VBM response are chosen to evaluate the damping ratio. The results are shown in Table 3-6. The damping ratio in each period is less than 1% and behaves continuously (all around 0.5%). The time integration factors used in FEA simulation produces a damping ratio within a reasonable range.

Table 3-6 Damping ratio in different time periods

Period	x_i	x_{i+n}	n	ζ
I	9.41	5.76	14	0.56%

II	5.55	3.23	19	0.45%
III	3.02	0.77	49	0.44%
I-III	9.41	0.77	82	0.49%

3.4.2 Numerical Hammering Test under Dry Condition

At the time instant of 2s, the external force is applied and the structure starts vibrating. Figure 3-15 (b) shows the frequency components of VBM, which are detected by Fast Fourier Transform (FFT) analysis of the measured VBM time series at the midship (see Figure 3-15 (a)). The dry natural frequency of the Barge ship can be estimated at 10.5 Hz.

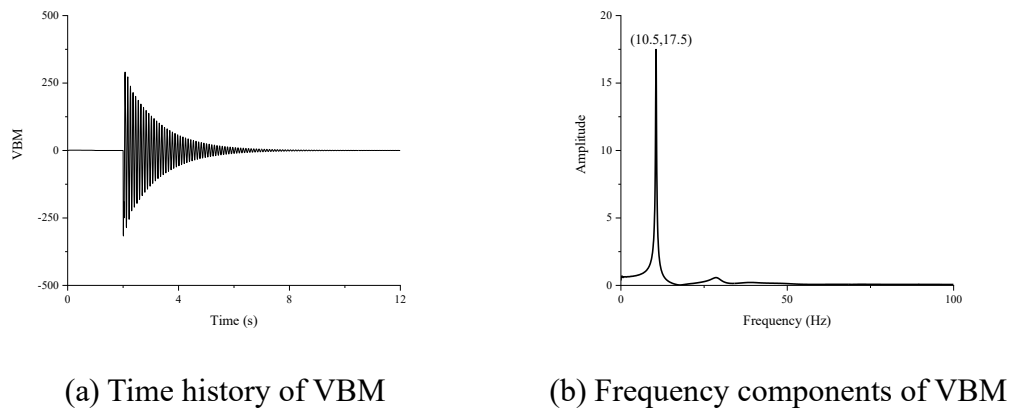
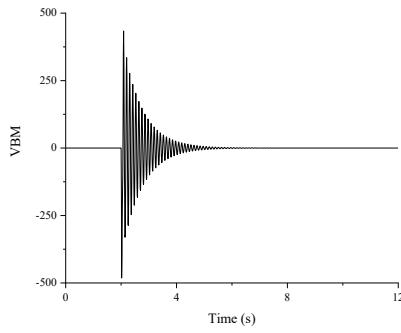
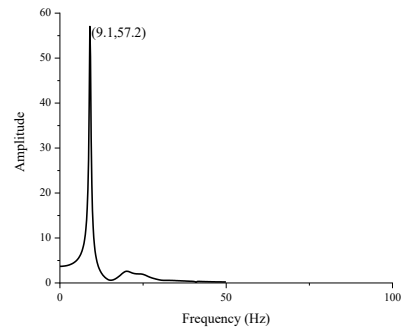


Figure 3-15 Time history and frequency components of VBM (Barge (A)), dry condition

Figure 3-16 (a) shows the time history of VBM at the midship of the KVLCC2 ship. Figure 3-16 (b) is the frequency components of VBM, which are also detected by FFT. The dry natural frequency of the KVLCC2 ship can be estimated at 9.1 Hz.



(a) Time history of VBM



(b) Frequency components of VBM

Figure 3-16 Time history and frequency components of VBM (KVLCC2), dry condition

3.4.3 Numerical Hammering Test under Wet Condition

The hammering test under wet condition was performed using two-way coupling between CFD and FEA model. This is an effective method to measure the wet natural frequency of the FE model [37]. The fluid domain used for the hammering test is shown in Figure 3-17. The external force is applied after confirming the calm water condition. Figure 3-18 and Figure 3-19 show the time history and frequency components of VBM for the Barge and KVLCC2 ship, respectively. The frequency components of VBM are detected by applying the FFT method to the time history of VBM, as in the wet condition.

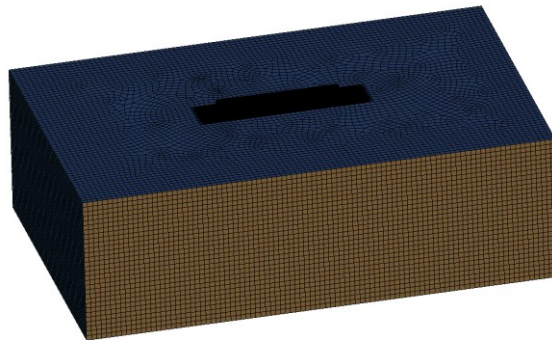


Figure 3-17 Fluid domain for hammering test under wet condition

From Figure 3-18 (b) and Figure 3-19 (b), the wet natural frequency of the Barge

and KVLCC2 ship structure can be estimated at 7.83 Hz and 7.22 Hz. Compared to the dry natural frequency, the wet natural frequency decreases due to the added mass effect of the fluid.

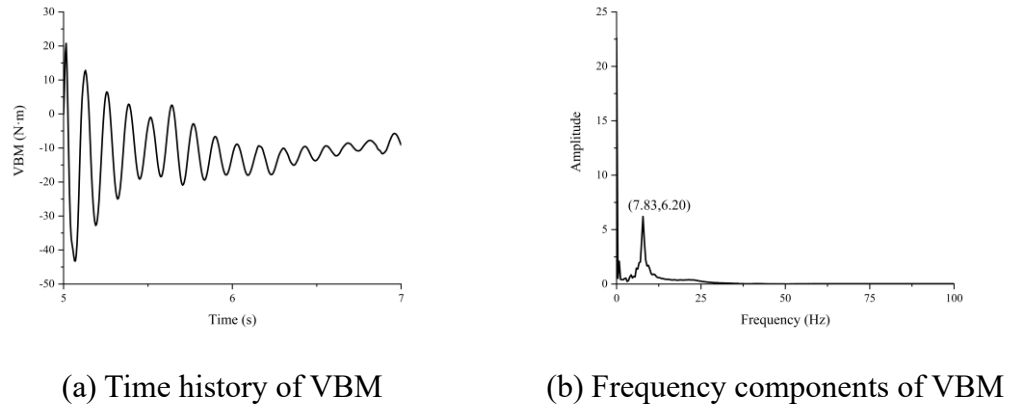


Figure 3-18 Time history and frequency components of VBM (Barge (A)), wet condition

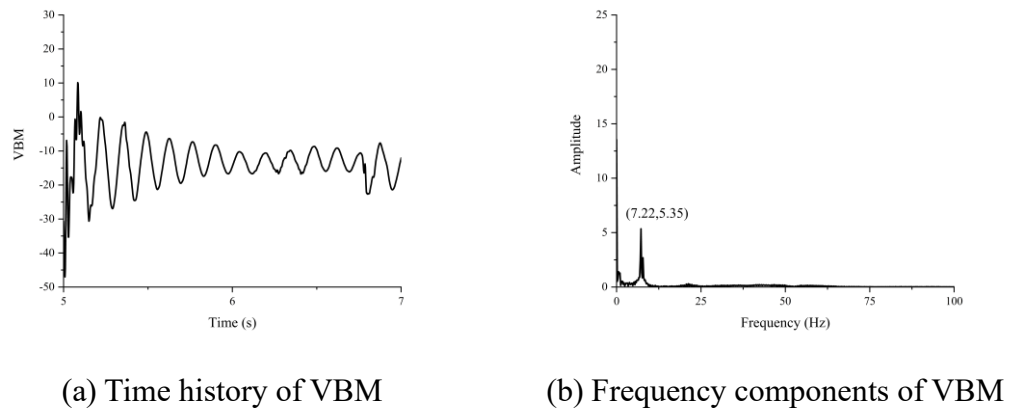
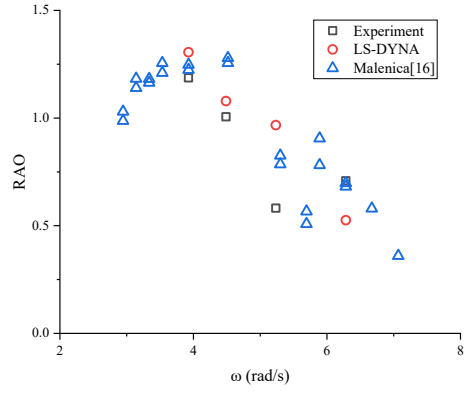


Figure 3-19 Time history and frequency components of VBM (KVLCC2), wet condition

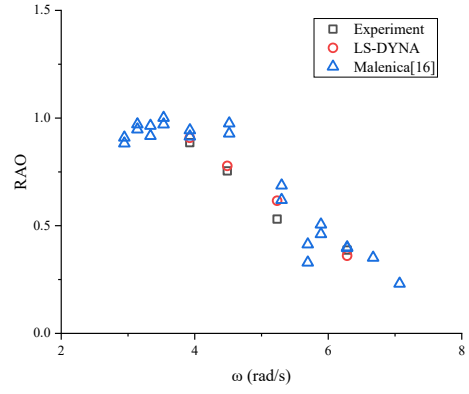
3.5 Validation

To validate the experimental and numerical model used in this study, the hydroelastic problem of a Barge ship is investigated based on Malenica's model

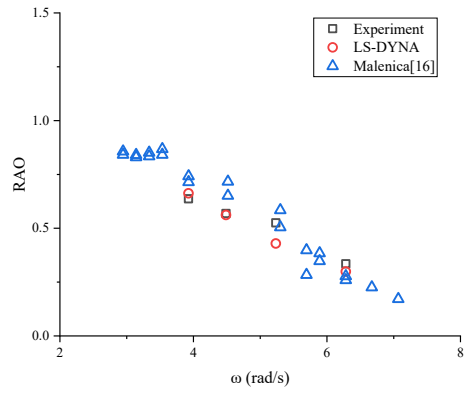
experiment [16], using the EX-A wave conditions for the validation.



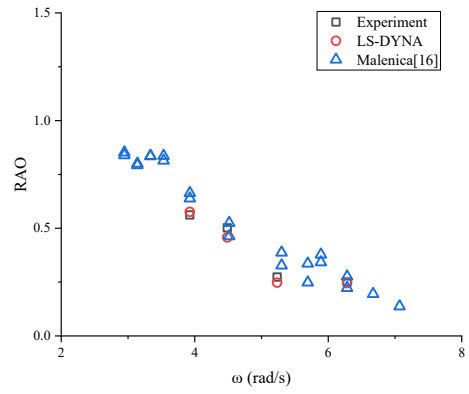
(a) P1



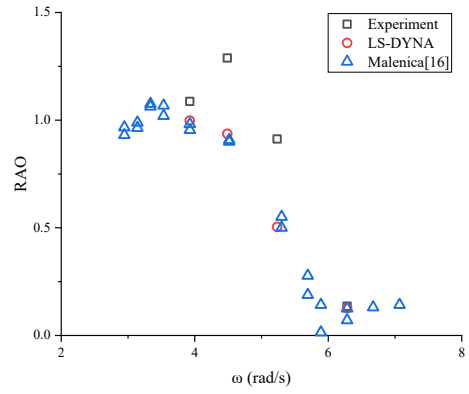
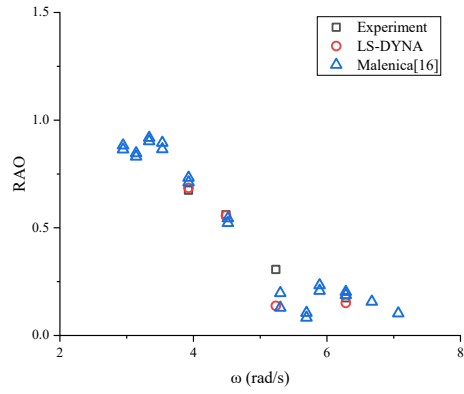
(b) P3



(c) P5



(d) P7

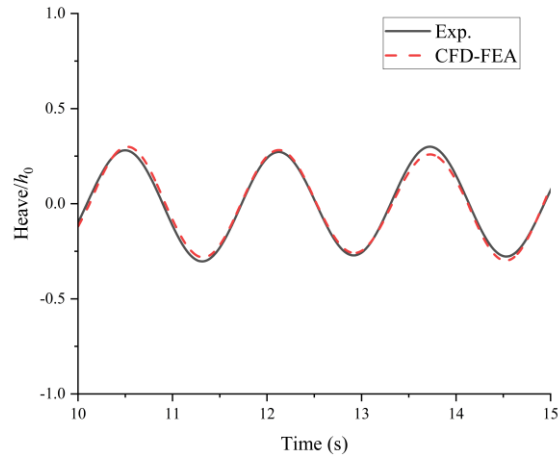
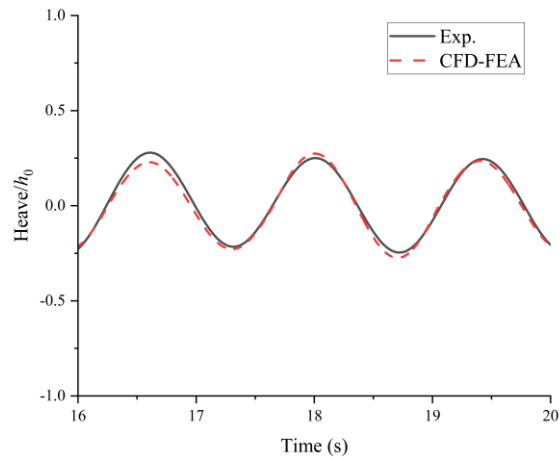


(e) P9

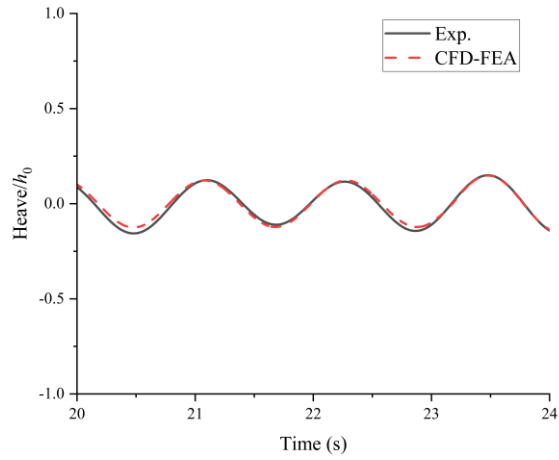
(f) P11

Figure 3-20 Comparison of the heave RAOs (EX-A, Barge (A))

Figure 3-20 shows the heave RAOs at each measuring point of the Barge (A) under different wave angular frequencies (ω). The black squares, blue triangles, and red circles represent experiment results, Malenica's results, and the CFD-FEA coupled simulation results by framework A, respectively. It can be observed from Figure 3-20 that the experiment results and the CFD-FEA coupled simulation results both show good agreement with Malenica's results.

(a) A1 ($\lambda=3.996\text{m}$)

(b) A2 ($\lambda=3.063\text{m}$)



(c) A3 ($\lambda=2.248\text{m}$)

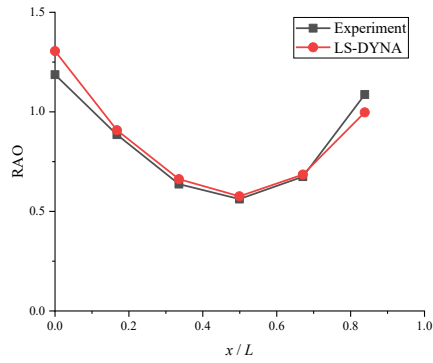
Figure 3-21 Time history of heave (EX-A, Barge (A))

Figure 3-21 compares the time history of heave motion between the experiment and the CFD-FEA coupled simulation. The experimental results are in good agreement with the simulation results. The experimental and numerical methods in this study have been validated.

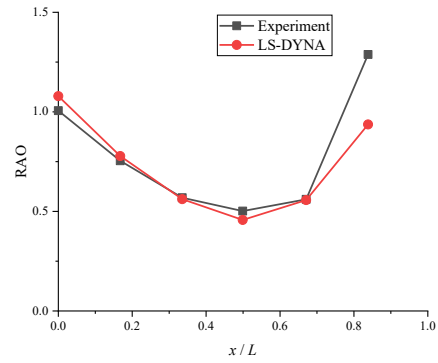
3.6 Results and Discussion

3.6.1 Motion RAOs

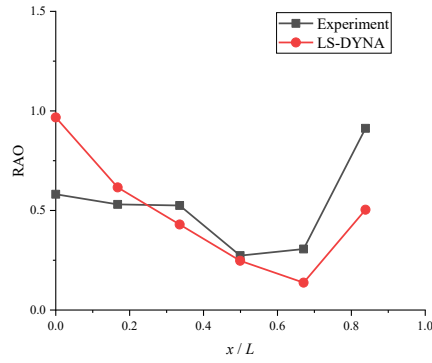
The comparisons of z -displacement RAOs of the Barge (A) between the experiment and numerical results are shown in Figure 3-22. It can be known from Figure 3-22 that the vertical motion response at the bow is the largest; in Figure 3-22 (a), (b) and (c), from bow and stern to midship, the vertical motion response gradually reduces. In the range of $\lambda/L_{pp} = 1.0$ to $\lambda/L_{pp} = 1.6$, the vertical motion response has a trend of increase with the increase of λ .



