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**Experimental and Numerical Study of Steady and Unsteady
Multi-hull Systems Moving in Calm Water**

by

Dongchi Yu

A dissertation submitted in partial satisfaction of the

requirements for the degree of

Doctor of Philosophy

in

Engineering - Mechanical Engineering

in the

Graduate Division

of the

University of California, Berkeley

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Fall 2018

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Dongchi Yu

Abstract

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Professor Ronald W. Yeung, Chair

A multi-hull system can be a vessel made by multiple component hulls bound together by decks and superstructures, or several individual vessels traveling in the vicinity of each other with different speeds and courses. The former can be considered as a steady-state multi-hull system with the absence of relative motions among component hulls, while the latter is called an unsteady multi-hull system.

The wave resistance and wave pattern of each component hull in a steady-state multi-hull system, e.g. multi-hull vessels, are significantly affected by the interference of ships waves generated by the moving bodies. To study such steady-state multi-hull wave interference, towing-tank tests of asymmetric steady-state di-hull systems were performed, with the resistance, sinkage and trim of each component hull measured in the process. Then, the experimentally-validated Multi-hull Simple-source Panel Method (MSPM) is utilized for an extensive study of di-hull systems with a large number of hull configurations to extract a basic understanding of the wave-interference phenomenon. Significant wave-resistance reduction was shown for configurations that were evaluated experimentally, especially for the component hull traveling at the rear of the formations with high forward speeds. Furthermore, the wave patterns and wave resistances of di-hull systems were discovered to be considerably affected by the interference of waves with the strongest free-wave spectral density of the component hull as a mono-hull in isolation, and the beneficial hull configurations with reduced wave resistance can be qualitatively determined with a knowledge of mono-hull free-wave spectrum and simple wave-angle alignments. Moreover, for di-hull systems with small spacing between component hulls, the near-field interaction, which is normally missing in the far-field analyses, was found to have non-negligible effects over the wave resistances and wave patterns of the individual component hulls as well as the di-hull systems they constitute.

For the unsteady modeling of multi-hull systems, a group of independent vessels traveling in the vicinity of each other during overtaking maneuvers with relative motions among component hulls was studied. The MSPM computation introduced in the steady-state study was generalized to simulate the moderate-speed overtaking maneuvers, and the results were vali-

dated by experimental measurements of towing-tank model tests. The influence of overtaking speed and the speed difference between vessels on the hydrodynamic loads were studied. It was found that the deformed free surface caused by the blockage of hull bodies, wave-making of the vessels, and the interference of unsteady wave patterns considerably affect the hydrodynamic interaction forces and moments between the vessels. In addition, the slower ship being overtaken generally experiences larger loads with stronger variations than the passing ship. The loads on both hulls become more similar to those of a steady-state di-hull system when the speed difference between vessels is small.

Lastly, with the abundant computational results provided by the GPU-accelerated MSPM computation developed in this work, a Deep Neural Network (DNN) was trained and tested to efficiently evaluate the calm-water transportation efficiencies of a large number of different trimaran configurations. In this manner, the global understanding of the influences of different design parameters, such as the sizes and shapes of the component hulls, hull spacing, and forward speed can be established, and the optimal configuration with the best transportation efficiency can be quickly determined at the early-design stage. It is discovered that with sufficient computational results, the DNN can be trained to accurately evaluate a large quantity of trimaran configurations of various design parameters with very little computing cost.

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Chapter 1

Introduction

Multi-hull systems refer to vessels consisting of multiple component hulls [2], or individual vessels traveling in the vicinity of each other in a fleet or during close-quarter maneuvering. In the present study, the former is termed as the steady-state multi-hull system with the absence of relative motions among component hulls, while the latter is called the unsteady multi-hull system since the time-dependent interactions exist among component hulls. The knowledge of hydrodynamic interactions between component hulls in multi-hull systems is critical to the design and optimization of multi-hull vessels, and the operational safety of ships during close-quarter maneuvering in a fleet or congested waterways. Therefore, in the present study, the hydrodynamic interactions of component hulls in a steady or unsteady multi-hull system are investigated using experimental, computational and statistical approaches.

In this chapter, the general background and motivations of the present study on multi-hull systems are presented, and a brief literature review on the previous scientific and engineering studies on the hydrodynamic interactions among hulls in a multi-hull system is provided.

1.1 Steady-state multi-hull systems: multi-hull vessels

Catamarans and trimarans are typical examples for multi-hull vessels, which are ships made by two and three component hulls, respectively. Compared with the conventional mono-hull ships, multi-hull vessels generally has the advantages of low resistance, large deck area, high stability and futuristic appearance [3]. Therefore, multi-hull vessels are becoming popular choices for high-performance maritime transportation, operation and recreation, and frequently serve as high-speed ferries [4], next-generation naval vessels [5] and luxurious yachts [6].

Reducing time and fuel consumption while delivering larger payloads has always been the major objective of ship designers [7, 8]. Therefore, among all welcoming features of multi-hull vessels, the low calm-water wave resistance for high speed cruising has in particular attracted considerable attention from hydrodynamic researchers and ship designers. Most noticeably, in addition to the reduced wave resistance brought by the utilization of slender component hulls, properly-selected configurations of hull spacing and forward speed can bring about significant wave-resistance reduction through interference of ship waves. Therefore, the study of the physics behind multi-hull wave interference and wave-resistance reduction is critical for the design and optimization of multi-hull vessels.

Much of the early endeavor in this regard has been focusing on the analytical thin-ship computation [9] and linear wave-pattern superpositions [10, 11, 12]. Hull thickness is ig-

nored in this type of computation with Havelock sources [13, 14] only distributed along the centerplane area of each component hull. Furthermore, most of the thin-ship computations concentrate on the far-field free-wave that only the single integral of Havelock source is evaluated [11, 15], while the influence of the double integral, which represents the near-field non-free ship waves [16] is ignored. Furthermore, the steady wave patterns of the component hulls when traveling as isolated mono-hulls are directly superposed to compute multi-hull wave interference; therefore, the blockage effect created by the nearby hull body is missing [3]. Consequently, the thin-ship computation is generally capable of capturing the wave resistances of multi-hull systems made by slender component hulls with large hull spacings; however, the absence of near-field interaction renders the prediction less satisfactory for realistic hull forms and multi-hull systems with closely spaced component hulls. Other numerical methods such as Reynold-averaged Navier-Stokes (RANS) simulation [17] and potential-flow panel method [18] are also utilized for studying calm-water wave resistances of catamarans. However, because of the high computational costs, it is difficult to apply the popular viscous RANS simulation to cover a large number of configurations in an optimization problem. Meanwhile, the three-dimensional potential-flow panel method appears to be a promising compromise between efficiency and accuracy; the two component hulls are solved fully coupled with the aforementioned near-field interaction and the thickness of component hulls taken into consideration. Still, little work has been done to utilize potential-flow panel method to cover di-hull systems with sufficient variations of hull spacing and forward speeds to support a comprehensive study of di-hull wave interference.

Experimental investigations of the multi-hull systems were mainly towing-tank tests of catamarans [19, 20, 21, 22]. In these experimental studies, catamarans with different lateral hull spacings and forward speeds were evaluated, and the wave-resistance components were extracted by the flat-plate assumption [23] from the measured total resistances. Furthermore, the wave patterns of multi-hull systems were also studied extensively [24, 25]. Moreover, it is worth mentioning that the classic wave-pattern analysis for deriving mono-hull wave resistance based on wave-cut data can be further utilized in the study of di-hull systems. In this regard, the wave amplitude function which represents the wave-making feature of a given hull is firstly deduced from experimental wave-cut data [26] and subsequently invoked by the analytical modeling to compute the multi-hull wave resistance [27]. Although such theory-experiment hybridization approach still suffers from the absence of near-field interactions, the uncertainty brought by thin-ship approximation can be avoided, and the wave resistance of multi-hull vessels with any hull configurations can be quickly determined by the mono-hull towing tank tests of each component hull. Still, experimental exploration of the asymmetric di-hull systems with non-zero longitudinal hull spacing is uncommon, and the theoretically predicted dramatic wave-resistance reduction of staggered di-hull systems [12] was rarely validated experimentally. In addition, in the relatively few existing works on the asymmetric di-hull systems [12, 11], effect of wave interference on the wave resistance, sinkage and trim of each individual component hull was not addressed. Again, similar to its RANS counterparts, the costs of towing tank tests make it unrealistic for extensive study of multi-hull systems with various combinations of hull spacing and forward speeds, and it is hard to evaluate the

influence of each variable from the limited amount of experimental data.