(a) A1 ($\lambda/L_{pp} = 1.6$)



(b) A2 ($\lambda/L_{pp} = 1.2$)

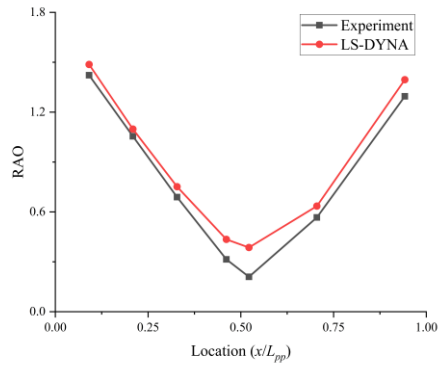


(c) A3 ($\lambda/L_{pp} = 1.0$)

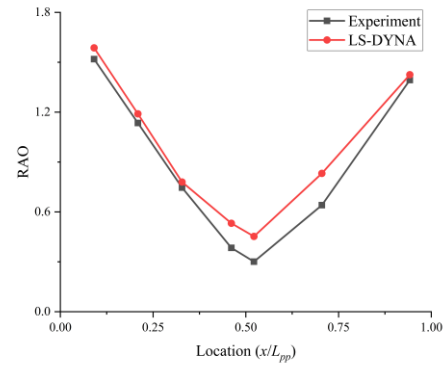
Figure 3-22 The z -displacement RAOs under different wave conditions (Barge (A))

Figure 3-23 shows the z -displacement RAOs of the KVLCC2, comparing the experimental results with those of the numerical simulation. The numerical results show good agreement with the experimental results, except for the midship region. At the midship, the numerical result in every case is obviously less than the experimental result. The difference between the numerical model (continuous model) and the experimental model (segmented model) is the reason for this. For the continuous model, all parts of the ship move together, and both sides of the ship will have a larger influence on the movement at the midship. Hence, this difference between numerical and experimental

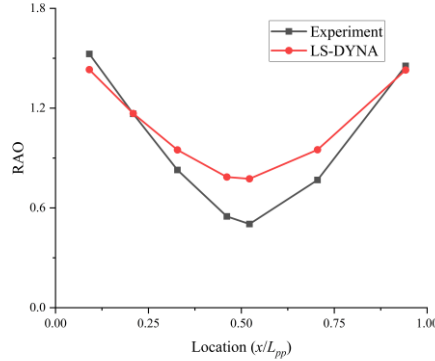
results at midship is reasonable. From Figure 3-23, the minimum heave RAO can be observed around the midship. The displacement RAO increases from the midship to the bow and stern. The vibration mode of the ship is primarily modeled as a 2-node mode. As the wavelength increases (from C3 to C5), the vertical motion response exhibits a trend of increment with the increase in λ .



(a) C3 ($\lambda/L_{pp} = 1.0$)



(b) C4 ($\lambda/L_{pp} = 1.2$)



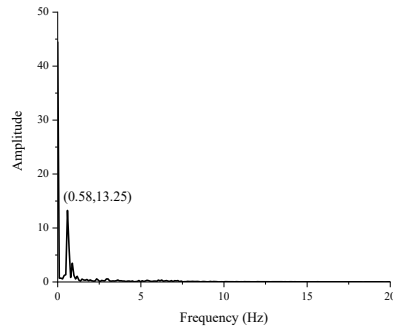
(c) C5 ($\lambda/L_{pp} = 1.5$)

Figure 3-23 The z -displacement RAOs under different wave conditions (KVLCC2)

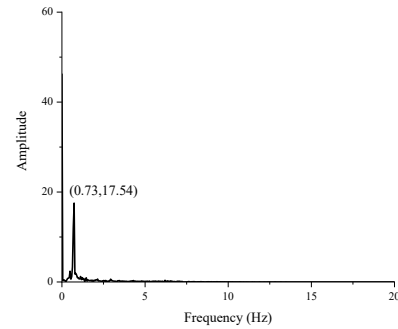
3.6.2 Wave-induced VBM

As wave-induced VBM typically contains wave frequency (WF) components and high-frequency (HF) components, the FFT is used to transform time history results into frequency components. Figure 3-24 shows the frequency components of VBM under

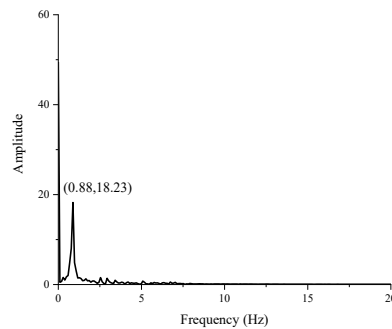
typical wave conditions for the Barge ship. The structure is primarily excited by the wave frequency, and the WF component matches well with the frequency of the incident wave.



(a) A1 ($\lambda/L_{pp} = 1.6$)



(b) A2 ($\lambda/L_{pp} = 1.2$)



(c) A3 ($\lambda/L_{pp} = 1.0$)

Figure 3-24 Frequency components of VBM under typical wave conditions (Barge
(A))

The frequency components of VBM of the KVLCC2 under the C3 condition are shown in Figure 3-25. It is clear that the WF component, as well as the higher-order harmonics, are observed. The peak of the WF and 2-node high frequency (HF) component is higher than other harmonic components.

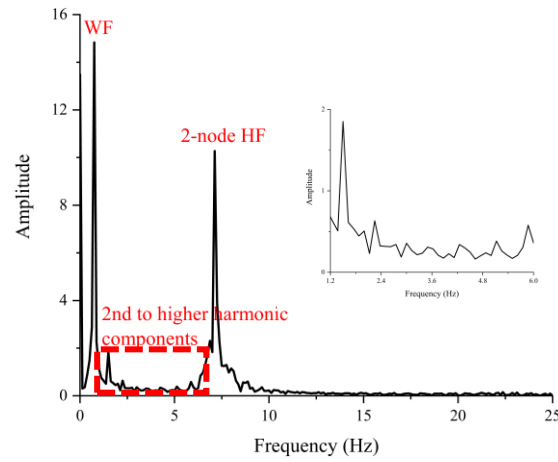


Figure 3-25 Frequency components of VBM at $\lambda/L_{pp} = 1.0$ (KVLCC2).

A FFT band-pass filter is used to divide the VBM into different frequency components, including the WF and higher springing harmonic orders. Some of these different frequency components are shown in Figure 3-26. The frequency range of each component is also shown in the figure. All the frequency components in Figure 3-26 have good agreement with the sinusoidal shape. The WF and 2-node HF responses are significantly higher than other harmonic components.

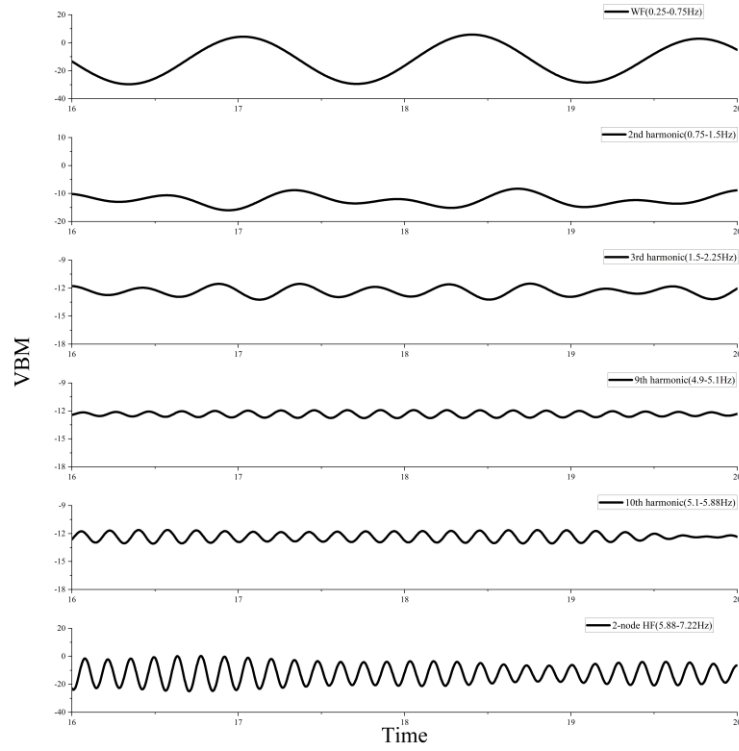


Figure 3-26 Time history of VBM with different frequencies at $\lambda/L_{pp} = 1.0$ (KVLCC2)

As discussed in the previous content, the wave-induced VBM typically contains both the WF and HF components. However, HF components are not the focus of this study. The structure's response to the wave frequency will be discussed. Hence, the FFT filter is adopted to obtain the WF components in VBM. The cut-off frequency used here is 1.1 Hz.

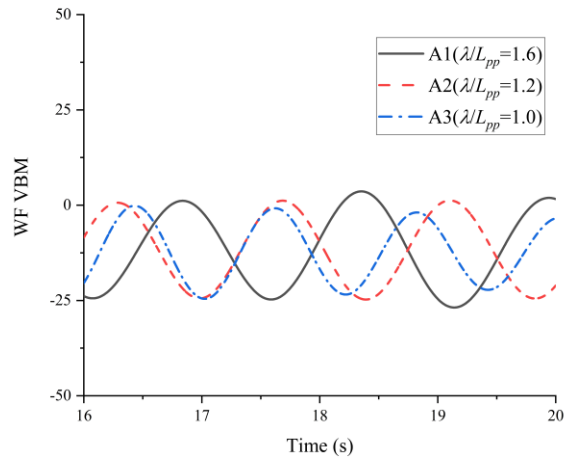


Figure 3-27 Comparison of VBM at the midship under typical wave conditions
(Barge (A))

Figure 3-27 illustrates a comparison of VBM at the middle of the Barge ship among three typical wave conditions. The black, red, and blue curves represent A1 - A3 wave conditions, respectively. As the wave height of each wave condition remains constant, the VBMs caused by different wave conditions are within a similar range. However, it can be seen from Figure 3-27 that the VBM has a trend of decreasing with the decrement of wavelength.

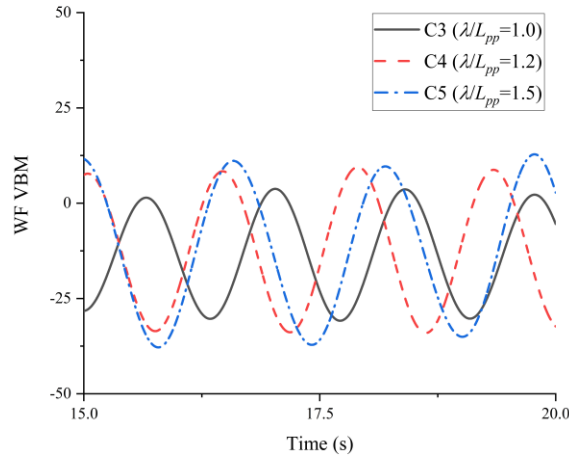


Figure 3-28 Comparison of VBM at midship under typical wave conditions
(KVLCC2)

The time history of VBM at the middle of the KVLCC2 ship is shown in Figure 3-28 under typical wave conditions. The black curve, red curve and blue curve represent C3 - C5 wave conditions, respectively. Similar to the result of the Barge ship, the VBMs caused by different wave conditions are within a similar range. The same trend of VBM can also be observed from Figure 3-28.

3.7 Conclusions

This chapter investigates the hydroelastic response of the Barge (A) and KVLCC2 model under different wave conditions using experimental analysis and strongly coupled two-way CFD-FEA method. By comparing numerical prediction results and tank test results of the Barge ship with published results, the experimental analysis and numerical methods were validated. By analyzing motion and VBM of the ship, conclusions can be summarized as follows:

(1) The strongly coupled two-way CFD-FEA method used in this study shows good accuracy in simulating ship-wave interaction problems. Nonlinearity combined

with both fluid and structure can be calculated with high stability using this method.

(2) The response of vertical displacement RAOs indicates that the ship structure is primarily in 2-node mode under various wave conditions in this study. At the midship, the simulation results of the KVLCC2 ship are larger than the experimental results due to differences in the models used, specifically the continuous and segmented model.

(3) The structural response is discussed in terms of midship VBM. The VBM has a tendency to decrease with decreasing wavelength. Meanwhile, the high-order harmonics of VBM response are observed through the FFT analysis. Further investigation on this is needed.

In the future, investigation on hydroelastic response of ship with various forward speeds and heading angles under regular and irregular waves using the CFD-FEA coupled method will be meaningful and interesting. Meanwhile, the higher-order components observed in the CFD-FEA coupled simulation will be further investigated in the next chapter.

Chapter 4

Higher-order Springing of a Segmented Ship

4.1 Introduction

In this chapter, the higher-order springing phenomenon is addressed for a segmented barge ship model through experimental and numerical measures. An efficient in-house two-way coupled numerical solver (framework B) between CFD-FEA is developed and validated against experimental results. The coupling method is based on a domain-separated approach and necessitates the resolution of individual boundary value problems in distinct domains. To ensure convergence within these individual domains, an implicit numerical scheme is employed and further facilitated exchange of variables for coupling. The current approach emphasizes the overall convergence between two solvers, maintaining a strongly coupled setup to comprehensively address fluid-structure interaction phenomena, including added mass and damping effects. A series of tank tests were conducted first to measure the wave-induced sectional loads and motions, during which the springing responses to very high-order harmonics of wave load were observed. By comparing the numerical prediction with the tank test results for rigid body motion and flexible vertical bending moment (VBM), the proposed numerical method demonstrated agreement with experimental results, affirming its validity and robustness. Finally, the springing response up to 14th order harmonics is discussed and investigated.

As a follow-up to the work by Pal and Iijima [134], in which a one-way coupled model was adopted to reproduce the wave-induced vibrations, the objectives of the present study can be summarized as follows:

- I. The springing vibrations are focused and clarified mechanically. The results are validated with a high-fidelity numerical solver.
- II. Two-way coupled code is developed further from the original work by Takami et

al. [32] for better stability and accuracy.

To this end, the springing case is preselected by monitoring the wave conditions in a towing tank, while the whipping case is excluded from the scope, although it is difficult to clearly distinguish springing from whipping vibrations in full-scale measurements.

4.2 Tank Test

The experiment was conducted using a backbone model, named Barge (B) in the University of Osaka towing tank. The particulars of the towing tank are: 100m in length, 7.8m in width, and 4.3m in depth.

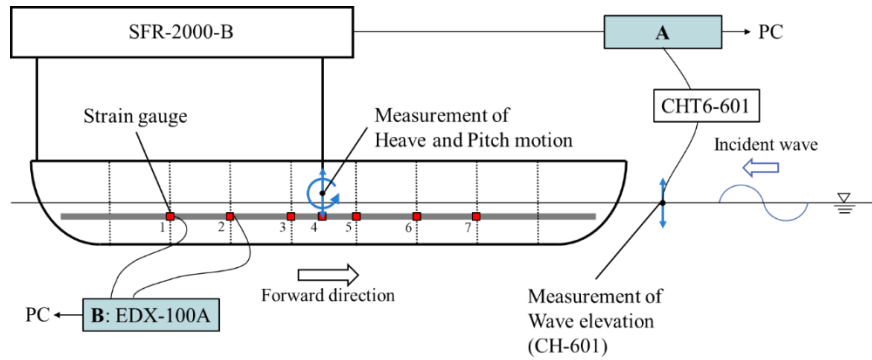


Figure 4-1 Experimental setup, conducted at the University of Osaka .

Figure 4-1 illustrates the detailed experimental setup. Cases with and without forward speed were conducted in the experiment. However, it should be noted that only cases without forward speed were chosen for this research to avoid any slamming. A segmented barge ship model with an aluminum backbone was used to conduct the experiment. The length, width, and depth of the ship hull are 2.7 m, 0.45 m, and 0.27 m, respectively. The body plan of the hull is shown in Figure 4-2.

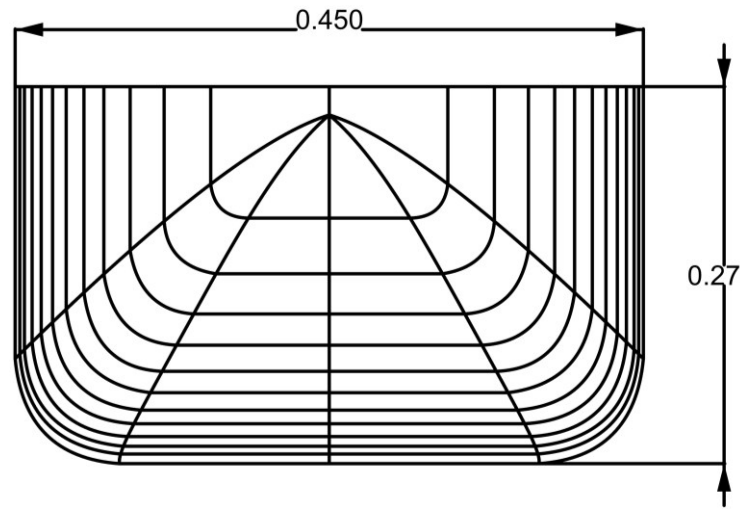


Figure 4-2 Body plan of the ship hull of Barge (B).

Table 4-1 Principle particulars of the experiment model

Length (L)	2.7 m
Breadth (B)	0.45 m
Depth (D)	0.27 m
Draft (d)	0.12 m
Weight	1239.5 N
Center of Gravity (OG)	0.178 m
Radius of Gyration (Kyy)	0.676 m
Modulus of Elasticity of backbone(E)	60.8 GPa
Modulus of Elasticity of hull (E)	63.5 GPa

The test model is a 1:100 scaled model of the original one. Table 4-1 lists the principal particulars of the experiment model. From bow to stern, there are a total of nine segments rigidly connected to the backbone with bolts (see Figure 4-3). Waterproof

elastic rubber is used to seal the gaps between the segments to prevent water from entering during the experiment. The total weight of the model used is 1239.5 N. The desired draft was achieved by putting external weight plates uniformly over the hull. The mass distribution of the ship along the longitude direction is shown in Figure 4-4.

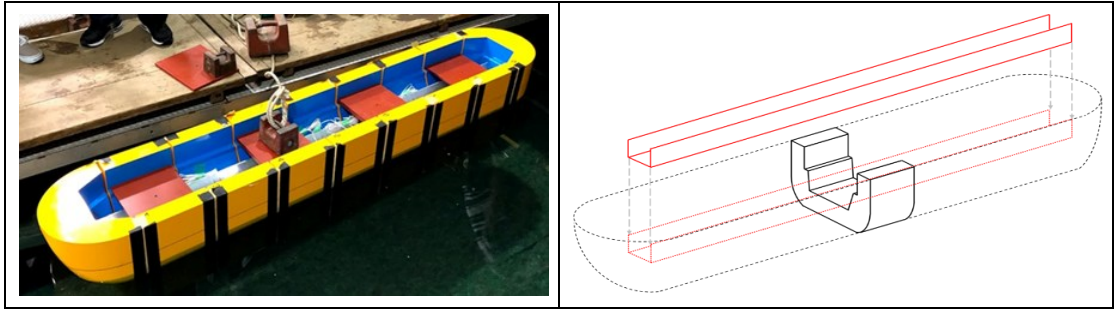


Figure 4-3 Backbone model with experimental setup, conducted at the University of Osaka.

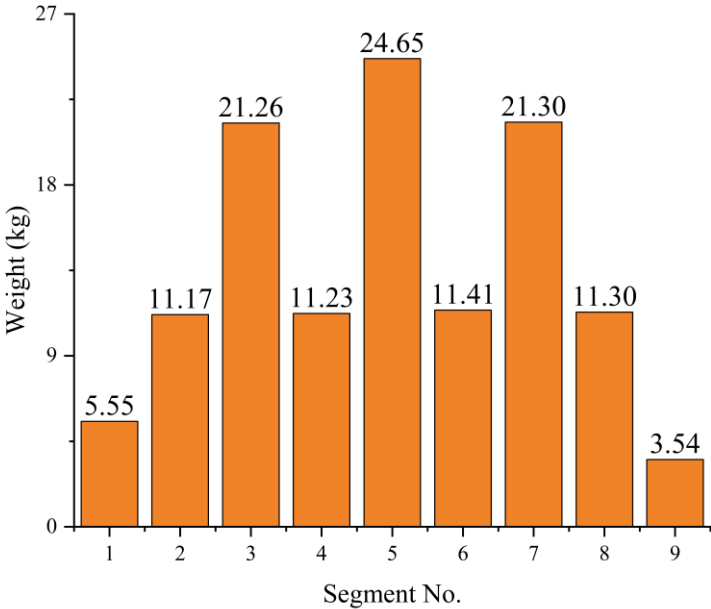


Figure 4-4 Mass distribution of the ship along longitude direction.

The backbone of the ship was specially designed so that there are no unintended forces generated by other hull components during movement in waves. Particularly, it should be noted that the fore and aft segments were comparatively light and firmly connected to the backbone to ensure stability. The backbone's height, width, and thickness are 50 mm, 75 mm, and 5 mm, respectively. Strain gauges are affixed to the backbone at the mid-location of two segments to accurately record strains caused by the motion of the ship. Figure 4-5 shows the placement and particulars of strain gauges. The motion of the hull is measured by attaching a gimbal to the mid-segment, which is depicted in Figure 4-1.

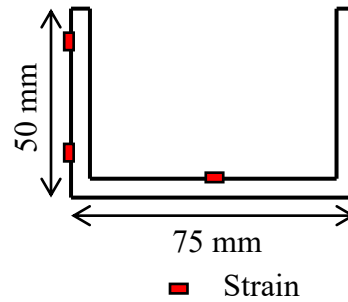


Figure 4-5 Location of strain gauges on the backbone of Barge (B).

The wave conditions of the experiment are shown in Table 4-2. In Table 4-2, λ , L , A_w , and F_n represent wavelength, ship length, wave amplitude, and Froude number, respectively. In this research, ship under different regular head waves without forward speed are investigated. The wave height of all test conditions is designed to be relatively small to prevent slamming during the experiment. Wave dissipators are used, and a significant amount of time is given to calm the water in the tank before a new test is conducted.

Table 4-2 Experiment wave conditions

Test Conditions	λ/L	A_w (m)	F_n
I	0.6	0.025	0.0
II	0.8	0.025	0.0
III	1.0	0.025	0.0
IV	1.1	0.025	0.0
V	1.2	0.025	0.0
VI	1.5	0.025	0.0
VII	2.0	0.025	0.0
VIII	2.5	0.025	0.0

4.3 Numerical Model

4.3.1 FEA Model

The dynamic implicit commercial solver based on LS-DYNA is adopted to calculate the structure response. Figure 4-6 shows an overview of the FEA model. The U-shaped backbone is discretized with 8-node solid elements, and the ship hull is modeled with 4-node shell elements. The mesh sizes of the backbone solid elements and the ship hull shell elements are 0.002 m and 0.005 m respectively. Along the thickness direction of the backbone, there are a total 3 layers of solid elements to calculate strain and stress accurately. The total number of elements is 0.7 million in the FEA model.

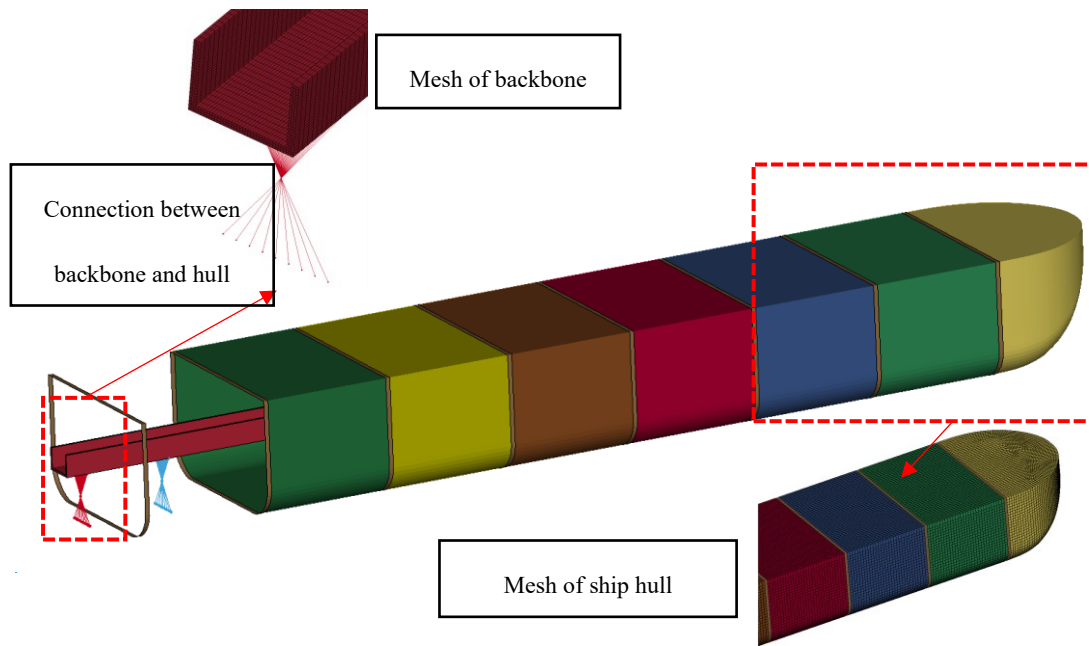
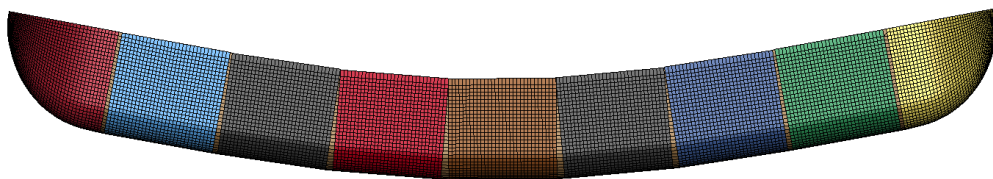
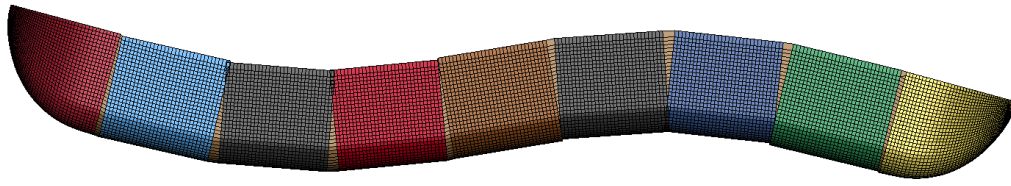


Figure 4-6 Overview of the FEA model.

The ship hull and the backbone are connected rigidly by a nodal rigid body constraint. It should be noted that in the experiment, the backbone and the ship hull are connected by constraint units and have physical contacts. However, in the FEA model, the connection is simplified to increase the stability and efficiency of the simulation. As for rigid body motion, only the heave and pitch of the ship are released in the simulation. As for the time integral constants, the Bath scheme, which has been discussed in Chapter 2, is adopted in this chapter. As shown in Figure 4-7, the natural frequencies of 2-node and 3-node vibration modes in dry conditions are found to be 9.5 Hz and 23.1 Hz, respectively, by the eigenvalue test.



(a) 2-node vibration mode of the FE model (9.5Hz)



(b) 3-node vibration mode of the FE model (23.1Hz)

Figure 4-7 Typical vibration mode of the FE model (Barge (B)).

4.3.2 CFD Model

The finite volume method (FVM) implemented in a commercial CFD solver, STAR-CCM+, is applied in the CFD calculation phase. Numerical regular wave is generated by prescribing the velocity and the free surface elevation at the velocity inlet boundary. The wave model at the inlet is based on the fifth-order Stokes wave. The calculation setup of CFD is shown in Table 4-3.

Table 4-3 Calculation setup of CFD

Multiphase	VOF
Physical model of water/air	Incompressible
Turbulence model	SST K- ω
Solution	SIMPLE unsteady implicit analysis

The overall CFD domain at cross-section $Y = 0$ is shown in Figure 4-8. The numerical tank is $5L$ in length, $3L$ in width, and 5 m in depth, which is large enough to avoid boundary reflections. The wave inlet boundary and bottom of the fluid domain are set to be velocity inlet conditions, while wave outlet boundary and top of the fluid domain are set to be pressure outlet conditions. The side boundary of the fluid domain is set to a symmetric condition to simulate the far field boundary condition. The ship

hull is set as a no-slip wall, where the fluid tangential velocity is always zero.

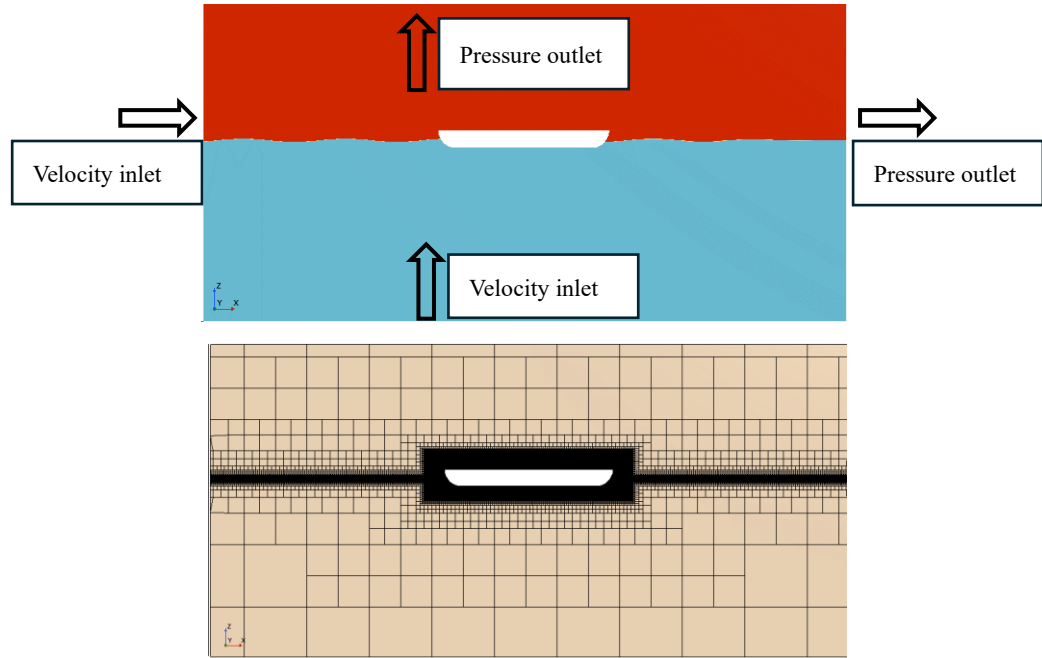


Figure 4-8 Overview of CFD domain (Barge (B)).

The mesh size at the free surface zone and around the ship hull is refined to capture the VOF wave and FSI load properly. The mesh size at the free surface zone is shown in Table 4-4, where dx , dy , and dz represent mesh sizes along the X , Y , and Z directions, respectively. In this research, a number of 80 cells per wavelength and 20 cells per wave height is adopted along the longitude (X) and vertical (Z) directions, which is recommended by ITTC [133]. To reduce the computational cost without losing the accuracy of the results, the mesh size along the Y direction is a bit larger, which is 20 cells per ship width. The mesh size around the ship hull is the finest over the whole fluid domain to calculate the FSI load on the hull with high accuracy. The minimum size of mesh on the ship hull is $\lambda/240$. Eight prism layers, together with wall functions, were used to accurately resolve the wall shear stress profile. Considering the balance

between cost and accuracy of the simulation, the maximum y^+ value is controlled within 5, avoiding the buffer layer. A total of 6.2 million cells are generated in the fluid domain.

Table 4-4 Mesh size at free surface zone

dx	dy	dz
$\lambda/80$	$B/20$	$A_w/20$

To avoid the reflection of waves from boundaries or the free-surface disturbance due to ship motion, the VOF wave forcing and damping algorithm (included in Star-CCM+) is used at the boundaries. In the VOF wave forcing and damping algorithm, the solution of the discretized Navier-Stokes equations is forced towards another solution (usually an analytical or simplified numerical solution) over a specified distance [135]. This reduces the computing effort by using a reduced-size solution domain.

4.3.3 CFD-FEA Coupling

Framework B discussed in section 2.3.5 is adopted in this chapter. The adopted accelerator and predictor are summarised in Table 4-5. It is well known that the Aitken's accelerator works properly if the target numerical sequence has a solution towards convergence. Although the number of sub-iterations could be decreased by using Aitken's accelerator and predictor, the coupling progress typically needs 7-20 sub-iterations before converging at each time step. The computational cost is still high. In this chapter, it took 7-11 days per 5s of the simulation with 18-core CPU. Out of this, 90% of the total time is taken up by CFD, 8% by FEA, and 2% by the coupling progress in this research.

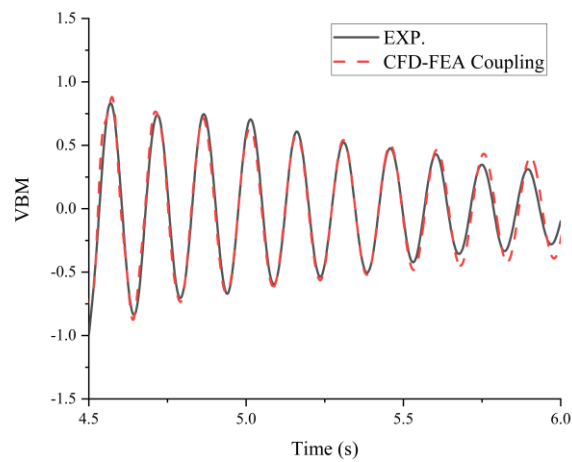
Table 4-5 Adopted accelerator and predictor, Barge (B)

Accelerator	Aitken
Predictor	3-step Adams-Bashforth

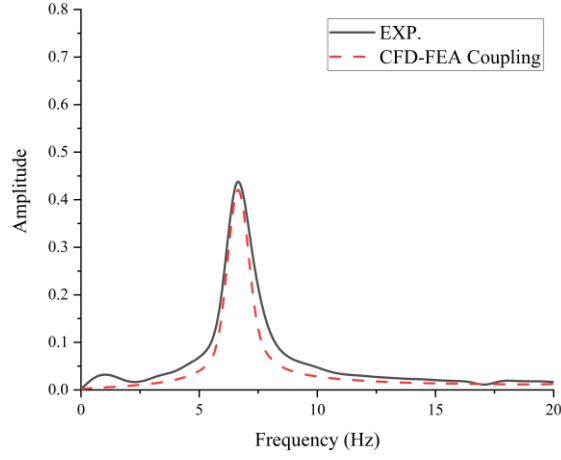
4.4 Results and Discussion

4.4.1 Harmering Test

The natural frequency of the 2-node vertical vibration mode of the experimental and numerical models was measured both by hammering tests under wet conditions. An impulse load was applied to the ship after confirming the calm water condition. Natural frequency was detected by Fast Fourier Transform (FFT) analysis of measured VBM time series at the mid-ship section of the backbone.



(a) Normalized VBM time history



(b) FFT of normalized VBM time history

Figure 4-9 Hammering test results under wet condition, Barge (B).

Figure 4-9 shows the time series and frequency characteristics of VBM by the experiment and the CFD-FEA coupled simulation. From Figure 4-9 (b), the wet natural frequency of the experimental and numerical models could be estimated at around 6.8 Hz. There is a slight difference in the wet natural frequency of these two models, which is reasonable as the FEA model is simplified compared with the experimental model. Compared with the dry natural frequency (9.8 Hz), the wet natural frequency decreases because of the added mass effect. The added mass could be defined as follow:

$$\omega_{\text{wet}} = \sqrt{\frac{1}{1+f_{\text{added mass}}}} \omega_{\text{dry}} \quad (4-1)$$

where ω_{dry} and ω_{wet} represent the dry and wet natural frequencies, respectively; $f_{\text{added mass}}$ is the factor to consider the added mass effect.

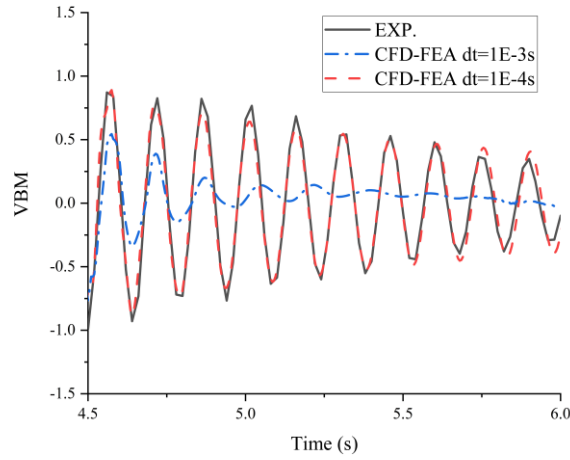


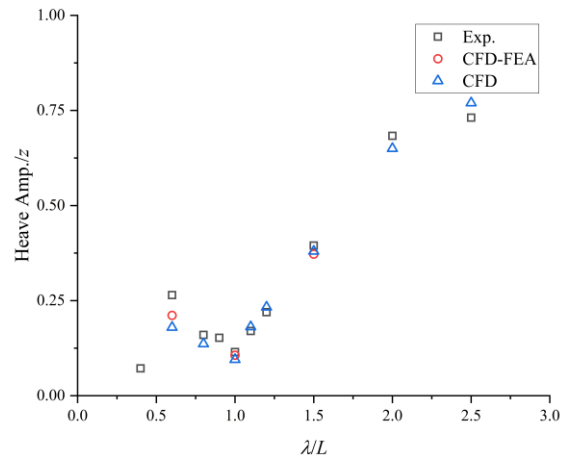
Figure 4-10 Comparison of the normalized VBM time history of hammering test, Barge (B).