Thus, in the present study, the steady wave interference of di-hull systems in calm water is investigated both experimentally and computationally. Asymmetric di-hull systems of selected hull configurations were evaluated through the towing-tank tests, and the influence of wave interference on the wave resistance, sinkage and trim of each component hull was examined. Then, the experimentally validated potential-flow panel method computation is utilized for an extensive study of di-hull systems with a large number of hull configurations to facilitate a better understanding of the di-hull wave interference. Moreover, the relation between the free-wave spectra [28] of the component hulls in isolation and the optimal wave-cancellation and wave-resistance reduction of the constituted di-hull systems are examined.

1.2 Unsteady multi-hull systems: ship-to-ship interaction in close-quarter maneuvering

As opposed to the steady-state multi-hull systems introduced in the previous section, an unsteady multi-hull system is formed when relative motions exist among component hulls. Typical examples include vessels operating in congested waterways such as harbor area, channels and in the vicinity of other vessels [29]. In particular, the study on ship-to-ship interactions in close quarters is essential for the safety of maritime transportations, as the collision accidents among ships usually lead to catastrophic environmental and financial consequences [30]. Furthermore, the advent of unmanned trans-ocean fleets [31] and swarm-intelligent surface robotics [32] call for investigations on the interactions between vessels traveling at moderate to high speeds in open waters, which is of practical interests for the operational safety and control strategies of the surface vessels involved.

The investigations of ship-to-ship interactions originate from the studies of the hydrodynamic loads on vessels moving in the proximity of fixed obstacles such as banks of canals [33, 34]. Then, the overtaking (ships travel in parallel courses and same direction with one ship going pass another in the process) and encountering (ships travel in parallel courses and opposite directions) maneuvers of vessels have been widely investigated by hydrodynamic researchers [35, 36, 37, 38]. Most of these studies focus on low-speed maneuvers of large vessels in restricted shallow waters, where the free-surface disturbance generated by the moving vessels is negligible. Therefore, most of the potential-flow approaches apply the classic double-body flow modeling [39, 33, 40] in which the free surface is treated as a rigid lid without vertical deformation. In the mean time, the viscous-fluid RANS simulation is also widely utilized to study the interactions between vessels in overtaking and encountering maneuvers [41]. In addition, comprehensive experimental investigations on the ship wave patterns [42, 43], ship-to-ship [29] and ship-bank interactions [44] were also conducted to study the wave field generated by a passing hull and the low-speed proximity maneuvers of vessels in practical engineering scenarios.

However, the unsteady interactions between vessels moving at moderate to high speeds,

carry unique features which are seldom investigated. First of all, as the wave-making of vessels is becoming much more significant at higher speeds, the interaction between vessels could be profoundly influenced by uneven free-surface elevations around hulls, and the associated pressure imbalance. Secondly, compared with the steady-state multi-hull systems, the unsteady flow field created by the speed differences between vessels could also have non-negligible influences on their hydrodynamic loads. Lastly, although the influence of wave interference on the wave resistances of steady-state multi-hull systems are widely studied, similar effects in an unsteady di-hull system with relative motions between vessels is rarely, if at all, investigated. With the considerable disturbances on the free surface due to the moving vessels, the assumption of rigid free surface in inviscid double-body flow computation is no longer justifiable. Besides, similar to the study on steady-state multi-hull systems, the high-fidelity RANS simulations and the experimental approaches are often too costly to cover sufficient amount of configurations to establish global knowledge of ship-to-ship interactions.

Therefore, in the present study, the ship-to-ship interactions of selected hull configurations are studied experimentally by towing-tank model tests. In the mean time, the potential-flow panel method introduced in the steady-state study is also modified to simulate the unsteady hydrodynamic interactions between two vessels in the overtaking maneuvers, where free-surface effect are taken into consideration. Then, the the influences of varying speed differences between hulls in the overtaking maneuvers are studied, and the free-surface effect in moderate-speed overtaking maneuvers is further examined.

1.3 Data-assisted multi-hull vessel design optimization

Optimization of multi-hull systems can be rather time-consuming because of the large number of possible configurations to be evaluated. It is usually difficult to cover sufficient amount of multi-hull configurations by direct computational and experimental approaches as mentioned above. In this regard, a promising solution may be offered by today’s advancing machine-learning technology. With sufficient amount of data provided by the direct approaches, an advanced learning algorithm such as the Deep Neural Network (DNN) [45] can be trained to accurately approximate the relation between design features of a multi-hull system and its target hydrodynamic performances. Then, with the help of modern advanced machine-learning programming framework [46], a large quantity of multi-hull configurations can be quickly and accurately evaluated with the inferencing capability of the learning algorithm, and the optimal configuration with the best target performance can be determined without repeatedly invoking the expensive direct hydrodynamic analysis.

It should be pointed out that similar data analytic methods have been widely used in the design and optimization of marine vehicles. For example, based on hydrodynamic analysis and engineering practice, researchers and engineers have come up with effective approaches for deducing the hydrodynamic performances of marine vehicles from those of the similar

designs with regression analysis [47, 48, 49]. Meanwhile, simple machine-learning algorithm such as the support vector machine (SVM) and clustering algorithm have been utilized to facilitate investigations in the area of naval architecture and ocean engineering [50, 51]. However, because of the insufficient computing power in the earlier days and the scarce of data on existing vessels, many of the early data-assisted design and optimization studies suffer from the lack of training data, oversimplified regression modeling, and over-conservative modifications, which results in unsatisfactory reliability and generality for large adjustments to design parameters.

Therefore, to validate the effectiveness and feasibility of data-assisted approaches in the design of multi-hull vessels, the study on deep-learning-based evaluation and optimization of trimaran configurations is carried out. Adequate training instances on trimaran calm-water transporting efficiency are firstly produced by the efficient potential-flow computation introduced in the previous section. Then, a DNN is trained with the computational results to establish the relations between the major design parameters (hull geometry, spacing and forward speed) and the calm-water merit coefficient of the trimaran. Eventually, large quantities of trimaran configurations can be evaluated simultaneously by the fast forward-propagation on the trained DNN, and the optimal design with the best calm-water transporting efficiency within a predefined forward speed range can be quickly identified.

1.4 Outline of the present study

The next chapter will first expound the mathematical formulation of steady and unsteady multi-hull systems. Then, an efficient potential-flow panel method computation, called the Multi-hull Simple-source Panel Method (MSPM) for studying the steady-state and time-dependent simulations of different multi-hull systems is presented in Chapter 3. Later, the hardware implementations and data sampling and processing methods for the towing-tank tests on the measurement of calm-water wave resistances, mono-hull wave-cut measurement, and moderate-speed overtaking experiments are documented in Chapter 4. The results and analysis of the steady and unsteady multi-hull systems are presented in Chapters 5 and 6, respectively. In Chapter 7, the DNN-based evaluation and optimization of trimaran calm-water efficiency is demonstrated. Lastly, the conclusions of the present study as well as some possible directions for future research are given in Chapter 8.

Chapter 2

Mathematical Formulation of Di-hull Interaction Problems in Calm Water

In this chapter, the mathematical formulation of steady and unsteady multi-hull systems in calm water is presented. A steady-state di-hull system is formed when both component hulls travel with the same constant forward speed with a constant formation. On the other hand, an unsteady di-hull system arises when two component hulls travel with different speeds. In the meantime, since the studies on di-hull systems and those of the conventional mono-hull vessels share many similarities, the mathematical modeling of mono-hull wave-making problem will also be addressed briefly.

2.1 Major assumptions and simplifications of the problem

In the present study, two hulls are restricted to move with constant forward speeds on calm-water surface in parallel and straight courses. The fluid domain is unbounded with infinite water depth. Previous studies [11, 10] and engineering practices show that ships traveling at moderate to high speeds generate significant trailing waves, and the interference of wave patterns can result in considerable change of the wave resistance of multi-hull vessels. Meanwhile, because of the comparatively large spacing between component hulls, physical observations show that the local influences of boundary layer and breaking waves may not be sufficiently large to noticeably affect the other hull. Therefore, the inviscid wave interactions between hulls is pursued in depth in the current research, while the influences from fluid viscosity and wave-breaking are considered secondary and beyond the scope of the study. Furthermore, in the present study each of the component hull is free to have sinkage and trim, while other modes of motions are restricted.