It should be noticed that the time step size used in CFD-FEA coupled simulation should be carefully considered. Figure 4-10 shows a comparison of CFD-FEA coupled simulation results with different time step sizes. The numerical damping effect from the sub-interactive progress could be significant if the time step size is not chosen properly for the coupled progress. It should be noted that higher order springing is difficult to excite and usually weak (see Figure 4-15 in the later part of this chapter). Therefore, large numerical damping might suppress the phenomenon. In this research, by balancing the cost and accuracy of calculation, the time step size of CFD-FEA coupled simulation is set to 1E-4s from the Figure 4-10.

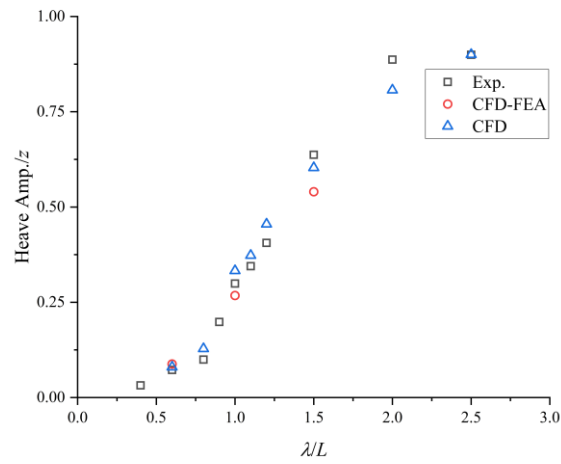
4.4.2 Comparison of ship motion

Firstly, the validation of the basic responses is done prior to investigating the springing response. The ship motion in the experiment and CFD-FEA coupled simulation is compared for heave and pitch Response Amplitude Operators (RAOs) for the different wave conditions. The amplitude of the heave and pitch of the ship are normalized by the wave amplitude ζ and the wave number k . In Figure 4-11, “CFD-FEA” means the motion from the two-way coupled analysis, whereas “CFD” means

rigid body CFD motion only. It could be confirmed that the simulation results closely match the experimental results under all wave conditions.



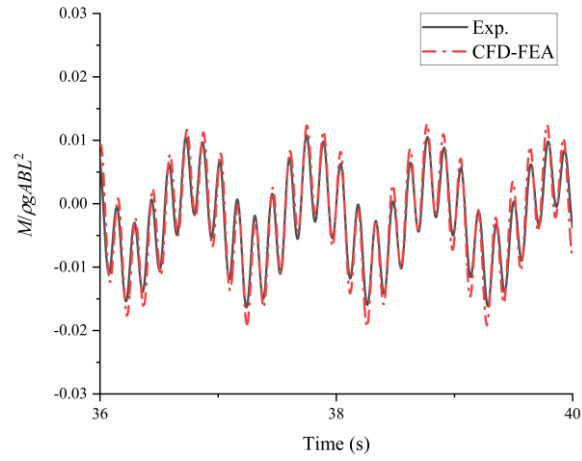
(a) Heave



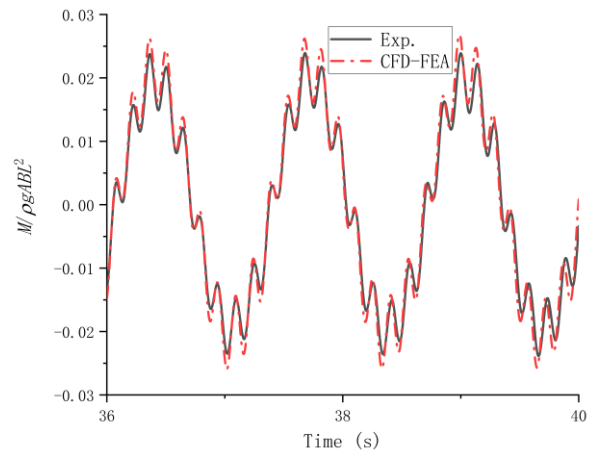
(b) Pitch

Figure 4-11 Comparison of ship motion RAOs, Barge (B).

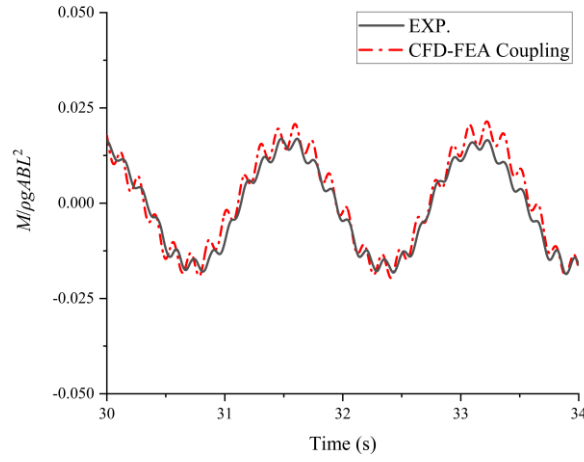
4.4.3 Comparison of VBM



(a) Test case I ($\lambda/L=0.6$)



(b) Test case III ($\lambda/L=1.0$)



(c) Test case VI ($\lambda/L=1.5$)

Figure 4-12 Comparison of normalized VBM time history under different wave conditions, Barge (B).

Figure 4-12 shows the time history of VBM from CFD-FEA coupled simulation and was validated against experimental results for three wave conditions only, excluding the repetitiveness in this article (see Table 4-2). The VBM in all the cases is measured at the midship section. It was carefully observed that the slamming was not present in any of the wave conditions mentioned in Table 4-2 during the experiment (see Figure 4-13). The springing vibration initiates after enough time has elapsed in the experiment. Hence, the time series window is tailored to the specific case when the comparison is made between experimental and CFD-FEA coupled VBM responses. The same can be seen in Figure 4-12 (a), (b), and (c). The full experimental VBM results can be found in the previous study [136]. It can be confirmed from Figure 4-12 (a), (b), and (c) that although the amplitude of the vibration is almost the same, a little difference in excitation frequencies results in small phase difference between experimental and simulation results.

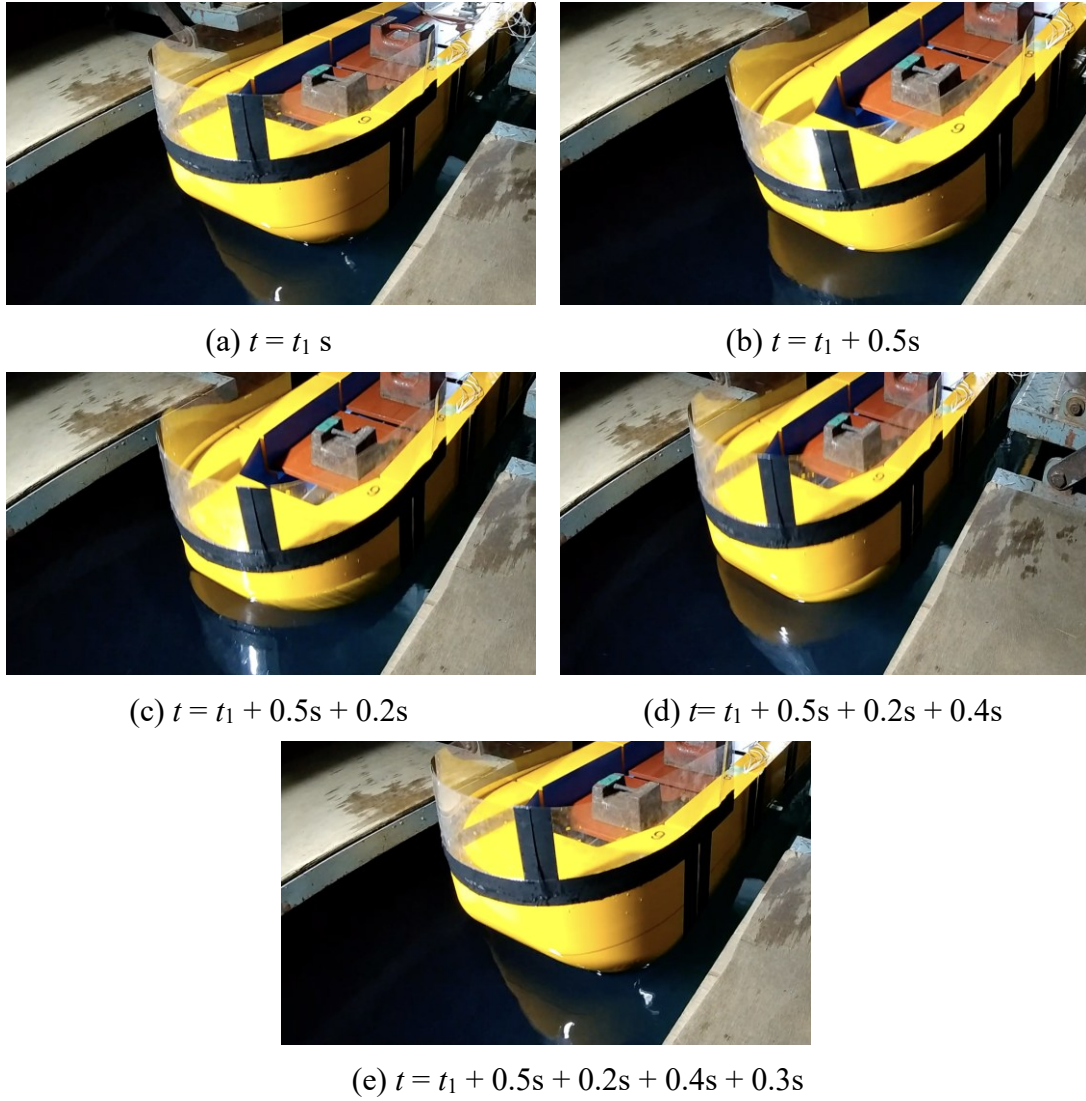


Figure 4-13 Smooth transition of wave profile at bow due to wave condition III
 $(\lambda/L=1.0)$, Barge (B).

4.4.4 Higher-order components

To investigate the springing phenomenon, the harmonic orders N are introduced:

$$\omega_n = \omega_e N \quad (4-2)$$

where ω_n is the excitation frequency of the ship and ω_e is the frequency of the encountered wave.

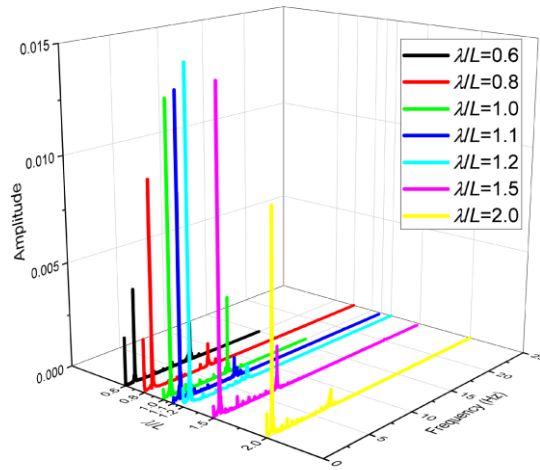


Figure 4-14 FFT analysis results of normalized experiment VBM time history, Barge (B).

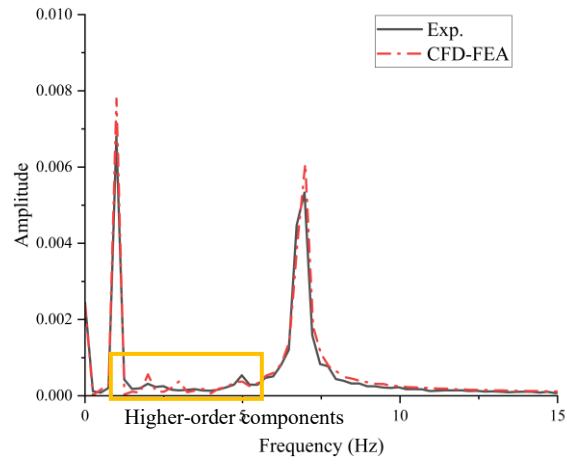
Table 4-6 shows the N of all the wave cases studied in this article. N is an almost perfect integer in all these cases and corresponds to 7th to 14th-order harmonics. The FFT analysis results of the normalized experiment VBM time history for the cases in Table 4-6 are shown in Figure 4-14. It is clear from the figure that higher-order components (the peaks) exist in the experiment.

Table 4-6 Computation of the higher order harmonics

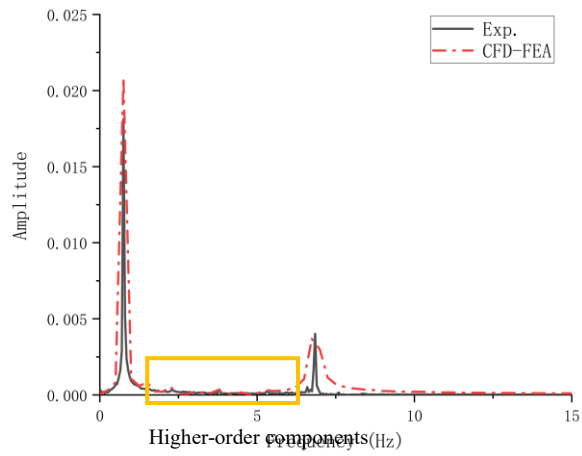
Test Conditions	Encounter Frequency (Hz)	Experimental Frequency (Hz)	Harmonic Order (N)
I	0.98	6.89	7
II	0.85	6.79	8
III	0.76	6.86	~9

IV	0.72	6.50	~9
V	0.69	6.89	10
VI	0.62	6.79	11
VII	0.54	6.99	13
VIII	0.48	6.79	14

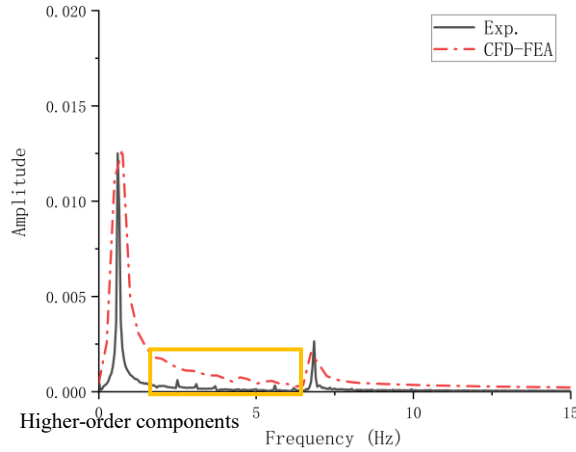
Figure 4-15 compares the FFT analysis results of normalized VBM time history between the experiment and simulation for different test cases (I, III, and VI). The FFT results of the experiment and simulation are almost the same at wave frequency (WF) and 2-node HF. Between WF and 2-node HF, there are higher-order components. These higher-order components could be observed in the FFT results but are relatively small compared with the WF peak and 2-node HF peak. It is worth noting that there is a lack of agreement for other harmonics rather than the WF and the 2-node HF peak between the experimental and simulation results. This is likely due to the relatively short simulation duration compared to the experiment. As a result, small components, such as higher-order parts other than the WF peak and 2-node HF, may be ill-defined during FFT analysis (see Figure 4-15). Hence, individual harmonics of the VBM time series are dissected and compared between experiment and numerical analysis. This is achieved by changing the frequency bandwidths of every harmonic in the bandpass filter (based on FFT and inverse FFT). The bandwidth of each harmonic is calculated based on encounter wave frequency and target harmonic order N (see Table 4-6).



(a) Test case I ($\lambda/L=0.6$)



(b) Test case III ($\lambda/L=1.0$)

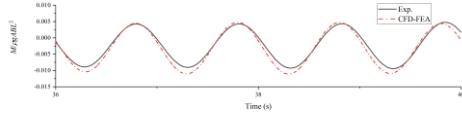


(c) Test case VI ($\lambda/L=1.5$)

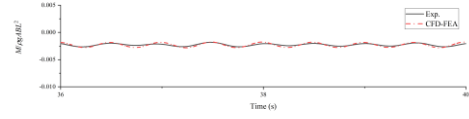
Figure 4-15 FFT analysis of normalized VBM time history under different wave conditions, Barge (B).

Figure 4-16 to Figure 4-18 show the comparison of the bandpass filter results of normalized VBM time history between the experiment and simulation for different test cases (I, III, and VI). It is noticeable that there is considerably good agreement between the experimental and simulation results at all the harmonics. Although there is a little mismatch in some harmonics, the overall trend is similar. The high-frequency components other than WF and 2-node may have been caused mostly by geometrical nonlinearity, while high-order nonlinear hydrodynamic pressure such as diffraction and radiation may have affected them a little. A similar discussion can be found in another research project by Sawada et al. [137], where the probable cause of the springing is investigated. They calculated and compared the forces on the fore segment of a backbone model of a large containership with or without variation of the wetted surface area considered and found that the nonlinear term is much larger for the calculation case with the wet surface area consideration. They concluded that the main cause of springing can be attributed to the change in the wetted surface area. When one of the high-frequency components hits the natural frequency, the springing vibrations are

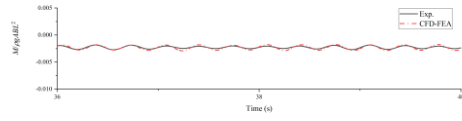
pronounced.



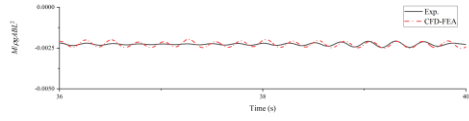
(a) WF (0.0-1.1Hz)



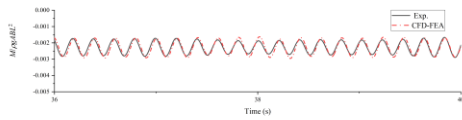
(b) 2nd order harmonics (1.1-2.1Hz)



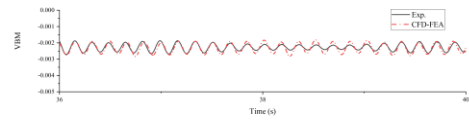
(c) 3rd order harmonics (2.1-3.1Hz)



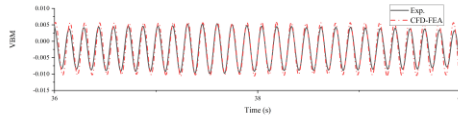
(d) 4th order harmonics (3.1-4.1Hz)



(e) 5th order harmonics (4.1-5.1Hz)

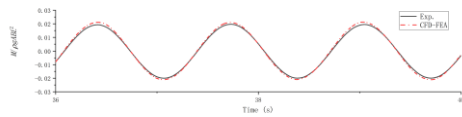


(f) 6th order harmonics (5.1-6.1Hz)

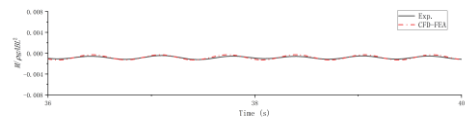


(g) 2-node HF (6.1-7.1Hz)

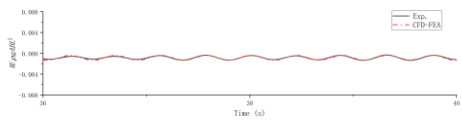
Figure 4-16 Higher-order components of VBM, test case I ($\lambda/L=0.6$), Barge (B).



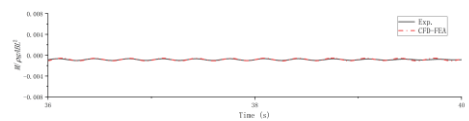
(a) WF (0.0-0.9Hz)



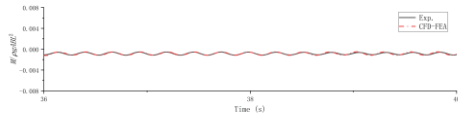
(b) 2nd order harmonics (0.9-1.7Hz)



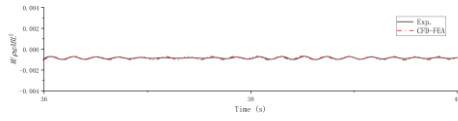
(c) 3rd order harmonics (1.7-2.5Hz)



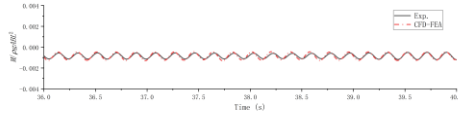
(d) 4th order harmonics (2.5-3.3Hz)



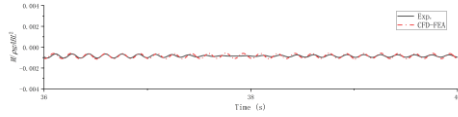
(e) 5th order harmonics (3.3-4.1Hz)



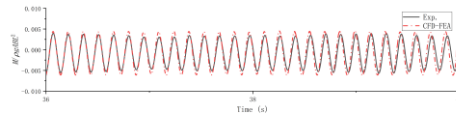
(f) 6th order harmonics (4.1-4.9Hz)



(g) 7th order harmonics (4.9-5.7Hz)

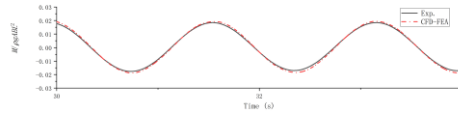


(h) 8th order harmonics (5.7-6.5Hz)

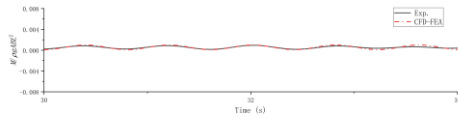


(i) 2-node HF (6.5-7.3Hz)

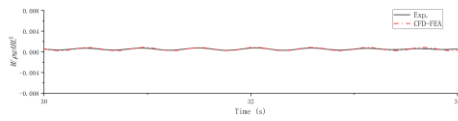
Figure 4-17 Higher-order components of VBM, test case III ($\lambda/L=1.0$), Barge (B).



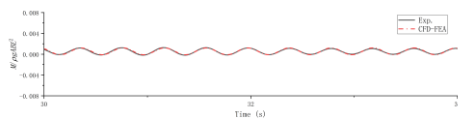
(a) WF (0.0-0.9Hz)



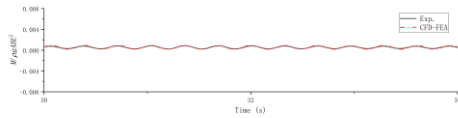
(b) 2nd order harmonics (0.9-1.5Hz)



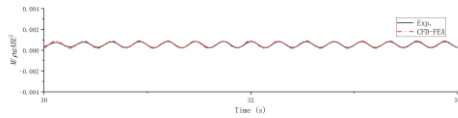
(c) 3rd order harmonics (1.5-2.1Hz)



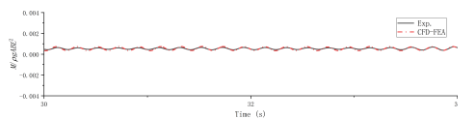
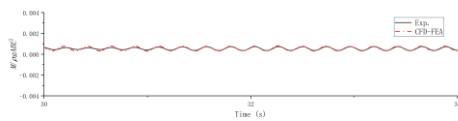
(d) 4th order harmonics (2.1-2.7Hz)



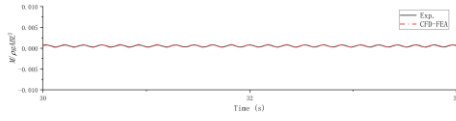
(e) 5th order harmonics (2.7-3.3Hz)



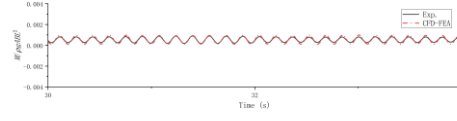
(f) 6th order harmonics (3.3-3.9Hz)



(g) 7th order harmonics (3.9-4.5Hz)

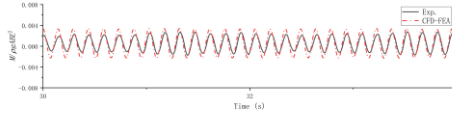


(h) 8th order harmonics (4.5-5.1Hz)



(i) 9th order harmonics (5.1-5.7Hz)

(j) 10th order harmonics (5.7-6.3Hz)



(i) 2-node HF (6.3-6.9Hz)

Figure 4-18 Higher-order components of VBM, test case VI ($\lambda/L=1.5$), Barge (B).

4.5 Conclusion

In this chapter, the higher-order springing phenomenon of a segmented Barge (B) ship model is investigated using experiment and an in-house two-way CFD-FEA coupled method. The coupled method is further developed from the previous work by Takami et al. [108], improving the effectiveness of the algorithm. Firstly, the motion responses are calculated and validated with the experiment. Then, the numerical hammering test is performed in calm water conditions with sudden excitation at the center of mass. The calculated wet natural frequency from the two-way coupled hammering test is compared with the experiment. Further, the springing phenomenon is investigated using experimental and numerical methods. The following could be concluded:

1. In this chapter, an in-house CFD-FEA coupled scheme developed further with a predictor-corrector is introduced. The results from the CFD-FEA coupled method agree well with the experimental results in terms of motion (heave, pitch) and flexible structure response (VBM at midship).

2. The current study showed the time step size in a two-way coupled simulation has a significant influence on the damping of the structural responses using the numerical wet hammering test.
3. The harmonic orders were calculated, confirming that the current model exhibits harmonics up to the 14th order – something not frequently reported in the literature.
4. The numerical time series of the individual harmonics have been dissected from the original signal and compared with experimental results, which further confirmed the effectiveness of the proposed method in terms of capturing springing.

Chapter 5

Slamming Induced Hydroelastic and Hydroelastoplastic Responses

5.1 Introduction

In this chapter, the hydroelasticity and hydroelastoplasticity of the wedge section model are investigated by the CFD-FEA coupled framework A. The CFD-FEA coupled framework A is validated for slamming-induced responses by published data within the elastic range. Then, to reproduce hydroelastoplastic responses, a full-scale wedge-type section is adopted. With impact velocities ranging from 9 m/s to 14 m/s, the slamming-induced responses within the elastoplastic range have been investigated.

The objectives of the present study could be summarized as follows:

1. The CFD-FEA coupled framework A is validated for slamming-induced responses by published data within the elastic range.
2. The CFD-FEA coupled framework A is able to predict hydroelastoplastic responses, which have the potential to replace the experiments and cross-verify the model for reduced-order models or data-driven models.

5.2 Validation

The validation of the CFD-FEA coupling framework A for slamming-induced hydroelastic and hydroelastoplastic responses is achieved by comparison with the strain obtained from a series of slamming experiments by Hosseinzadeh et al. [85,86,138]. A non-prismatic aluminum (alloy 5083-H111) wedge section was designed (see Figure 5-1), and the material properties are presented in Table 5-1.

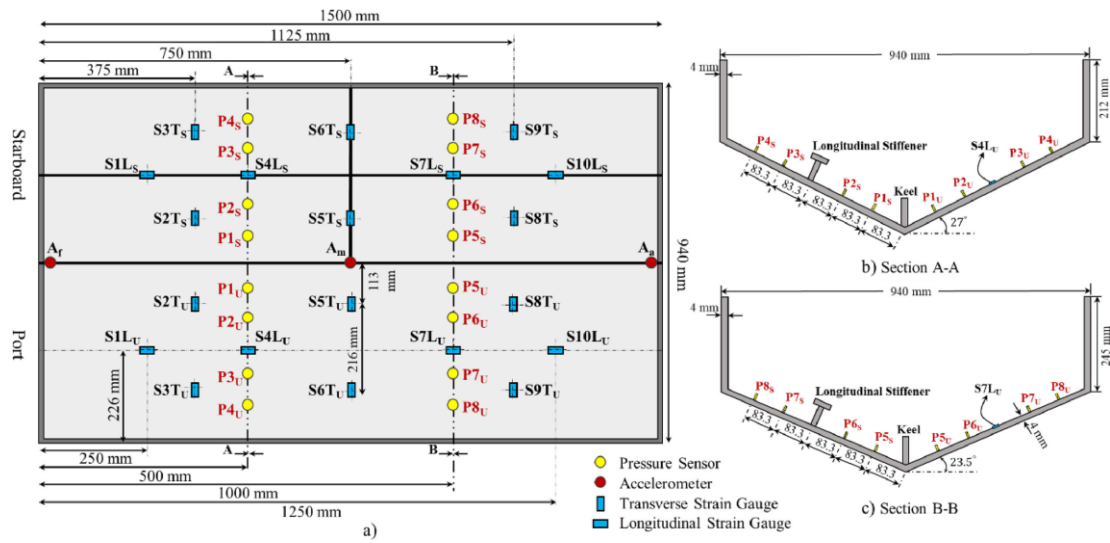


Figure 5-1 Wedge section, designed by Hosseinzadeh et al. [85,86,138].

Table 5-1 Material properties of the aluminum wedge section

Density	2700 kg/m ³
Young's Modulus	68 GPa
Poisson Ratio	0.33
Thickness	4 mm
Mass	55 kg

To reduce the computational cost, the numerical model has been simplified into 2D (the section with S2 and S3), and only the unstiffened side is considered elastic; otherwise, it is considered rigid. A prescribed motion, which comes from the experiment, is applied to the numerical model. The mesh size near the structure is set to 0.001m following Hosseinzadeh et al. [85,86,138] (see Figure 5-2). Time step size independence has been further checked, as it has been reported that the CFD-FEA coupling is sensitive to the time step size [124].

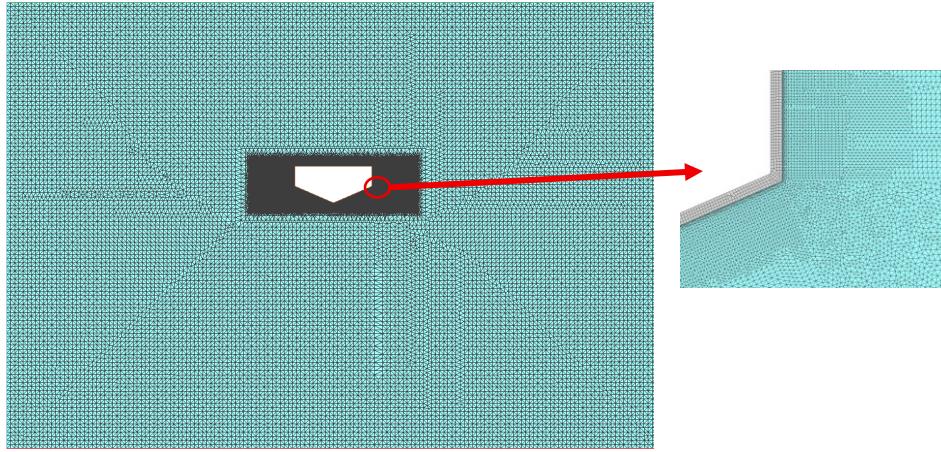


Figure 5-2 Mesh of the simulation domain, Wedge section.

Figure 5-3 shows the comparison of the strain time history between the experiment and different time step sizes at S2. Considering the 2D effect, the peak value predicted by the simulation is estimated to be higher than the experimental one; however, decreasing the time step size too much will lead to high computational cost. In this research, a time step size of 5×10^{-5} is adopted.

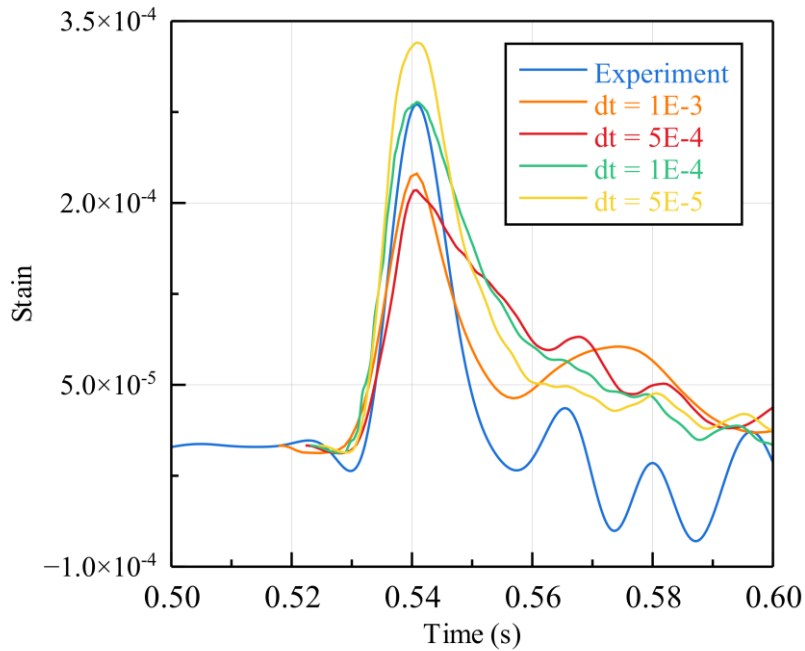


Figure 5-3 Comparison of strain time history, S2.

5.3 Target Model

Reproducing slamming-induced plastic responses at the model scale is typically challenging. Hence, a full-scale model was adopted. The target model was derived from the wedge (section B-B) shown in Figure 5-1, enlarged by approximately 40 times (see Figure 5-4). The material properties are summarized in Table 5-2.

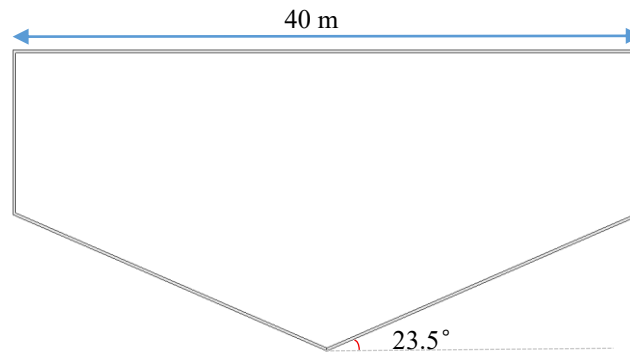


Figure 5-4 The subject structural model.

Table 5-2 Material properties of the full-scale wedge section

Density	7800 kg/m ³
Young's Modulus	210 GPa
Yield Stress	240 MPa
Poisson Ratio	0.30
Thickness	10 mm

Different from the model-scale one used in Section 5.2, only a small length (0.8m) of the panel was set to ideal elastoplastic (see Figure 5-5). The two ends of the elastoplastic panel were constrained in six degrees of freedom (DoF) locally. The mesh

size of both the FEA model and the CFD domain was set to 0.1m, and the time step size was set to 0.01s. Constant prescribed motions varying from 9m/s to 14m/s were applied to the model.

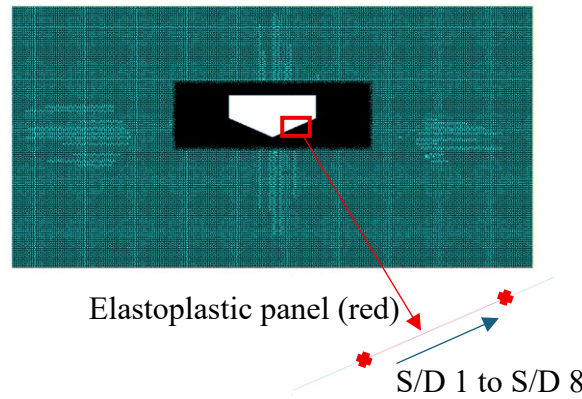
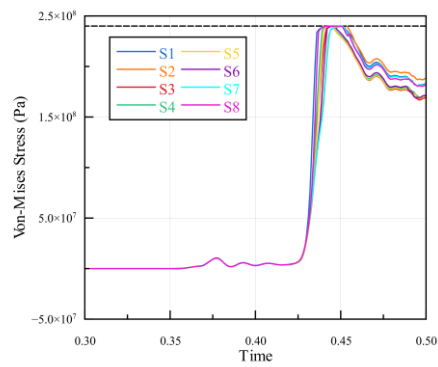


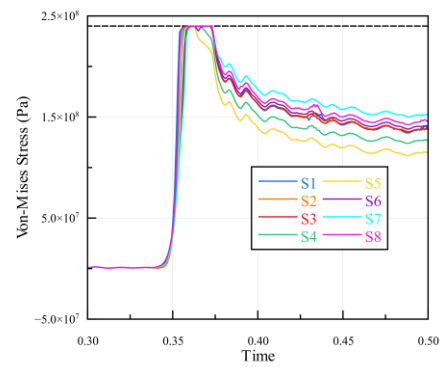
Figure 5-5 The full-scale model for CFD-FEA.