2.2 Coordinate systems, hull geometry and the mathematical formulations

Consider two hulls denoted as hull 1 and hull 2 moving in parallel at constant speeds U_1 and U_2 in an inviscid fluid domain of infinite depth. In addition to the earth-fixed coordinate system $Oxyz$, two moving coordinate systems $O_jx_jy_jz_j$ ($j = 1, 2$) are utilized and fixed on the corresponding component hulls, as shown in Fig. 2.1. The wetted surface of each hull is

denoted as S_{H_1} and S_{H_2} , and the calm water surface is denoted as S_F . Each of the moving coordinate system travels with the same speed as the corresponding hull in the positive x_j direction, and the origin is placed at the intersection of mid-ship section, centerplane and calm-water surface. The longitudinal distance between component hulls, stagger st is measured from the midship section of hull 1 to that of hull 2, while the lateral distance between hull, separation sp is taken as the absolute distance between the centerplanes of two hulls,

$$\begin{aligned} st &= x_2 - x_1, \\ sp &= |y_2 - y_1|. \end{aligned}$$

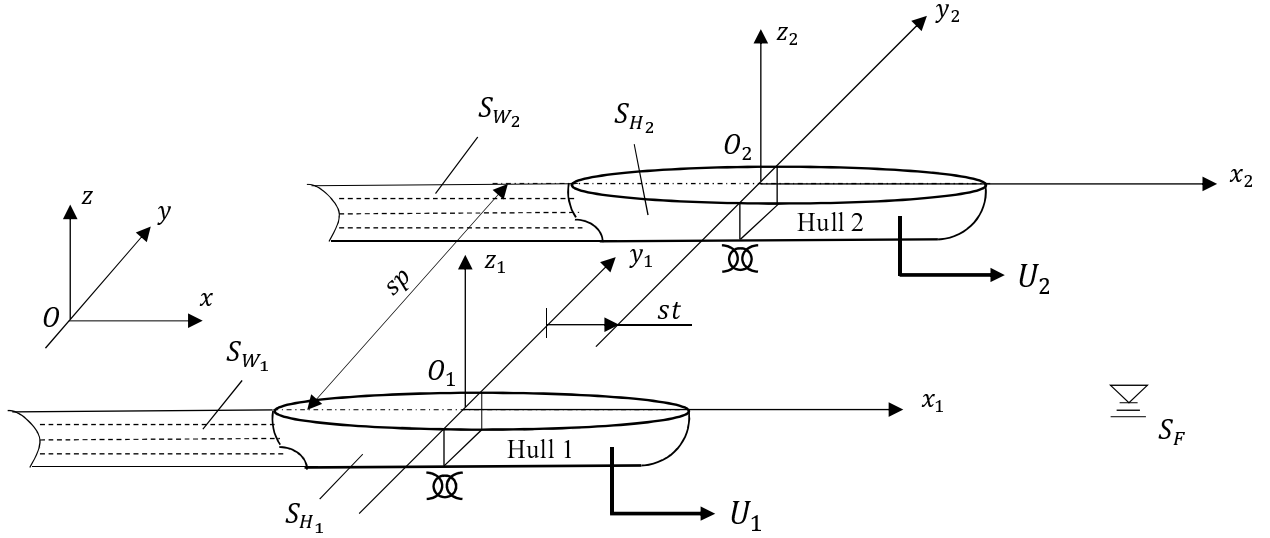


Figure 2.1: Moving coordinate systems of the present study.

Potential-flow theory is applied that the water is assumed inviscid, incompressible and the flow irrotational. Then the velocity field at time t observed in earth-fixed system can be described by the velocity potential ϕ governed by the Laplace equation,

$$\nabla^2 \phi(\mathbf{x}, t) = 0, \quad j = 1, 2 \quad (2.1)$$

where $\mathbf{x} = (x, y, z)$ denotes a point in the fluid domain.

Water particles should never penetrate the hull surfaces, therefore the kinematic boundary conditions to be satisfied on the hull surfaces S_{H_1} and S_{H_2} are given by

$$\phi_n|_{S_{H_j}} = U_j(n_{x,j}), \quad j = 1, 2 \quad (2.2)$$

where $\mathbf{n}_j = (n_{x,j}, n_{y,j}, n_{z,j})$ is unit interior normal to the hull surface S_{H_j} . At very large water depth the disturbances caused by the moving hulls decays to, therefore, the deep water condition to be satisfied takes shape as

$$\nabla\phi \rightarrow 0. \quad (2.3)$$

Further, the upstream radiation condition states that the disturbance of the fluid must vanish sufficiently rapidly far ahead of the ship, which is given by

$$\sqrt{r}|\nabla\phi| \rightarrow 0, \quad x_j \rightarrow \infty, \quad j = 1, 2. \quad (2.4)$$

where $r = \sqrt{x^2 + y^2}$. To implement the upstream radiation condition numerically, the values of ϕ and the free-surface elevation are set to zero at the upstream boundary of the computational domain.

Meanwhile, it is noted that water particles on the free surface should remain on the free surface, while the gauge pressure on the free surface is taken as a constant value in excess of atmospheric pressure, which is taken as zero everywhere. Consequently, the kinematic and dynamic free-surface boundary conditions are written as

$$\begin{aligned} \zeta_t(x, y, t) &= \phi_z(x, y, \zeta, t) - \phi_x \zeta_x - \phi_y \zeta_y, \\ \phi_t(x, y, \zeta, t) &= -g\zeta(x, y, t) - \frac{1}{2}|\nabla\phi|^2, \end{aligned} \quad \text{on } z = \zeta \quad (2.5)$$

where g is the gravitational acceleration taken as 9.81 m/s^2 , and ζ represents the free-surface elevation. In the present study, the wave slope is assumed small, and the above nonlinear boundary conditions which are to be satisfied on the deformed free surface are further linearized about the calm-water surface $z = 0$, and the linearized free-surface conditions on the calm-water surface S_F are given as

$$\begin{aligned} \zeta_t(x, y, t) &= \phi_z(x, y, z = 0, t), \\ \phi_t(x, y, z = 0, t) &= -g\zeta(x, y, t), \end{aligned} \quad \text{on } S_F. \quad (2.6)$$

In addition, to account for the forward-speed effect of the moving frame of reference of each hull, in the steadily-moving coordinate system $O_j x_j y_j z_j$ of each hull, the above free-surface conditions are rewritten as

$$\begin{aligned} \zeta_t &= \phi_{z_j} + U_j \zeta_{x_j}, \\ \phi_t &= U_j \phi_{x_j} - g\zeta, \end{aligned} \quad j = 1, 2 \quad \text{on } S_F. \quad (2.7)$$

Lastly, the asymmetric flow pattern of the problem gives rise to a non-zero circulation around each moving hull in the horizontal plane. Therefore, Riemann type wake sheets S_{W_1} and S_{W_2} are introduced extending downstream from the sterns [35], as shown in Fig. 2.1. Across the wake sheet, the normal velocity is continuous while the potential is discontinuous. The potentials on the port side and starboard side of the wake sheet S_{W_j} are given by $\phi_j^+(x, z, t)$ and ϕ_j^- , respectively.

Let $C_j(z)$ be a contour in the horizontal plane of constant z that goes around the hull j with starting and ending points at the starboard side and port side of the stern, respectively. The instantaneous hull circulation along $C_j(z)$ is then given by

$$\Gamma_j(z, t) = \oint_{C_j(z)} \nabla \phi \cdot d\mathbf{s} = [\phi_j^+(x_{TE,j}, z, t) - \phi_j^-(x_{TE,j}, z, t)], \quad j = 1, 2, \quad (2.8)$$

where $x_{TE,j}$ is the x -coordinate of the trailing edge (stern) of hull j . It should also be mentioned that Eq. (2.8) and the straight, flat wake sheets as shown in Fig. 2.1 are only accurate for bodies with sharp trailing edges. In addition, in the present study, the circulation of a stationary hull is assumed zero.

Upon solving the above boundary-value problem, the resulting velocity potential ϕ on the hull surfaces is substituted into Euler's Integral [52] for computing the dynamic pressure on each hull in its associated moving frame of reference

$$p = -\rho(\phi_t - U_j \phi_{x_j} + \frac{1}{2} |\nabla \phi|^2 + gz_j), \quad j = 1, 2, \quad (2.9)$$

where ρ is the density of water taken as 1000 kg/m³ in the present study, and the time derivative of ϕ is computed by fourth-order backward finite difference method [53]. Even though we only intend to solve the linearized problem, the terms which are quadratic in velocity in Eq. (2.9) are retained in the pressure calculation, so that the dynamic pressure on hull surfaces can be more accurately computed with minimal additional effort. Then, The longitudinal force, lateral force and yaw moment acting on each hull are then computed by pressure integration on the hull surfaces [40]

$$F_{xj} = \iint_{S_{H_j}} p n_{x_j} dS, \quad (2.10a)$$

$$F_{yj} = \iint_{S_{H_j}} p n_{y_j} dS, \quad (2.10b)$$

$$M_{zj} = \iint_{S_{H_j}} p x_j n_{y,j} dS, \quad j = 1, 2. \quad (2.10c)$$

where F_{xj} , F_{yj} and M_{zj} denote the longitudinal force, lateral force and yaw moment acting about origins of the moving coordinate systems, respectively. It should be mentioned that for a typical marine vessel with slender hull form such as the Series 60 hulls [54] used in the present study, $B/L \sim O(\epsilon)$ and $D/L \sim O(\epsilon)$, the x -component of the normal vector $n_{x,j}$, and the value of leverage arm y are also of $O(\epsilon)$, where $\epsilon \approx B/L0.1$ is a small value representing the beam-to-length ratio of the hull. Therefore, the contributions from longitudinal (axial) force to the yaw moment M_{zj} is $O(\epsilon^2)$ and neglected, though it can be easily included for exactness.

The above mathematical formulations assume no restrictions on the traveling speeds of both component hulls. Therefore, in the present study the classic ship overtaking scenario is modeled with hull 2 initially behind and eventually passing hull 1 on the port side with $U_2 > U_1$ in a parallel, straight course. Meanwhile, a steady-state di-hull system will have $U_2 = U_1$, which will be further explained in the next section.

2.3 Steady-state di-hull wave-making problem

For a steady-state di-hull system in which both component hulls travel with the same constant speed in a fixed formation, the flow field around both hulls can be described in one global moving frame of reference $O_0x_0y_0z_0$ traveling with the di-hull system at speed $U_1 = U_2 = U$, and the time-dependent linearized free-surface conditions of Eq. (2.7) are modified as the combined linearized free-surface condition without time dependency:

$$\phi_{x_0x_0} + k_0\phi_{z_0} = 0, \quad k_0 = g/U^2 \quad \text{on } S_F. \quad (2.11)$$

Upon acquiring the potential values on the free surface, the steady-state free-surface elevation ζ_0 observed in the steadily-moving frame is then given by [55, 18]

$$\zeta(x_0, y_0) = \frac{U}{g} \phi_{x_0}(x_0, y_0, 0). \quad (2.12)$$

In addition, in the steady-state translation of a vessel on calm-water surface, the sinkage and trim of the vessel are affected by the uneven distribution of the dynamic pressure and the misalignment of the direction of thrust and pressure center [56]. In the present study, sinkage h refers to the absolute change of draft at the mid-ship section of the hull, while trim represents the steady-state non-zero pitching angle around the origin of the moving coordinate system of the hull, with the positive sign indicating trimmed by the stern, as shown in Fig. 2.2

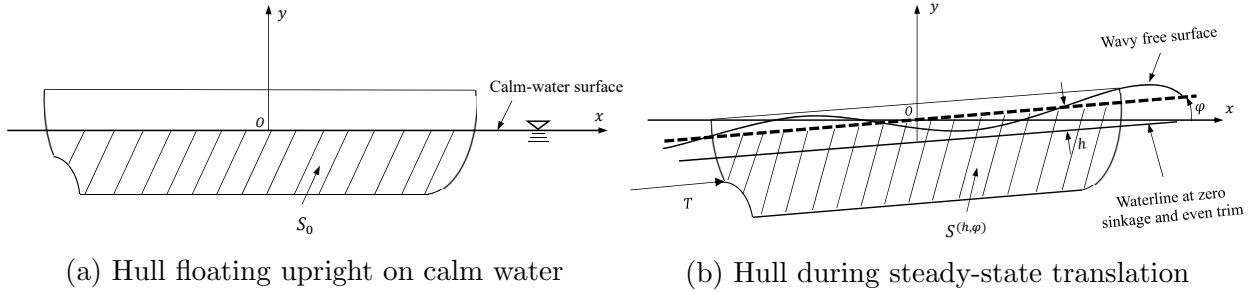


Figure 2.2: Schematics for hull floating still in calm water (left) with the initial wetted-surface area S_0 , and that of the steady-state translation with the associated wetted-surface area $S^{(h,\varphi)}$, where corrections from sinkage h and trim φ are taken into considerations. T indicates thrust applied by propeller.

It should be noted that in the present study, the thrust T is assumed to always act horizontally on the center of hydrodynamic pressure in the x direction, so that its potential influence on the value of trim is neglected. The sinkage and trim can alter the underwater body shape of a hull which in turn affect its wave-making performances, while the change of wetted surface also leads to the change in viscous resistance. In this regard, the sinkage and trim of a component hull are computed in an iterative manner, by solving the following set of equations [56]

$$\begin{aligned}
x_{A_j}\varphi_j + h_j &= \frac{1}{\rho g A_{wp_j}} \iint_{S_{H_j}} p_{0j}(n_z)_j dS, \\
x_{A_j}h_j + \frac{I_{yy_j}}{A_{wp_j}}\varphi_j &= \frac{1}{\rho g A_{wp_j}} \iint_{S_{H_j}} p_{0j}x_j(n_z)_j dS,
\end{aligned}
\quad j = 1, 2, \quad (2.13)$$

where x_{A_j} is the longitudinal center of flotation, A_{wp_j} is the waterplane area and I_{yy_j} is the waterplane area moment of inertia about a transverse axis passing through the center of flotation. p_0 is the steady-state dynamic pressure on the hull surface computed by Eq. (2.9) with the time-dependent term. The acquired values of sinkage and trim will be used to update the underwater shape of the hull, and the velocity potential and pressure distribution are solved again with the updated geometry. Then, Eq. (2.13) will be invoked again for the next iteration until force equilibrium is reached. It should be mentioned that to comply with the experimental set-up in the towing tank tests, each component hull is allowed with individual sinkage and trim in the computation of steady-state di-hull wave resistances.

Wave resistance and wave-pattern analysis

In a steady-state di-hull system, the time-dependent term of Eq. (2.9) is dropped to compute the steady-state dynamic pressure on the hull surface, and the longitudinal force acting on each of the component hulls in the di-hull system is interpreted as its wave-making resistance R_{w_j} in the di-hull formation

$$R_{w_j} = -F_{x_j}, \quad j = 1, 2. \quad (2.14)$$

which can be further normalized as the wave resistance coefficient C_{w_j}

$$C_{w_j} = \frac{R_{w_j}}{0.5\rho U^2 S_j}, \quad j = 1, 2. \quad (2.15)$$

where S_j is the wetted surface area of the hull. While the total wave resistance of a di-hull system can be simply acquired by adding up the wave resistances of its component hulls, in an effort to isolate the resistance component brought by the wave interference, Yeung et al. [11] suggest that the total wave resistance of a di-hull system, R_{w_d} , can also be viewed as the sum of the mono-hull wave resistance of each component hull, plus an extra term representing the influence of wave interference

$$R_{w_d} = R_{w_1} + R_{w_2} = R_{w_{m1}} + R_{w_{m2}} + R_{w_{1\rightleftharpoons 2}}, \quad (2.16)$$

where $R_{w_{1\rightleftharpoons 2}}$ is called the interference resistance introduced by the wave-pattern interference between two sets of ship waves, while $R_{w_{m1}}$ and $R_{w_{m2}}$ are the mono-hull wave resistances of the two component hulls when they are traveling alone in calm water as conventional mono-hull vessels.

Furthermore, wave resistance also represents the rate of energy loss of the moving vessel in creating ship waves [57]; therefore, the wave resistance of a moving hull can be deduced from its ship-wave pattern. It is well known that a conventional mono-hull vessel traveling in isolation on unbounded calm-water surface leaves a wedge-like trailing wave pattern with ship waves propagating in various directions [52], as shown in Fig. 2.3.