5.4 Results and Discussion

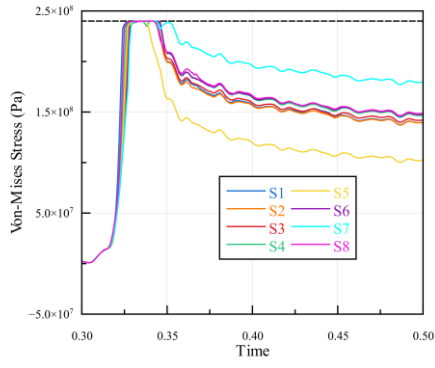
Figure 5-6 shows the Von-Mises stress at the outer integration point with different drop velocities. In each case, as soon as the Von Mises stress reaches the yielding stress (240 MPa), the structural behavior enters the plastic range.



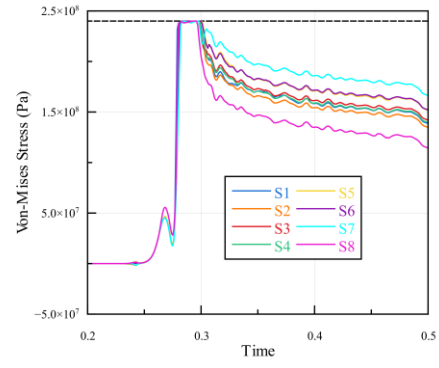
(a) $v = 9\text{m/s}$



(b) $v = 11\text{m/s}$



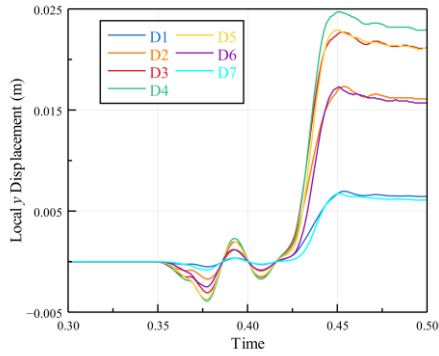
(c) $v = 12\text{m/s}$



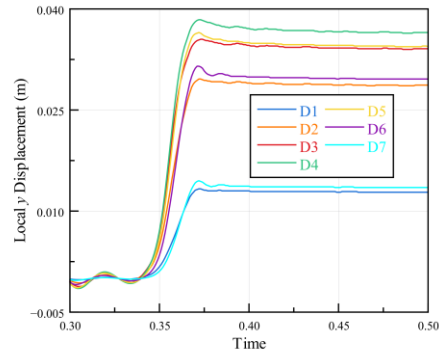
(d) $v = 14\text{m/s}$

Figure 5-6 Von-Mises stress at the outer integration point, with different drop velocities.

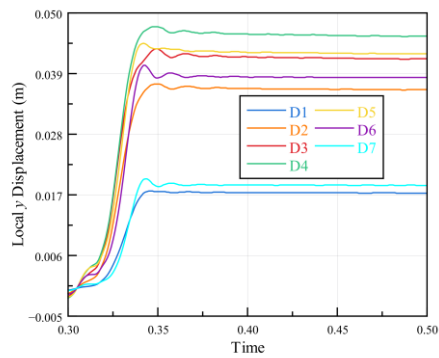
Figure 5-7 shows the local y displacement with different drop velocities. In each case, the local y displacement will maintain a constant value after some time, representing the final plastic deformation.



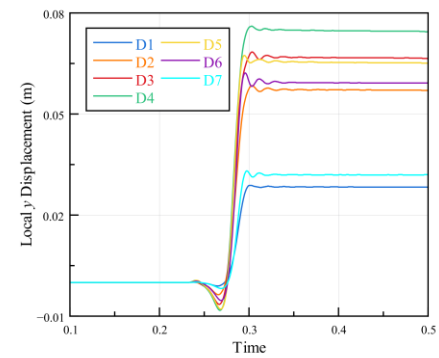
(a) $v = 9\text{m/s}$



(b) $v = 11\text{m/s}$



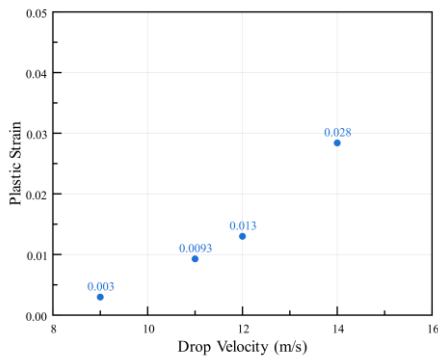
(c) $v = 12\text{m/s}$



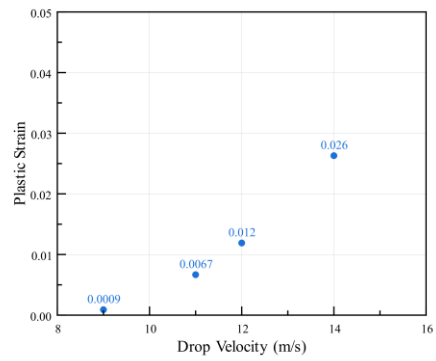
(b) $v = 14\text{m/s}$

Figure 5-7 Local y displacement, with different drop velocities.

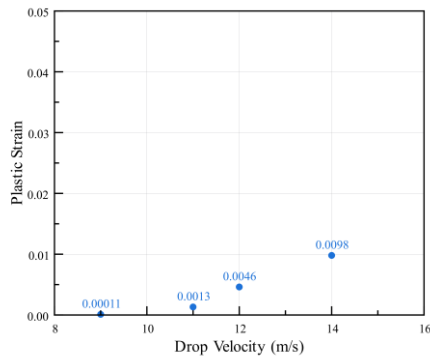
Figure 5-8 compares the plastic strain at different measuring points. It is not difficult to notice that with an increase in impact velocity, the plastic response exhibits a trend of increasing.



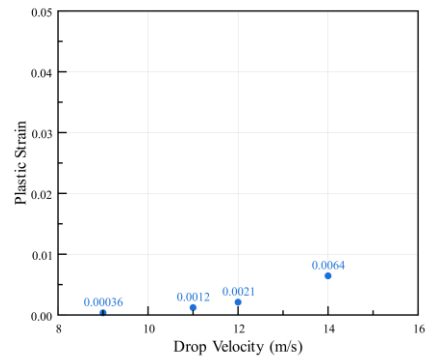
(a) S1



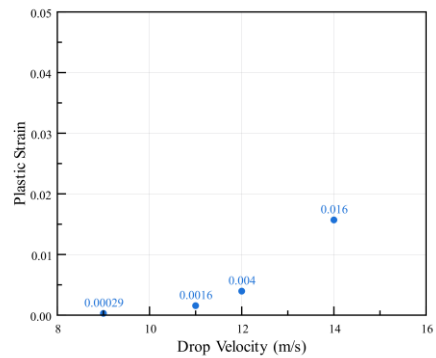
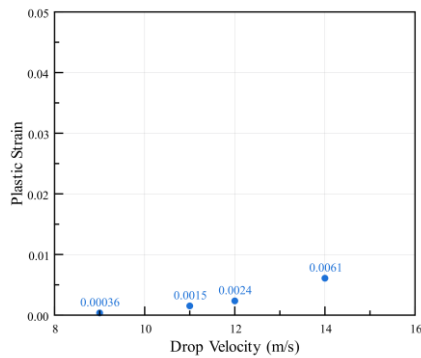
(b) S2



(c) S3



(d) S4



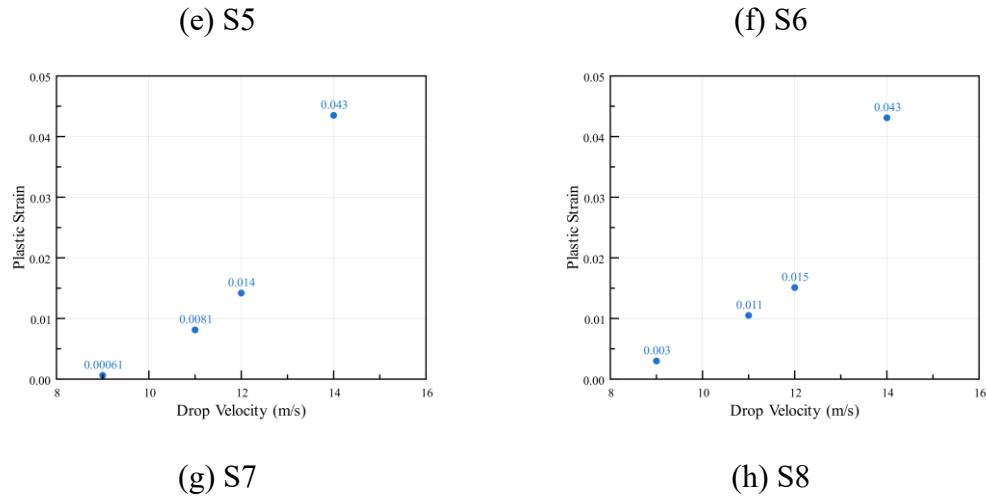


Figure 5-8 Comparison of plastic strain with different drop velocities.

5.5 Conclusion

This chapter investigates the slamming-induced hydroelastic/hydroelastoplastic behavior using a high-fidelity CFD-FEA coupled model. The results within the elastic range have been validated against experimental data. The analysis confirmed that the CFD-FEA coupled model is suitable for slamming problems and can serve as a basis for future research.

In this chapter, a high-fidelity CFD-FEA coupled model is adopted and validated for the slamming problems. This could be a useful tool for future cross-verification for a reduced-order model or a data-driven model.

Chapter 6

Nonlinear Hydroelastic Waves in Deep Water: Propagation and Profile Inversion

6.1 Introduction

This chapter investigates the hydroelastic interaction between nonlinear regular waves and a thin elastic plate floating on deep water. The two-way fluid–structure interaction (FSI) framework A is employed to reproduce nonlinear hydroelastic wave responses. To account for the high structural flexibility and to improve computational efficiency, a Laplace matrix scheme is newly incorporated into the framework. The numerical results are validated against experimental data and analytical solutions reported in earlier studies. The simulations reveal nonlinear wave profiles characterized by a steepened crest and a flattened trough, as well as their inverted counterparts, across different incident-wave frequency regimes. In the inverted-wave regime, a transition in wave profile is observed during propagation: the response shifts from an elevated form on the weatherside to a depressed form near the mid-plate region.

In this chapter, the nonlinear structural behavior of a thin elastic plate encountering deep-water regular waves is discussed. A CFD-FEA coupled model, which is designed to handle highly flexible FSI problems, has been adopted. As a high-fidelity numerical method, CFD-FEA coupling has the potential to become a robust model. Due to the nature of CFD-FEA coupling, it is efficient in capturing different nonlinearities without explicitly modeling. The adopted CFD-FEA coupled model is validated against experimental data and analytical solutions. Then, the hydroelastic responses of the flexible and thin elastic plate are further discussed. The primary objectives of this paper could be summarized as follows:

- To investigate the deep-water wave propagation on the flexible and thin elastic plate. The CFD-FEA coupled model could account for the nonlinear hydrodynamics directly, where the nonlinear characteristics like free-surface

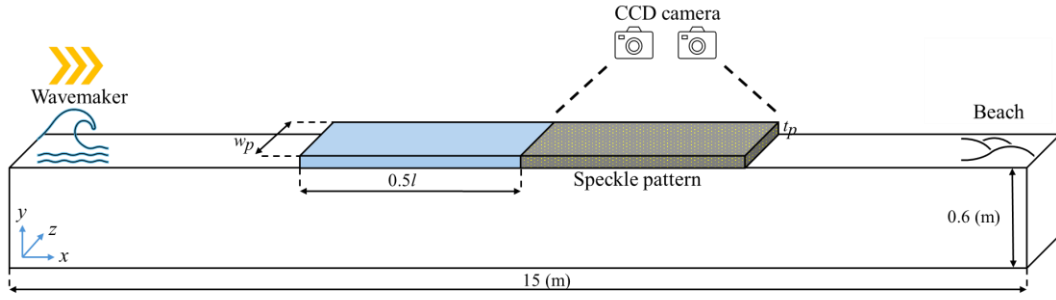
nonlinearity could be observed.

- To confirm the predictions by the analytical solution which has been derived by some of the present authors. In its derivation, it is assumed that the length of the structure is infinite. The effect of the structural end boundary on the nonlinear wave propagation may be clarified with the coupled CFD-FEA model.
- To validate the adopted CFD-FEA coupled model for the highly flexible FSI problems. This validation establishes the model as a reliable tool for future research, such as simulating interactions between waves (especially cnoidal waves) and long, elastic floating plates in shallow water [139].

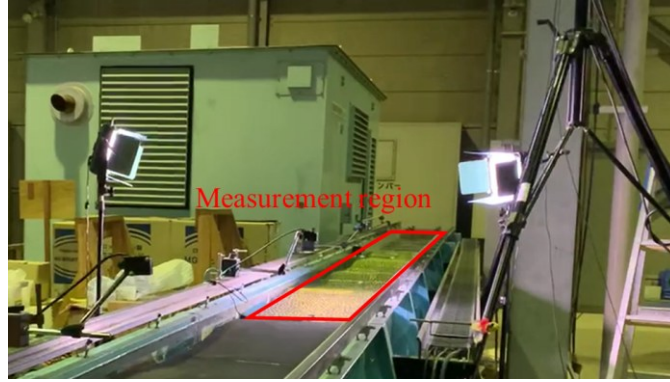
6.2 Tank Test

Experimental investigation of hydroelastic behavior of floating plate is one of the potential options to validate the coupled approaches. In this paper, the experimental data by Iijima et al. [140] is used for validation. The stereo Digital Image Correction (DIC) technique was used to capture the structural response of a thin, flexible floating plate under regular waves. This noncontact optical system allowed them to measure the time histories of deformation, including deflection and strain distributions.

In this section, a brief introduction to the experimental test is given. Further details can be found in Iijima et al. [140]. The experiment was conducted in a 15 m long, 0.5 m wide tank at the National Maritime Research Institute (NMRI), Mitaka, Tokyo, Japan. A constant water depth of 0.6 m was maintained for all tests. The hydroelastic behavior of VLFS could typically be simplified to an elastic plate floating on the free surface of the fluid. A plunger-type wavemaker installed at one end of the tank could generate regular waves with frequencies ranging from 0.62 to 1.18 Hz, while a wave-absorbing beach was installed at the other end for wave elimination.



(a) Set up of the tank test



(b) Snapshot of the tank test

Figure 6-1 Tank test of the thin elastic plate, conducted at NMRI.

Two sets of high-speed 12-megapixel charge-coupled device (CCD) cameras, integrated with a DIC system, measured the model's response by identifying its deformation and strain distribution. The thin elastic plate was made from C-4405, a lightweight, low-stiffness, closed-cell rubber sponge with a density of 190 kg/m^3 and Young's modulus of 2.0 MPa . The plate has a length (l) of 10 m , a width (w_p) of 0.48 m , and a thickness (t_p) of 0.02 m . The characteristic length $L^* = 2\sqrt{2}\pi^4 \sqrt{\frac{D^*}{\rho g}}$ (D^* represents the flexural rigidity of the plate or the ship beam following the discussion of Suzuki et al. [141], ρg is the specific weight of water) of the target model is 0.99 m , a value which is small relative to the overall model length. The bending stiffness EI of the target model is around 1.33 Nm for unit width. The plate model is sufficiently flexible, which allows wave propagation underneath the model.

6.3 Numerical Model

To solve a highly flexible FSI problem, it is always challenging for two-way CFD-FEA coupling. The primary concern associated with it is that the calculation may not converge; if it does, it may exhibit only linear convergence, which may be unacceptably slow [142]. The CFD-FEA coupled framework A based on commercial software LS-DYNA was adopted in this chapter. This framework proved effective and accurate in investigating the hydroelasticity of ship elastic response in Chapter 3. However, the current model exhibits high flexibility, which makes the problem unique from the previous study. Meanwhile, benefiting from the consideration of the interface Laplace matrix and stabilization matrix (see Section 2.3.4), this framework can handle the highly flexible FSI problem with high stability. Additionally, an internal wave generator based on momentum source term has been implemented into the numerical tank, which could avoid reflection and radiation waves in the fluid domain for long-duration simulations.

This section details the individual CFD and FEA models. Material nonlinearity has not been considered in this research. But critical nonlinearities such as large deflection and free-surface nonlinearity are considered in the numerical model.

6.3.1 FEA Model

Key parameters for the simulation are summarized in Table 6-1. The time integral constants (α , β , and γ) were selected based on the Bathe scheme [112]. This approach was chosen because while directly increasing β and γ can add numerical damping for stability, the Bathe scheme is known to provide better energy and momentum conservation. The time step size was set to the same as the CFD simulation, which is 0.001 s, to realize the so-called “strong two-way coupled simulation”. The displacement and energy relative convergence tolerance were specially set as 1.0×10^{-4} and 1.0×10^{-5} for the implicit simulation, respectively.

Table 6-1 Calculation setup of FEA, thin elastic plate

Solution	Dynamic implicit analysis based on FEM
Time integral constant β	0.25
Time integral constant γ	0.5
Time integral constant α	0.5
Mass proportional damping constant f_m	0.1
Stiffness proportional damping constant f_k	0.01
Time step size Δt	0.001 s

Table 6-2 Material properties of the FEA model, thin elastic plate

Density ρ_s	190 kg/m ³
Young's modulus E	2.3 MPa
Poisson's ratio	0.3

The FEA model is shown in Figure 6-2. The model was discretized by 2D shell-type element, and the mesh size was set as 0.002 m. The material properties of the FEA model are shown in Table 6-2. Plane strain assumptions were applied in the FEA simulation phase. The condition exists in an object that is constrained in the z -direction by rigid walls; the formulation only allows the resolution of strain in the xy plane (see Figure 6-3):

$$\varepsilon_{zz} = \gamma_{xz} = \gamma_{yz} = 0 \quad (6-1)$$

where ε_{zz} is the strain normal to the xy plane; γ_{xz} and γ_{yz} are the shear strain.