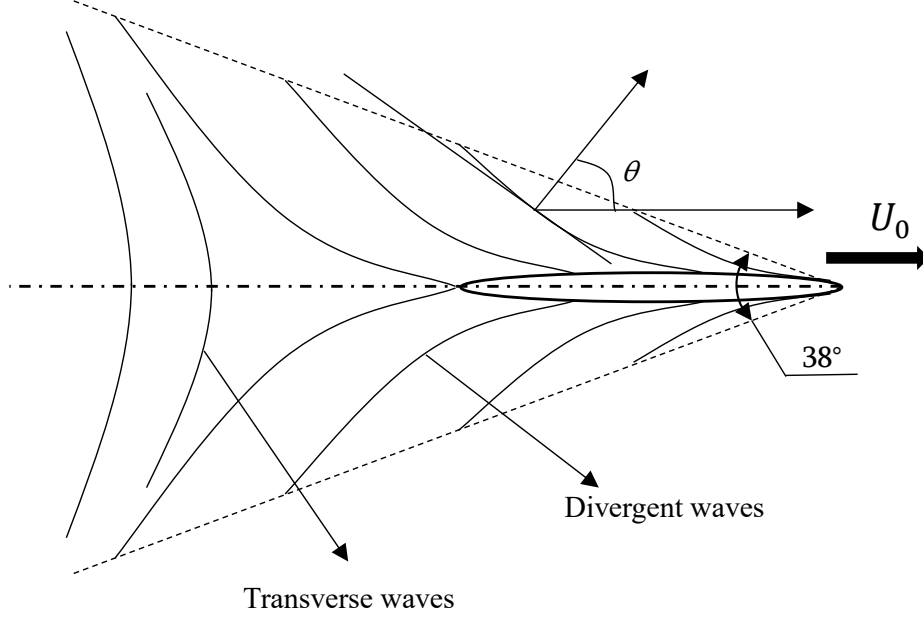


Figure 2.3: Wave-making pattern of an isolated mono-hull traveling in calm water.

Wave angle θ is defined as the angle between the forward direction of the hull and the outward perpendicular of the wave crest, is utilized to represent the propagating directions of the three dimensional ship waves. Previous studies [11, 10, 27] have shown that by a Fourier-type transformation with respect to the wave angle θ , the free-surface elevation ζ is associated with a wave-angle spectrum by the following equation:

$$\zeta(x, y) = \text{Re} \int_{-\pi/2}^{\pi/2} A(\theta) e^{[-ik_0 \sec^2 \theta (x \cos \theta + y \sin \theta)]} d\theta \quad (2.17)$$

where the spectral density $A(\theta)$ is frequently termed as the (complex) amplitude function of the hull. In fact, for a conventional mono-hull, $A(\theta)$ is only determined by the forward speed and geometric shape of the hull and serves as a unique wave-making signature of a given hull form. Eventually, the mono-hull wave resistance of a component hull can be computed from $A(\theta)$ by [26]

$$R_{w_{mj}} = \frac{\pi}{2} \rho U^2 \int_{-\pi/2}^{\pi/2} |A(\theta)|^2 \cos^3 \theta d\theta, \quad j = 1, 2. \quad (2.18)$$

For clarity, the readers must be reminded again of the difference between R_{w_j} and $R_{w_{mj}}$: the former represents the wave resistance of a component hull in the di-hull formation (“component hull wave resistance”) which is subjected to the influences of its nearby counterpart, while the latter represents the wave resistance of a component hull when traveling alone in unbounded fluid domain like a conventional mono-hull (“mono-hull wave resistance”).

Multiple published works [10, 11] make further assumptions that the interference of ship waves only happens in the far field, and the wave pattern of a di-hull system can be constructed by direct superposition of the mono-hull wave patterns of its component hulls. Consequently, the interference resistance $R_{w_{1\rightleftharpoons 2}}$ of a di-hull system as introduced in Eq. (2.16) can also be computed using $A_j(\theta)$ of the component hulls by [11]

$$R_{w_{1\rightleftharpoons 2}} = \pi \rho U^2 \int_{-\pi/2}^{\pi/2} d\theta \cos^3 \theta \cos[k_0 s p \sin \theta \sec^2 \theta] \times \{ \text{Re}(A_1 \overline{A_2}) \cos[k_0 s t \sec \theta] + \text{Im}(A_1 \overline{A_2}) \sin[k_0 s t \sec \theta] \}, \quad (2.19)$$

where the overhead bar represents the complex conjugate of the amplitude function. Consequently, with the help of Eqs. (2.16), (2.18), and (2.19), the total wave resistance of steady-state di-hull system with any hull spacing and forward speed can be computed with the prior knowledge of $A_j(\theta)$ of each component hull.

The above method, which is frequently called the wave-pattern analysis, has been described in detail in many published works [11, 10, 26]. Di-hull systems with various configurations can be easily evaluated with the help of Eqs. (2.18) and (2.19), since instead of repeatedly solving the complex di-hull wave-making problem of a large amount of different configurations, the wave resistance of a di-hull system of any given configurations can be easily computed based on the wave-making signature $A_j(\theta)$ of the component hulls, which can be relatively easily acquired from the conventional mono-hull towing-tank tests [26] or computations [55].

However, the disadvantages of the above method are also evident from its fundamental assumptions. First of all, the far-field assumption which neglect the near-field interactions between component hulls may result in the loss of accuracy when studying di-hull systems with small hull spacing, for which the wave-diffraction effect of hull bodies is significant but not accounted for in the direct superposition of mono-hull wave patterns. More specifically, the propagation of ship waves will be obstructed by the hull bodies, and the reflected waves could have non-negligible effect on the wave pattern and wave resistance of the di-hull system [3]. This effect is explored in depth in Chapter 4. Secondly, the influence of the near-field ship waves are also neglected in the far-field based modeling [11]. Last but not least, the individual wave resistance of each component hull in the di-hull system, R_{w_1} and R_{w_2} , can not be recovered from Eqs. (2.18) and (2.19), which precludes the investigations on the influence of hull configurations on the resistance distribution among component hulls.

Evaluation metric for di-hull wave resistance optimization

By comparing mono-hull wave resistance integral of Eq. (2.18) and the interference resistance integral of Eq. (2.19), it is not hard to discover that while the mono-hull wave resistance in isolation is always non-negative, interference resistance $R_{w_{1\rightleftharpoons 2}}$ can be made negative with selected hull configurations of forward speed and hull spacing. Therefore, with the careful selection of hull configurations of forward speed U and hull-spacing parameters st and sp , the di-hull system can achieve wave-resistance reduction via wave interference, that the total wave resistance of the di-hull system can be made smaller than the sum of the mono-hull wave resistance of its component hulls.

Therefore, one major objective of multi-hull system optimization lies in achieving beneficial wave interference for significant wave-resistance reduction. In this regard, $R_{w_{1\rightleftharpoons 2}}$ of a di-hull system should be made as negative as possible, while the opposite may lead to unfavorable wave interference which increases the total wave resistance of the di-hull system. To represent the influence of wave interference on the wave resistance of each component hull and the entire di-hull system, interference parameter α_1 , α_2 and α_d are defined as follow:

$$\alpha_1 = \frac{R_{w_1} - R_{w_{m1}}}{4R_{w_{m1}}}, \quad \alpha_2 = \frac{R_{w_2} - R_{w_{m2}}}{4R_{w_{m2}}}, \quad \alpha_d = \frac{R_{w_{1\rightleftharpoons 2}}}{2(R_{w_{m1}} + R_{w_{m2}})}. \quad (2.20)$$

Therefore, a negative α_j indicates that the j -th component hull of the di-hull system enjoys smaller wave resistance than its mono-hull wave resistance in isolation; meanwhile, the reduction of wave resistance for the di-hull system is represented by a negative α_d , which indicates that the total wave resistance of the di-hull system is less than the sum of mono-hull wave resistances of its component hulls. An optimal di-hull configuration with maximum wave-resistance reduction should possess the most negative α_d for a given forward speed, while a poor selection of hull configurations may result in positive α_d value that increases the total wave resistance.

Chapter 3

Numerical Simulation by Multi-hull Simple-source Panel Method (MSPM)

In this chapter, the details for numerically solving the boundary-value problem defined in Chapter 2 is discussed. A potential-flow panel method called the Multi-hull Simple-Source Panel Method (MSPM) is developed to compute the steady-state di-hull wave resistance, and simulate the process of unsteady di-hull interactions in moderate-speed overtaking maneuvers. In addition, the implementations of MSPM simulation on a Graphic Processing Units (GPU) to accelerate the computation are also briefly discussed.

3.1 Boundary integral formulation and discretization

Potential-flow panel method has been widely used in the marine industry for simulating irrotational flow of an inviscid and incompressible fluid. By applying Green's third identity, the Laplace equation governing the velocity potential ϕ in the entire fluid domain can be used to form a boundary integral formulation [52, 58] as

$$\begin{aligned} 2\pi\phi(\mathbf{x}, t) = & \int_{S_{H_1} \cup S_{H_2}} \phi(\mathbf{x}', t) G_n(\mathbf{x}; \mathbf{x}') dS - \int_{S_{H_1} \cup S_{H_2}} \phi_n(\mathbf{x}', t) G(\mathbf{x}; \mathbf{x}') dS \\ & + \int_{S_F} \phi(\mathbf{x}', t) G_n(\mathbf{x}; \mathbf{x}') dS - \int_{S_F} \phi_n(\mathbf{x}', t) G(\mathbf{x}; \mathbf{x}') dS + \int_{S_{W_1} \cup S_{W_2}} \Delta\phi(\mathbf{x}', t) G_n(\mathbf{x}; \mathbf{x}') dS \end{aligned} \quad (3.1)$$

where $\mathbf{x} = (x, y, z)$ is a field point of interests in the fluid domain and $\mathbf{x}' = (x', y', z')$ is the integration point. $\Delta\phi$ denotes the dipole strength on the wake sheet S_{W_j} . $G(\mathbf{x}; \mathbf{x}')$ is the simple-source Green function given as [59]

$$G(\mathbf{x}; \mathbf{x}') = \frac{1}{|\mathbf{x} - \mathbf{x}'|} \quad (3.2)$$

To solve the above boundary integral equation, the classic discretization method introduced in [60] is utilized in the present study, and the enclosing surface of the computational domain is discretized into flat, quadrilateral panels with mixed source and dipole distributions. Within each panel, the strength of source or dipole is constant and equal to the that of the collocation point at the center of the panel, and the discretized form of Eq. (3.1) is

given as

$$\begin{aligned}
& \sum_{j=1}^{N_H} [\delta_{ij} - C_{ij}^D] \phi_j + \sum_{j=N_H+1}^{N_H+N_F} \phi_{n_j} C_{ij}^S \\
&= - \sum_{j=1}^{N_H} \phi_{n_j} C_{ij}^S + \sum_{j=N_H+1}^{N_H+N_F} [-\delta_{ij} + C_{ij}^D] \phi_j \\
&+ \sum_{j=N_H+N_F+1}^{N_H+N_F+N_W} \Delta \phi_j C_{ij}^D, \quad i = 1, 2, \dots, (N_H + N_F)
\end{aligned} \tag{3.3}$$

where δ_{ij} is the Kronecker delta accounting for the left-hand-side term of Eq. (3.1). The unknowns to be solved for are placed on the left-hand-side of Eq. (3.3). The indices of panels on the hull surfaces, free surface and wake sheets fall into the ranges of $[1, N_H]$, $[N_H + 1, N_H + N_F]$ and $[N_H + N_F + 1, N_H + N_F + N_W]$, where N_H , N_F and N_W are the numbers of panel on the hull surfaces, free surface and wake sheets, respectively.

Meanwhile, C_{ij}^D and C_{ij}^S are influence coefficients given by

$$\begin{aligned}
C_{ij}^D &= \frac{1}{2\pi} \int_{S_j} G_n(\mathbf{x}_i; \mathbf{x}') dS, \\
C_{ij}^S &= \frac{1}{2\pi} \int_{S_j} G(\mathbf{x}_i; \mathbf{x}') dS.
\end{aligned} \tag{3.4}$$

In the present study, the above integrals are computed by the analytical expressions introduced in Morino et al. [61].

For brevity, the potential and its normal derivative on the hull surfaces are labeled as ϕ^H and ϕ_n^H , and those on the free surface as ϕ^F and ϕ_n^F . With the above notations, Eq. (3.3) can be re-organized as a linear system to solve for the unknowns ϕ^H and ϕ_n^F :

$$\begin{bmatrix} \mathbf{I} - \mathbf{C}^D & \mathbf{C}^S \\ -\mathbf{C}^D & \mathbf{C}^S \end{bmatrix} \begin{bmatrix} \phi^H \\ \phi_n^F \end{bmatrix} = \begin{bmatrix} -\mathbf{C}^S & \mathbf{C}^D \\ -\mathbf{C}^S & -\mathbf{I} + \mathbf{C}^D \end{bmatrix} \begin{bmatrix} \phi_n^H \\ \phi^F \end{bmatrix} + [\mathbf{C}^D][\Delta\phi] \tag{3.5}$$

where $\mathbf{I}$ is the identity matrix. The entries of $\mathbf{C}^D$ and $\mathbf{C}^S$ are computed using Eq. (3.4) with appropriate indices i and j . The values of ϕ_n^H are given by the hull surface condition of Eq. (2.2), and the potential values of free surface ϕ^F are given by Eq. (2.7), while the dipole strength on the wake sheets are provided by the shedding condition, which will be explained in Section 3.3.

3.2 Steady-state di-hull wave resistance computation

The MSPM computation is first utilized to compute the wave resistances of steady-state mono-hull and di-hull systems. The results of steady-state computation not only reveal the influences of hull spacing and forward speed on the wave resistance of a di-hull system, but also can be used as the initial conditions for the unsteady time-domain simulation.

With the linearized combined free-surface condition given by Eq. (2.11), for the steady-state computation, the boundary integral formulation of Eq. (3.1) is modified as

$$\begin{aligned} 2\pi\phi(\mathbf{x}) = & \int_{S_{H_1} \cup S_{H_2}} \phi(\mathbf{x}') G_n(\mathbf{x}, \mathbf{x}') dS + U n_x \int_{S_{H_1} \cup S_{H_2}} G(\mathbf{x}, \mathbf{x}') dS \\ & + \int_{S_F} \phi(\mathbf{x}') G_n(\mathbf{x}, \mathbf{x}') dS + \frac{1}{k_0} \int_{S_F} \phi_{xx}(\mathbf{x}') G(\mathbf{x}, \mathbf{x}') dS + \int_{S_W} \Delta\phi(\mathbf{x}') G_n(\mathbf{x}, \mathbf{x}') dS \end{aligned} \quad (3.6)$$

Then, by regrouping the unknown values to be solved on the left-hand-side of the equation, the linear system of Eq. (3.5) becomes

$$\begin{bmatrix} \mathbf{I} - \mathbf{C}^D & -\mathbf{C}^D - \frac{1}{k_0} \mathbf{C}^S_{x_0 x_0} \\ -\mathbf{C}^D & \mathbf{I} - \mathbf{C}^D - \frac{1}{k_0} \mathbf{C}^S_{x_0 x_0} \end{bmatrix} \begin{bmatrix} \phi^H \\ \phi^F \end{bmatrix} = U_0 [-\mathbf{C}^S][\mathbf{n}_x] + [\mathbf{C}^D][\Delta\phi] \quad (3.7)$$

where the second derivative in the x_0 direction for influence coefficient $\mathbf{C}^S$ is computed by Dawson's upstream finite-differencing scheme [39]. For slender hull with small waterline entrance angle such as the Series 60 hull in the present investigation, the numerical error introduced by such treatment is negligible [40], and the analysis on its error is given below:

To apply finite difference scheme on a curvilinear grid such as the free-surface mesh of the present study, we can first linearly map the physical domain (x, y) with curved grid lines into a Cartesian computational grid (ξ, η) . By the chain rule, we have

$$\phi_{xx} = \phi_\xi \xi_{xx} + \phi_\eta \eta_{xx} + 2\phi_{\xi\eta} \xi_x \eta_x + \phi_{\xi\xi} (\xi_x)^2 + \phi_{\eta\eta} (\eta_x)^2. \quad (3.8)$$

Consistent with the linearized free-surface boundary condition, the flow velocity associated with the disturbance of the moving vessel $(\phi_x, \phi_y) \sim O(\epsilon_1)$, where ϵ_1 is a small number (wave amplitude to wavelength ratio). Furthermore, the vessel in question (Series 60 hull) is slender with small waterline entrance angle; therefore, the distortion of the mesh in the y -direction is small with $\xi_x \sim 1 + O(\epsilon_2^2)$ and $\eta_x \sim O(\epsilon_2)$, where ϵ_2 is again a small number (ship beam to length ratio 0.1). Also, $\xi_{xx} \sim O(\epsilon_2^2)$, $\eta_{xx} \sim O(\epsilon_2^2)$. Hence, only the leading order term, $\phi_{\xi\xi} (\xi_x)^2$ is kept for the computation of ϕ_{xx} :

$$\phi_{xx} = \phi_{\xi\xi} (1 + O(\epsilon_2^2)) + O(\epsilon_1 \epsilon_2) \sim \phi_{\xi\xi}. \quad (3.9)$$

Therefore, the second derivative in the x -direction is taken along the slightly curved free-surface mesh line instead of the actual x_0 -axis. The above treatment has been widely utilized in the computation of linearized wave-resistance of ships [55, 40], and reasonably good agreements with experimental results were observed for slender vessels traveling at moderate to high speeds. Meanwhile, on the hull surfaces, the three-point upstream differencing scheme is utilized for computing derivatives of the potential values because of superior numerical stability and convergence [53].