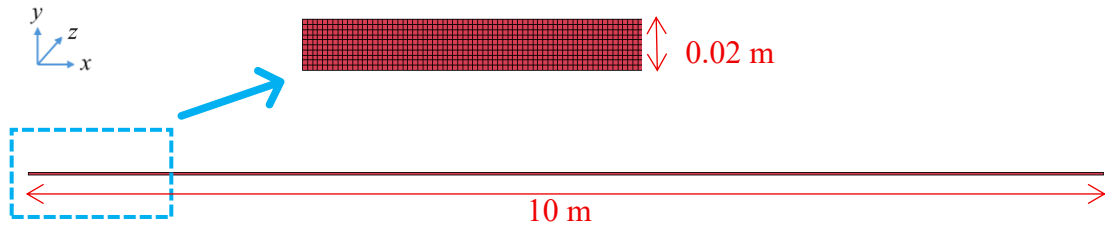


Figure 6-2 FEA model (thin elastic plate).

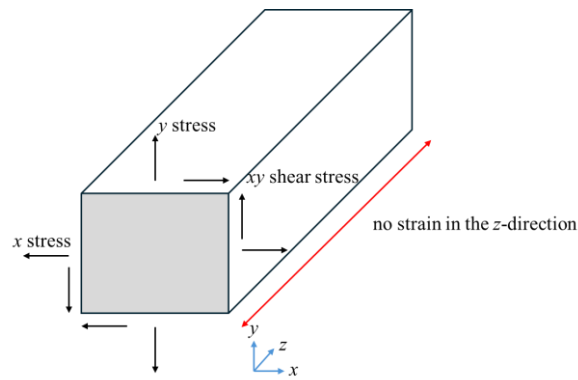


Figure 6-3 Plane strain analysis.

Rayleigh-type damping defined by Equation (2-18) was applied to the FE model. f_m and f_k have a relationship with the damping ratio ζ , which is defined by Equation (2-19).

A free vibration test was conducted in the experiment to obtain the material properties and estimate the damping constants. As illustrated in Figure 6-4, a free vibration test was conducted on a 1.0 m long, 0.02 m thick specimen by applying a load at midspan and then releasing with an initial downward offset. From the subsequent measurement, the vibration frequency was determined to be 13.7 rad/s, and a critical damping ratio was 0.077 rad/s [140].

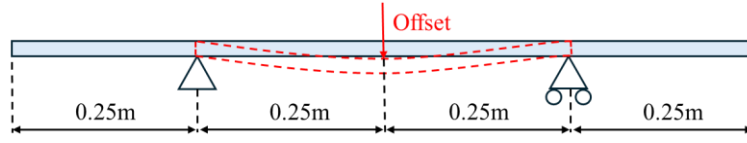


Figure 6-4 Setup for free vibration test.

Similarly, the free vibration test was conducted using an FEA simulation to tune the numerical settings. However, with two unknowns (f_m and f_k) and only one equation, equation (2-18) is underdetermined and admits no unique solution. In this chapter, the f_m and f_k are estimated by the following procedure:

1. The mass-proportional contribution is insignificant for the present system; thus, initial mass damping constant f_m^0 is set as a minimum value (≈ 0). Equation (2-19) then becomes:

$$\zeta \approx \frac{\omega f_k}{2} \quad (6-2)$$

and f_k could be calculated by equation (6-3) with the measured vibration frequency and critical damping ratio;

2. Apply f_m and f_k to the FEA model and perform the simulation;
3. Evaluate the numerical structural responses by comparing with the experimental ones. If good agreement is not reached, adjust the f_m from its previous value and then repeat step 2. Iterate until a satisfactory agreement is achieved.

f_m and f_k were finally estimated as 0.1 and 0.01 for the FEA simulation. Figure 6-5 shows good agreement between the simulated and experimental time histories for the midspan free vibration displacement. This result validates that the numerical model accurately captures the structural behavior of the physical specimen.

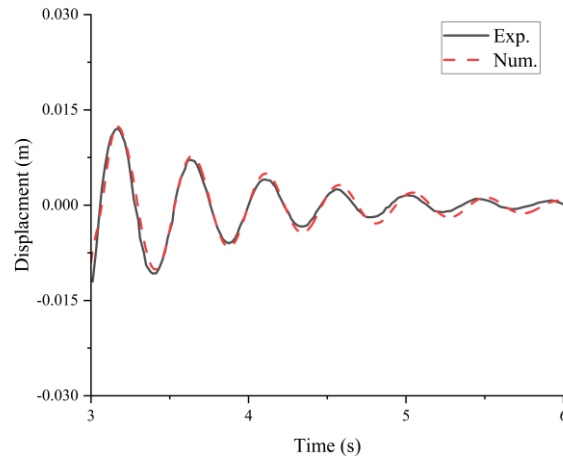


Figure 6-5 Time history of displacement at midspan, free vibration test.

6.3.2 CFD Model

The overall CFD domain is shown in Figure 6-6. The CFD calculation setup is summarized in Table 6-3.

Table 6-3 Calculation setup of CFD, thin elastic plate

Solution	Unsteady implicit analysis based on FEM
Physical model of fluid	Incompressible
Turbulence model	Variational multiscale approach (VMS)
Multiphase model	Level set
Time step size	0.001 s

For long-duration CFD simulations, wave generation via boundary conditions is limited by imperfect wave absorption: FSI-induced reflection and radiation waves are not fully absorbed at the open boundary and undergo secondary reflections, which then influence the solution. The author implemented an internal wave generator in the CFD solver for the 2D numerical tank, which could avoid reflection and radiation waves in

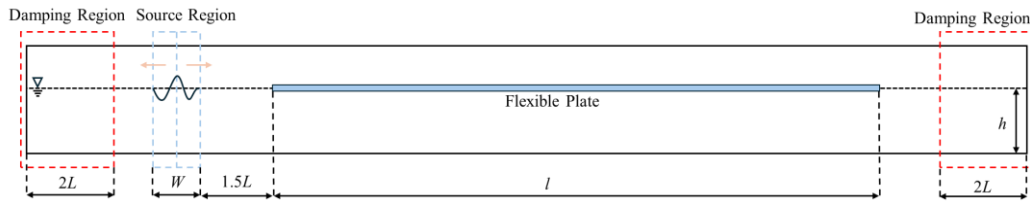
the fluid domain. The external force terms, also known as the momentum source S_{Mom} , were added to the momentum equation of the N-S equations. For monochromatic regular waves in a 2D domain, S_{Mom} could be defined as follows [143]:

$$S_{Mom} = -g(2\beta_0 x)e^{-\beta_0 x^2} \frac{D}{\omega} \sin(-\omega t) \quad (6-4)$$

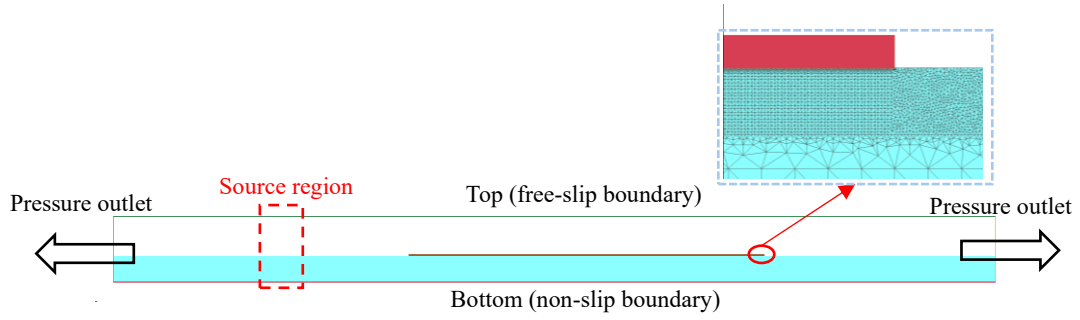
where,

$$D = \frac{2A(\omega^2 - \alpha_1 g k^4 h^3) \cos \theta}{\omega I_1 k (1 - \alpha_0 k^2 h^2)} \quad (6-5)$$

where angular frequency ω , wave number k , wave direction θ and wave amplitude A are the wave parameters; $I_1 = \sqrt{\frac{\pi}{\beta}} e^{\frac{k^2}{4\beta_0}}$ where $\beta_0 = \frac{80}{(\delta L)^2}$, $\delta = \frac{2W}{L}$ is a relevant parameter for the width of internal wave generation region, L is the wavelength, W is the width of the source region; $\alpha = \frac{z_\alpha}{h} (\frac{z_\alpha}{2h} + 1)$ where h is the depth of fluid and $z_\alpha = -0.53h$; $\alpha_1 = \alpha_0 + \frac{1}{3}$; x represents x coordinate. Similarly, the damping is achieved by adding a source term S_d to the momentum equations (see (3-1) to (3-3)).



(a) Illustration showing a flexible floating plate in a numerical water basin, with specifying the damping and wave source regions



(b) Mesh and boundary conditions.

Figure 6-6 Overview of the CFD domain, thin elastic plate.

In the CFD simulation setup, a triangular type of cell was used. To accurately capture the wave shape and FSI loads, the mesh at the free surface and around the flexible thin plate was locally refined (see Figure 6-6). A mesh resolution of at least 80 cells per wavelength and 20 cells per wave height is recommended by the ITTC [133]. However, in this chapter, balancing the mesh quality and simulation cost, a mesh size of $\lambda/5$ m (10 cells per wave height) was used at the free surface and around the flexible plate. The maximum mesh size in the fluid domain was $L/80$ m. The fluid domain was discretized into 2.1 million triangular cells. Figure 6-7 compares the free surface elevation for regular wave propagation between experimental measurement and numerical simulation. The simulation result is in good agreement with the experiment, with a slight mismatch in peak amplitude values.

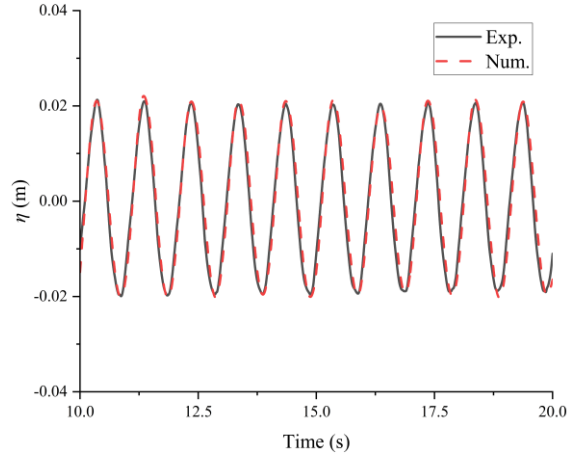


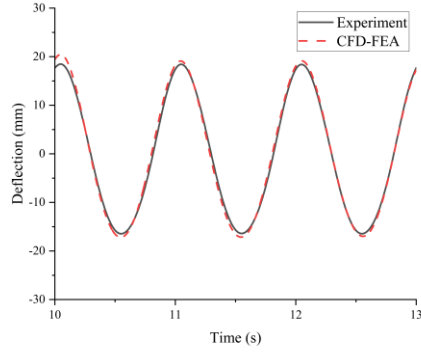
Figure 6-7 Free surface elevation for regular wave propagation, $A = 0.02$ m, $\omega = 6.28$ rad/s

6.4 Results and Discussion

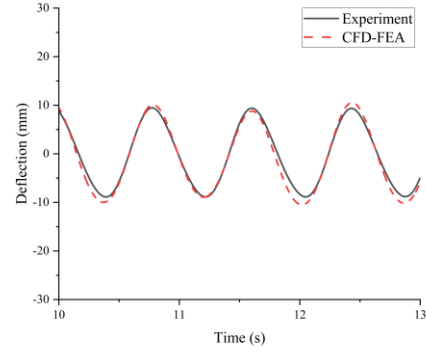
This section discusses the behavior of nonlinear deep-water waves interacting with a flexible plate. The coupled CFD–FEA framework is further validated by comparison with experimental results. Following Pambada et al. [100], behavioral transitions are expected near the instability frequency; therefore, two frequencies (6.28 rad/s and 7.60 rad/s), which are close to the instability frequency (7 rad/s), are selected for investigation.

6.4.1 Time History of Deflection and Strain

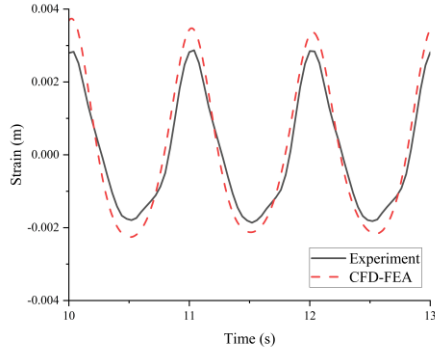
Figure 6-8 shows the time histories of deflection and strain, respectively, at the plate's centerline for incident wave frequencies of 6.28 rad/s and 7.60 rad/s. In general, the simulation and experimental results show good agreement. However, some discrepancies exist, particularly for the strain at 7.60 rad/s (see Figure 6-8 (b2)), which are attributed to higher-order components. This will be further discussed in section 6.4.3.



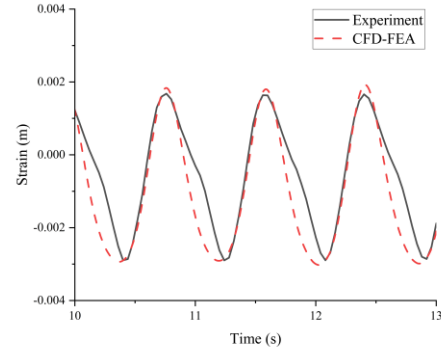
(a1)



(b1)



(a2)



(b2)

Figure 6-8 Time history of the deflection and strain at midspan: (a) $\omega_0 = 6.28$ rad/s;

(b) $\omega_0 = 7.60$ rad/s

A notable asymmetry is observed between the positive and negative peaks in the time histories of both experimental and numerical results (see Figure 6-8). For the deflection at 6.28 rad/s (see Figure 6-8 (a1)), the positive peak is larger in magnitude than the negative peak; however, this relationship inverts at 7.60 rad/s (see Figure 6-8 (b1)). This asymmetry is more pronounced in the strain results (see Figure 6-8 (a2) and (b2)). At the incident wave frequency of 6.28 rad/s, the positive strain peak is approximately 0.003 m, while the negative peak is approximately -0.002 m. Conversely, for the 7.60 rad/s case, this asymmetry is inverted, with the negative peak becoming larger in magnitude compared with the positive peak. The asymmetry is caused by the

superposition of the 1st and 2nd order components, which will be further discussed in section 6.4.2.

6.4.2 FFT Results of Deflection and Strain

To confirm the presence of higher-order harmonics, a Fast Fourier Transform (FFT) analysis was performed, with the results for deflection and strain at an incident wave frequency of 6.28 rad/s and presented in Figure 6-9. The frequency axis (the horizontal one) is normalized by the primary wave frequency ω_0 . In both the experimental and simulation results, a distinct second-order harmonic is clearly visible. As shown in Figure 6-9 (a), third-order harmonics are observable. However, the peak value of the simulation results is smaller than the experimental one. Similarly, in Figure 6-9 (b), higher-order harmonics up to the fifth order could be found in the strain response, and the peak value of the simulation results is smaller than the experimental one.

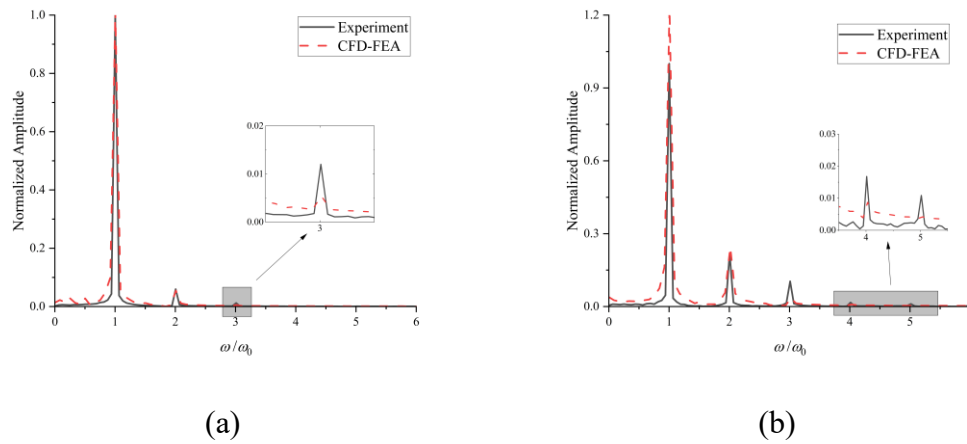


Figure 6-9 FFT results of deflection and strain at midspan, $\omega_0 = 6.28$ rad/s: (a) deflection; (b) strain

Similarly, FFT analysis was performed, with the results for deflection and strain at an incident wave frequency of 7.60 rad/s and presented in Figure 6-10. The simulation

underestimates the higher-order harmonic peaks of deflection and strain compared with the experimental ones. As discussed in section 3.1, balancing the mesh quality and simulation cost, a mesh size of $A/5$ m was used at the free surface and around the flexible plate, which is larger than the ITTC recommendation ($A/10$ m). This will decrease the accuracy of the CFD prediction. Meanwhile, an improperly chosen time step size can also lead to significant numerical damping from the CFD-FEA coupling process, which may result in numerical dissipation. A similar observation was also noticed in previous research by Xie et al. [144]. However, for CFD-FEA coupled simulation, a finer mesh size or smaller time step size may lead to unacceptable computational cost. With current settings, a single simulation will take 3-4 weeks using an AMD EPYC™ 7K62 server (96 cores, 3.3 GHz). Predicting higher-order terms accurately remains challenging.