Mesh generation for MSPM

The computational domain should cover a sufficiently large area of free surface to accurately capture the wave-making behavior of each hull [62]. In addition, sufficient space is required upstream of the bow and downstream of the stern to enforce radiation conditions and ensure unhindered propagation of ship waves. In the mono-hull computation, the hull is placed along the centerline of a computational domain which measures $3L$ and $2L$ in x and y directions, where L is the length of the hull, and the length of domain upstream of bow and downstream of stern measures $0.5L$ and $1.5L$, respectively, as shown in Fig. 3.1a. Based on the offset table of the Series 60 hull, 648 flat quadrilateral panels (36 longitudinally and 18 girth-wise) are applied on the surface of each component hull. Meanwhile, the free surface is discretized by an algebraic advancing-front meshing scheme [63, 64] into 108 rows and 38 columns of panels in x and y directions, respectively. The grid size of the free-surface mesh is uniform in the x -direction, while the y -coordinates on the free-surface mesh on either side of the hull are controlled by an algebraic fitting equation

$$y_{p,q} = y_{p,0} + (y_{p,n_q} - y_{p,0}) \frac{b + 1 - ca^{\chi_q}}{a_q^{\chi} + 1}, \quad 0 \leq p \leq n_p, \quad 0 \leq q \leq n_q \quad (3.10)$$

where p, q are the indices of nodes in the x and y directions, and n_p and n_q are the numbers of nodes in the x and y directions, respectively, e.g. $y_{p,0}$ is the y coordinate of a grid node on the first layer of the mesh, which consists of the straight centerline of the domain and the half-waterplane profile of the hull. Meanwhile, y_{p,n_q} denotes the y coordinate at the far end on the free-surface mesh of either side of the hull. Furthermore, a, b, c and χ_q are the algebraic coefficients given as

$$\begin{aligned} b &= 11, & c &= b - 1 \\ a &= \frac{b - 1}{b + 1}, & \chi_q &= \frac{q}{n_q} \end{aligned} \quad (3.11)$$

Based on Eq. (3.10), the free-surface mesh is generated by marching from the first layer $y_{p,0}$ and evolve gradually into straight-line edge of y_{p,n_q} at the far side of the free surface. In addition, Eq. (3.10) also shows that the distribution of nodes in the y -direction can be adjusted by the value of b . A larger b value will result in more uniform grid size in the y direction, while the opposite tends to result in denser free-surface mesh near the waterline of the hull.

In the di-hull wave resistance computation, the computational domain is divided into two separated patches along the centerline of the computational domain, as shown in Fig. 3.1b. Both patches are equal in sizes in the x and y directions, and each component hull makes equal lateral distance towards the centerline of the computational domain.

In the present steady-state computation, in addition to the mono-hull cases, di-hull systems with 19 different stagger values between 0 and 1.524 m (one hull length) and 15 separation values between 0.34 and 1.524 m are evaluated for wave resistances. The wave resistances for both the mono-hull and di-hull systems are computed for 17 Froude numbers ranging from 0.25 to 0.41.

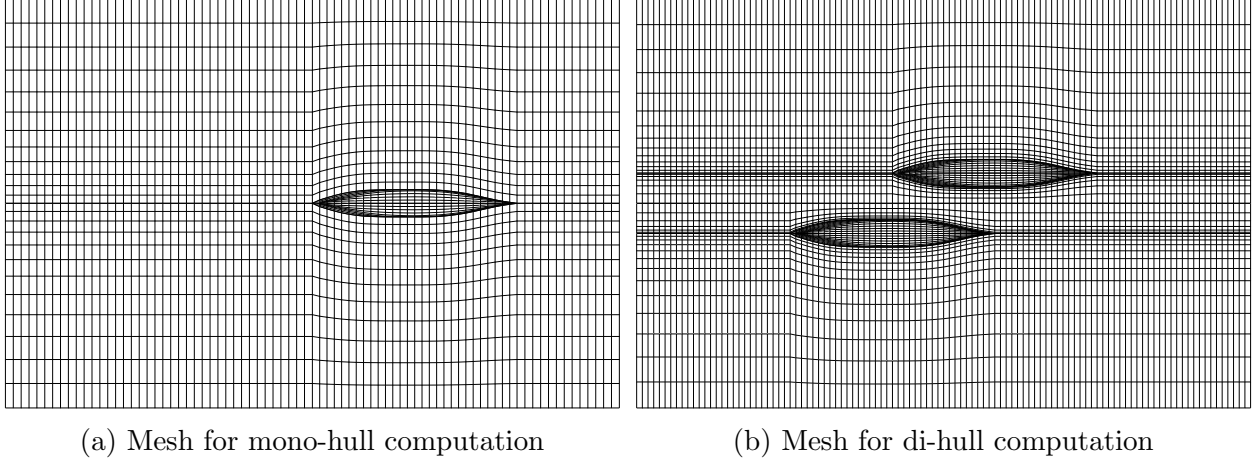


Figure 3.1: Typical meshes of free-surface and hull-surface meshes for wave resistance computation of mono-hull (a) and di-hull system(b).

Implementation of the steady-state Kutta condition

A quasi-two-dimensional Kutta condition is applied on each horizontal slice of the component hull of different z -coordinates. In a steady-state di-hull system, the rate of change of circulation on the hull surface is zero; therefore, the dipole strength on the wake sheet of each moving body section is constant in the streamline direction, and the mesh of wake sheet of a component hull extends from the sterns to the edge of the computational domain in the negative x -direction, as shown in Fig. 3.2.

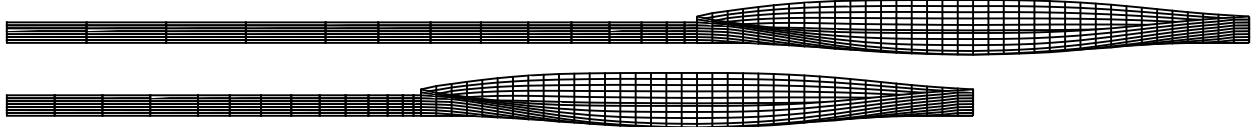


Figure 3.2: Meshes of hull bodies and wake surfaces.

A common practice for specifying the trailing edge flow and determining the dipole strength on wake sheet of a steadily-moving body is given by Morino et al. [61], which requires that the tangential velocity across the stern of a moving hull to be continuous, i.e.

$$\phi_s^+(x_{TE}, z) = \phi_s^-(x_{TE}, z). \quad (3.12)$$

where ϕ_s^+ and ϕ_s^- are the tangential derivatives in the longitudinal direction for the velocity potential on the port side and starboard side, respectively. The above equation is numerically implemented in a finite-differencing manner with the potential values of the panels adjacent to the trailing edge

$$\phi_s^+(x_{TE}, z) \approx \frac{\phi_{H,TE}^+(z) - \phi_{W,TE}^+(z)}{\Delta x} = \frac{\phi_{H,TE}^-(z) - \phi_{W,TE}^-(z)}{\Delta x} \approx \phi_s^-(x_{TE}, z), \quad (3.13)$$

where $\phi_{H,TE}$ and $\phi_{W,TE}$ denote the values of ϕ on the panels adjacent to the trailing edge on the hull and wake sheet, while $+$ and $-$ on the superscripts represent the values on the port side and starboard side, respectively. Eventually, Eq. (3.13) implies

$$\Delta\phi_{H,TE}(z) = \Delta\phi_{W,TE}(z) \quad (3.14)$$

Therefore, the dipole strength of the wake sheet is taken as the potential jump across the stern of the moving component hull section with the same z -coordinate. In this manner, the previously unknown dipole strength $\Delta\phi$ is directly associated with the potential values on the hull surface; therefore it can be solved together in Eq. (3.7), by moving the last term on the right hand side of the equation to the left hand side.

However, Eq. (3.12) only specify the circulation of a moving hull and its trailing-edge velocity in the horizontal plane of constant z -coordinate, while the influences associated with velocity components of other directions, i.e. the cross-flow effect [65] is missing in the modeling of a three-dimensional lifting body, which may lead to abnormal pressure spikes around the trailing edge [53]. Therefore, measures are taken to adjust the dipole strength on wake sheet to contain and minimize the pressure difference across the trailing edge, i.e.

$$\Delta C_p = |C_p^+ - C_p^-| < \epsilon' \quad (3.15)$$

where $C_p = p/(0.5\rho U^2)$ is the pressure coefficients on panels on the stern. The small parameter ϵ' is chosen as 10^{-4} in the present study as the criterion for convergence.