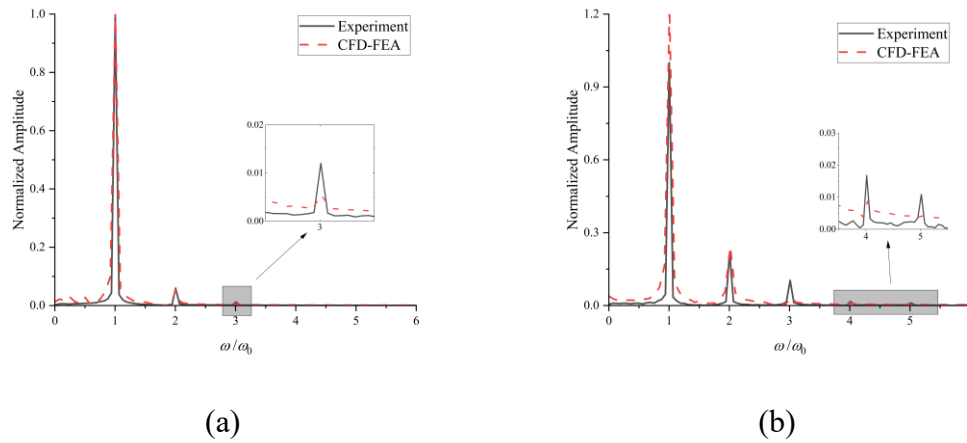
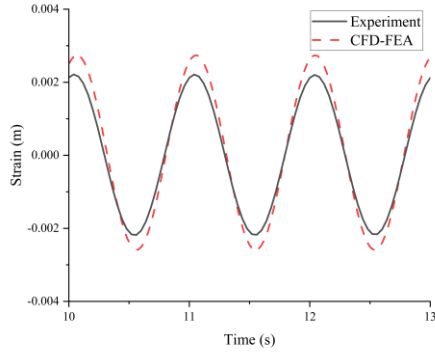


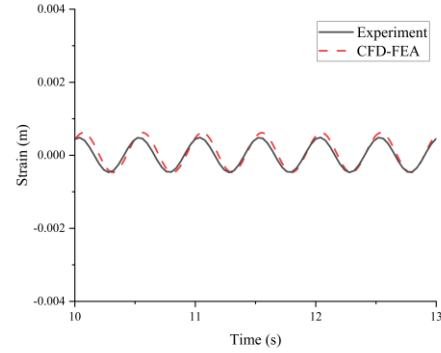
Figure 6-10 FFT results of deflection and strain at midspan, $\omega_0 = 7.60$ rad/s: (a) deflection; (b) strain

Figure 6-11 shows the comparison of the bandpass filter results of midspan strain time history between the experiment and simulation for the 6.28 rad/s incident wave. It

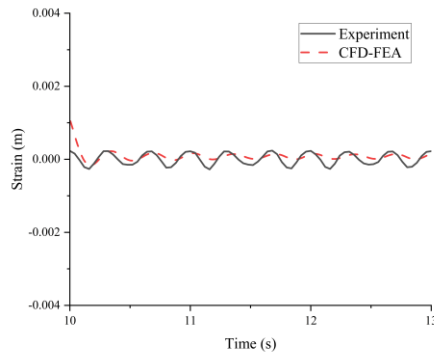
is noticeable that there is considerably good agreement between the experimental and simulation results at the 1st and 2nd order. The mismatch of the strain shown in Figure 6-8 (a2) and (b2) is caused by the components higher than 3rd order.



(a)



(b)



(c)

Figure 6-11 Higher order components of strain at midspan, $\omega_0 = 6.28$ rad/s: (a) 1st order component; (b) 2nd order component; (c) 3rd to 5th order component.

Similarly, by choosing an appropriate band for filtering, the 1st and 2nd order components of the midspan deflection and strain for the 6.28 rad/s incident wave are shown in Figure 6-12. At t_1 , the positive peak of the 1st order component meets the positive peak of the 2nd order component, which results in a large positive peak. Similarly, at t_2 , the negative peak of the 1st order component meets the positive peak

of the 2nd order component, which results in a smaller and flatter negative peak.

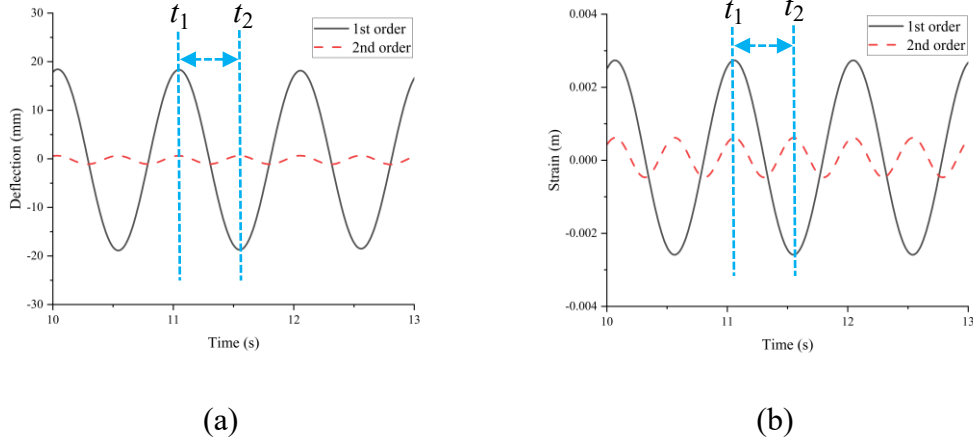


Figure 6-12 FFT filtering results of deflection and strain at midspan (CFD-FEA), $\omega_0 = 6.28 \text{ rad/s}$: (a) deflection; (b) strain

6.4.3 Wave Profiles

The results and discussion so far have revealed that the wave profile in the middle of the structure has a larger crest and flatter trough at the wave frequency 6.28 rad/s , which is common for a free-surface Stokes wave. However, it is the opposite for the wave frequency 7.60 rad/s : a smaller crest and a larger trough are observed as predicted by the analytical solutions. A significant question here is how the hydroelastic wave propagates at a frequency of 7.60 rad/s , starting from the edge boundary, to the middle part of the structure, and then to the leeward edge. It is natural to assume that the behavior at the edges of the plate follows that of free-surface or takes an elevated shape. However, during propagation, the wave profile flips to an inverted shape.

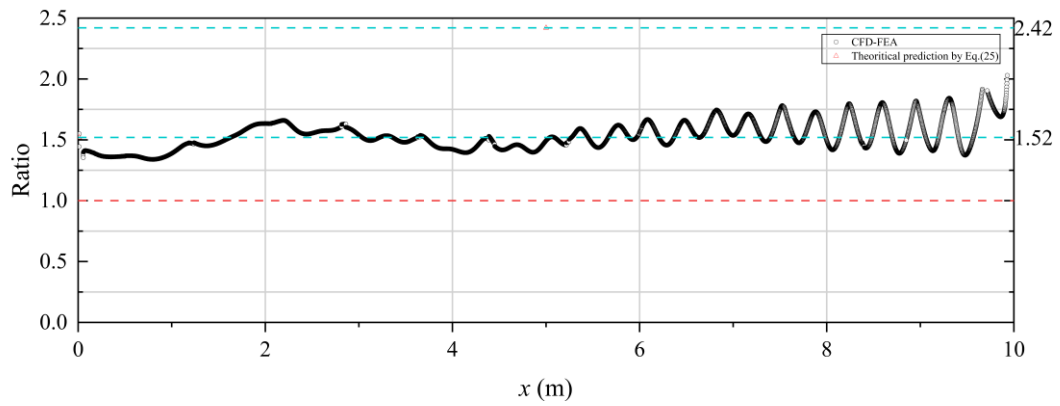


Figure 6-13 Distribution of the strain ratio between the positive and negative strain peaks, $\omega_0 = 6.28 \text{ rad/s}$

Figure 6-13 shows the distribution, along the plate, of the strain ratio R_S between the positive and negative peaks at the wave frequency of 6.28 rad/s . Several scattered points have been excluded close to the two ends of the plate, as both the positive and negative peaks are close to 0. Throughout the plate, the ratio remains greater than 1, which is characteristic of a free-surface Stokes wave. It should be noted that the ratio is not a constant value even in the middle of the plate since a standing wave is formed on the plate.

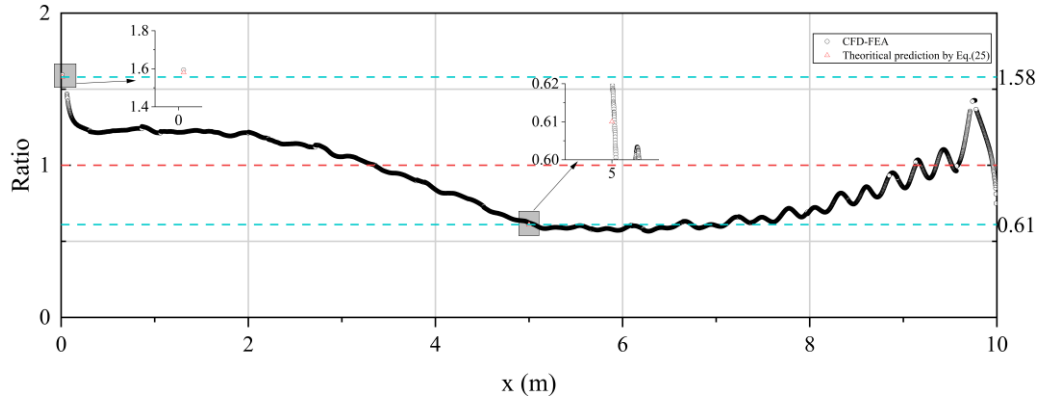


Figure 6-14 Distribution of the strain ratio between the positive and negative strain peaks, $\omega_0 = 7.60 \text{ rad/s}$

Figure 6-14 shows the distribution, along the plate, of the strain ratio R_S between the positive and negative peaks at the wave frequency of 7.6 rad/s. Similarly, several scattered points have been excluded close to the two ends of the plate. The ratio exceeds unity at both edges, while it remains nearly constant and below one in the central region. It then increases again toward the leeward edge, surpassing one once more. The wave profile becomes inverted in the middle part of the plate, and the transition from an elevated to an inverted profile occurs approximately 3 m from the weatherside edge. Given the first order wave component, given as $w_1 = a \cos(kx - \omega t)$ where a is the wave amplitude of the linear component, ω the incident wave circular frequency, k the wave number, the incident wave circular frequency and t the time, the analytical solutions predict the second order term w_2 for deep water as follows:

(For free-surface Stokes wave)

$$w_2 = \frac{1}{2} a^2 k \cos(2kx - 2\omega t) \quad (6-6)$$

where $\omega^2 = kg$ (g : the gravitational acceleration constant)

(For hydroelastic Stokes wave)

$$w_2 = \frac{1}{2} a^2 k \cos(2kx - 2\omega t) \quad (6-7)$$

where $\beta = 1/(2 - \frac{J_2}{J_1})$, $\omega^2 = kJ_1$, $J_1 = -\omega^2 \frac{m}{\rho} + EI \frac{k^4}{\rho} + g$, and $J_2 = -(2\omega)^2 \frac{m}{\rho} + EI \frac{(2k)^4}{\rho} + g$.

Here, m denotes the mass of plate per unit area. It is 10 kg/m² in the considered case. The free-surface formula can be regarded as a special case with $\beta = 1$. For the frequency regime below the instability frequency—where $J_2/J_1 = 2$ holds—the wave profile is elevated, exhibiting a larger crest and a flattened trough. In the higher-frequency regime where $J_2/J_1 > 2$, β becomes negative and the wave profile inverts, resulting in a flattened crest and a sharper, more pronounced trough.

The bending strain of the plate is proportional to the curvature, which is further given by the second derivative of the deflection. Then, the ratio between the positive and negative strains at the deck R_S may be evaluated as:

$$R_S \left(= \frac{\text{amplitude of positive strain}}{\text{amplitude of negative strain}} \right) = \frac{ak^2 + \frac{1}{2} a^2 k \beta (2k)^2}{ak^2 - \frac{1}{2} a^2 k \beta (2k)^2} = \frac{1 + 2a\beta k}{1 - 2a\beta k} \quad (6-8)$$

Table 6-4 summarizes the estimated results of the strain ratio R_S using Equation (6-8). The deflection amplitude is adopted from the coupled CFD-FEM analysis results. These results are also plotted in Figure 6-13 and Figure 6-14. They reveal that the theoretical predictions by the analytical solutions almost perfectly explain the tendency. The behavior at both edges of the plate is characterized by the behavior of a free-surface Stokes wave, while the behavior in the middle of the plate is characterized by the hydroelastic Stokes wave.

Table 6-4 Prediction of the strain ratio R_S . The deflection/wave amplitude is taken from the Coupled CFD-FEM analysis

Wave frequency	7.6 rad/s	6.28 rad/s
ω		

Type	Free surface	Hydroelastic	Free surface	Hydroelastic
Wave number k	5.87 m ⁻¹	5.28 m ⁻¹	4.03 m ⁻¹	3.93 m ⁻¹
Coef. β	1.0	-1.68	1.0	2.33
Amplitude a	0.019 m *	0.014 m	0.024 m *	0.022 m
Ratio R_s	1.58	0.61	1.52	2.42

* measured on the weatherside

6.5 Conclusions

This chapter investigated linear and nonlinear hydroelastic behavior of a thin elastic plate encountering deep-water waves using a high-fidelity CFD-FEA coupled model and validated against experimental data and analytical solutions that had been derived for nonlinear hydroelastic waves. The analysis confirmed that the CFD-FEA coupled model is suitable for highly flexible FSI problems and can serve as a basis for future research. The results revealed significant nonlinear components, including the second-order and third-order harmonics, in both deflection and strain, which result in asymmetrical and skewed response shapes. These nonlinear properties are found to agree well with the experimental data and predictions by analytical solutions. The key findings and conclusions of this study are as follows:

- In this chapter, a high-fidelity CFD-FEA coupled model is adopted and validated for the highly flexible FSI problems. This could be a useful tool for future cross-verification. Furthermore, the Laplace matrix at the FSI interface demonstrates good performance, enabling the stabilization of the coupled simulation despite a large added mass effect.
- The second and third-order harmonics were identified in the time histories of both deflection and strain, which resulted in the asymmetrical and skewed response shapes.
- In higher harmonics (4th and 5th), the simulation underpredicted the peak

amplitudes compared to the experiment. This discrepancy suggests that the numerical settings, particularly the CFD-FEA coupling scheme, require careful refinement to improve the accuracy, potentially by addressing sources of numerical damping.

At certain frequencies, the wave profile exhibits a flattened crest and sharper trough due to nonlinear effects. The degree of nonlinearity, quantified as the ratio of the positive to negative peak amplitudes, agrees well with the predictions of the analytical solutions. The transition from an elevated to an inverted profile occurs after several cycles of wave propagation.

Chapter 7

Conclusions and Future Work

This thesis aimed to develop and systematically validate high-fidelity, strongly coupled Computational Fluid Dynamics–Finite Element Analysis (CFD–FEA) frameworks for predicting the linear and nonlinear hydroelastic and hydroelastoplastic responses of marine structures. The research focused on enhancing the efficiency and robustness of fluid–structure coupling to address complex phenomena, including springing, slamming, and nonlinear hydroelastic wave propagation. Based on the investigations conducted in the preceding chapters, the main conclusions are summarized as follows:

- **Development and adoption of Enhanced Two-Way Coupled Frameworks**

Two distinct coupling frameworks were adopted to address different numerical challenges in fluid-structure interaction (FSI):

- **Framework A (Stabilizer):** A framework incorporating an interface Laplace matrix and a stabilization matrix was adopted. This approach effectively stabilized the coupling at the FSI interface, particularly when the added-mass effect was significant, without requiring modification to the solvers for predictors.
- **Framework B (Predictor):** To accelerate convergence, a rigid body motion (RBM)-deformation decomposed predictor was proposed. This method significantly reduced the number of sub-iterations required for convergence in partitioned coupling schemes, enhancing computational efficiency for problems where RBM contributes significantly to the total motion.
- **Hydroelastic Behavior of Segmented Ships**

Using Framework A, the hydroelastic responses of segmented Barge (A) and KVLCC2 models were investigated in regular waves. The numerical predictions for

motion Response Amplitude Operators (RAOs) showed good agreement with towing-tank experiments. The results confirmed that the ship structures responded predominantly in the 2-node vertical bending mode under the investigated conditions.

- **Higher-Order Springing Analysis**

The springing phenomenon was studied using the predictor-enhanced Framework B on a segmented Barge (B) model. The framework successfully reproduced the measured heave, pitch, and flexible vertical bending responses. Notably, the simulation resolved springing vibrations up to the 14th harmonic of the wave load, a phenomenon rarely documented in the literature. The study revealed that high-frequency components were largely driven by geometrical nonlinearity, providing new physical insights into the spectral characteristics of high-order springing.

- **Slamming-Induced Hydroelastic and Hydroelastoplastic Responses**

Framework A was applied to predict slamming loads on a wedge section. The framework was validated against experimental data within the elastic range. Furthermore, the study extended to full-scale models to investigate hydroelastoplastic responses. The results demonstrated that material plasticity significantly modifies the transient hydrodynamic and structural responses, highlighting the necessity of considering hydroelastoplastic effects during the design stage for extreme loading conditions.

- **Nonlinear Hydroelastic Waves in Deep Water**

The propagation of nonlinear waves along a highly flexible, thin elastic plate was investigated using Framework A with the Laplace matrix scheme. The simulations accurately reproduced key nonlinear wave profile features, including crest flattening and trough sharpening. Crucially, the model captured the profile inversion phenomenon, in which the wave profile shifts from an elevated to a depressed form during

propagation. The degree of nonlinearity, quantified by the ratio of positive to negative peak amplitudes, showed good agreement with analytical predictions.

While the proposed CFD-FEA frameworks have demonstrated high fidelity and robustness, several avenues remain for future research to further enhance their applicability and efficiency:

1. Investigation of Complex Sea Conditions

The current study primarily focused on regular waves with zero forward speed (except for general discussion). Future work should extend the application of the developed coupled frameworks to investigate hydroelastic responses under irregular waves, various forward speeds, and different heading angles to better represent realistic operational environments.

2. Shallow Water and Cnoidal Waves

The validation of the highly flexible FSI model establishes a reliable foundation for studying more complex wave-structure interactions. Future research could apply this model to interactions between long, elastic floating plates and cnoidal waves in shallow-water environments.

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