In this regard, a multi-dimensional Newton-Raphson iterative scheme, called the Iterative Pressure-Kutta Condition (IPKC) [53] is adopted to adjust the dipole strength on the wake sheet:

$$[\Delta\phi]^l = [\Delta\phi]^{l-1} - ([\mathbf{J}]^{l-1})^{-1}[\Delta\mathbf{C}_p]^{l-1}, \quad (3.16)$$

where the superscript l is the number of iteration, and $[\mathbf{J}]$ is the Jacobian matrix for the iteration which is given by

$$J_{ij} = \frac{\partial(\Delta C_{p,i})}{\partial(\Delta\phi_j)} \approx \frac{\Delta C_{p,i}^\beta - \Delta C_{p,i}^0}{\Delta\phi_j^\beta - \Delta\phi_j^0}, \quad (3.17)$$

where the variables with superscript 0 are the initial values associated with the solutions acquired with Eq. (3.12), while the values with the superscript β are acquired by applying a small perturbation to the dipole strength on the wake panel.

In short, the procedures for implementing the IPKC can be summarized as follows:

1. Solve the linear system Eq. (3.7) with the dipole strength on the wake sheet given by Eq. (3.14) and acquire initial dipole strength $\Delta\phi^0$ and pressure difference across the stern ΔC_p^0 . Exit the iteration if Eq. (3.15) is already satisfied with the initial solutions, otherwise continue onto the next step.
2. Perturb the dipole strength on the wake sheet by $\Delta\phi^\beta = (1 - \beta)\Delta\phi^0$, in which β is taken as 0.001 in the current study. Solve Eq. (3.7) again with the perturbed dipole strength on the wake sheet and compute the corresponding value of ΔC_p^β based on the updated solutions.
3. Compute all elements of the Jacobian matrix $[J]$ by Eq. (3.17) based on the results of $\Delta\phi^0$, ΔC_p^0 , $\Delta\phi^\beta$, and ΔC_p^β .
4. Modify the dipole strength by Eq. (3.16), solve the linear system of Eq. (3.7) again with the modified dipole strength on the wake sheet, compute $[\Delta C_p]$ based on the resulting velocity potential for the next iteration.
5. Repeat Step 4 until Eq. (3.15) is satisfied and output the solutions of velocity potential at this moment as the final solution.

Strictly speaking, matrix $[J]$ should be updated during each iteration to accelerate the convergence of computation. However, it is observed that the elements in $[J]$ change trivially with each iteration step; therefore, matrix $[J]$, once computed, is kept constant throughout the iteration process.

3.3 Computation of unsteady di-hull interactions in the moderate-speed overtaking maneuvers

The moving-patch method

The mesh utilized in the unsteady simulation is similar to that of the steady-state computation. However, minor adjustments are needed to accommodate the relative motions between the two component hulls.

In this regard, a novel moving-patch method is developed and implemented [62] for simulating unsteady di-hull interactions. The free-surface mesh of the computational domain, which is still generated by the algebraic meshing scheme introduced above, is also divided into two equal sized patches as in the steady-state cases, and each patch moves with the corresponding hull with the same forward speed. In addition, the size of each patch is substantially expanded to accommodate the relative motion between hulls and the downstream wave-damping zone [66] to gradually reduce the ship waves approaching the downstream boundary of the computational domain and prevent wave reflection. Eventually, each patch measures eight and three ship lengths in the x and y directions, respectively, as shown in Fig. 3.3.

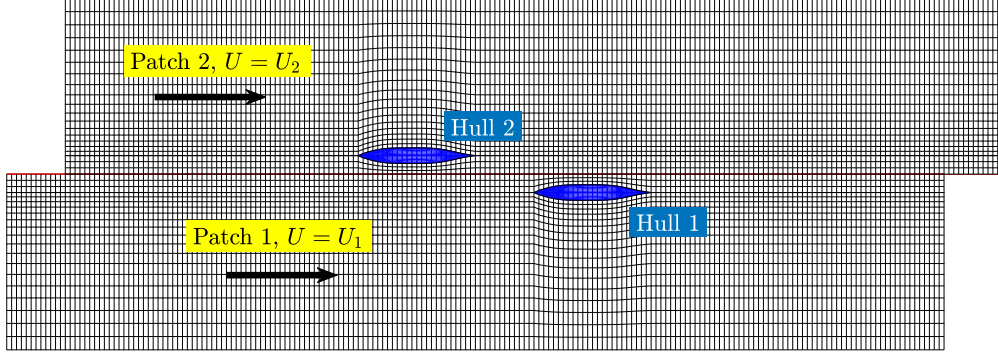


Figure 3.3: Schematic of the mesh for unsteady computation. The free-surface mesh is divided into two the moving patches. Grid density is reduced for illustration purpose.

In this manner, at each time step one simply need to update the x coordinates of the mesh on each patch by the distance traveled by the corresponding hull, and the need to re-mesh the entire computational domain is eliminated. In addition, since the potential value and free-surface elevation are constant within each panel, alignment of the two mesh patches along the centerline of the domain is unnecessary.

Furthermore, in the unsteady overtaking simulations, the sinkage and trim of each hull are kept constant and equal to their initial values given by the steady-state computation (see Section 3.3); therefore, the relative position between any two panels within the same patch is kept constant throughout the entire simulation. Consequently, one only need to update half of the influence coefficients in Eq. (3.5) for each time step, which significantly reduces the computing time.

Time-marching scheme and the initial conditions

In each time step, the linear system of Eq. (3.5) is solved for the values of ϕ^H and ϕ_n^F . The right-hand-side value of ϕ_n^H is given by the hull surface condition of Eq. (2.2), while the values of ϕ^F are taken as those of the present time step. Then, the computed results of ϕ_n^F are substituted into the kinematic free-surface condition of Eq. (2.7) to update the free-surface elevations ζ , which is subsequently invoked by the dynamic condition of Eq. (2.7) to update ϕ^F for the next time step. For different patches, the correct forward speed U_j must be used for each patch to update the values of ϕ^F and ζ .

In addition, ϕ^F and ζ of each patch are updated by the predictor-corrector method in time with Forward Euler as the predictor and trapezoidal method as the corrector, while the x derivatives of ϕ^F and ζ are computed by second-order upstream finite difference scheme introduced in [67], which is preferred for its better stability and lower numerical damping [68, 69]. Moreover, in all computational cases hull 2 starts at the initial position of $st/L = -3$, and the overtaking process ends with $st/L = 1.5$.

Previous studies on the time-domain simulations of ship-to-ship interactions on calm

water usually start with stationary ships as the initial condition, which accelerate to the designated speeds before overtaking/encountering takes place. Alternatively, the ships can also start impulsively from rest at the designated overtaking speed [69]. However, in the computations of higher maneuvering speeds, the former approach will require a significant amount of simulation time and large computational domain to accommodate the acceleration stage, while the latter approach may suffer from numerical instability brought by the sudden disturbance of the calm-water surface.

Therefore, in the present unsteady computation of moderate-speed overtaking maneuvers, the initial conditions are given by the solutions of the steady-state computations introduced in Section 3.2. In the case of hull 2 passing a stationary hull 1 (termed as “zero-speed overtaking”), i.e. $U_1 = 0$, the initial conditions are given by solving the steady-state mono-hull Neumann-Kelvin problem of hull 2 moving at the ultimate overtaking speed U_2 alone in the fluid domain made by the two patches. The presence of hull 1 is ignored since the distance between hulls is very large at this moment and the interaction effect is negligible. On the other hand, if hull 2 is to overtake a moving hull 1 with $U_2 > U_1 \neq 0$, the computation of steady-state di-hull systems is conducted first to provide the initial conditions, in which two hulls moves at the same, slower speed of the overtaken hull i.e. $U = U_1$. Then, at the beginning of the unsteady computation, hull 2 speeds up to the designated overtaking speed U_2 with a constant acceleration of 0.5 m/s^2 before the overtaking maneuver takes place. The accelerating process usually finishes when hull 2 is still more than two ship lengths downstream of hull 1 ($st/L < -2$), and then hull 2 overtakes hull 1 with a constant forward speeds. Furthermore, the sinkage and trim of each hull in the time-domain simulation are kept constant as those of the initial values provided by the steady-state computation.

Implementation of the unsteady wake-shedding condition

To account for the vortex shedding of the moving hulls in the unsteady simulations, dipole-type wake sheets [70] are introduced extending from the sterns of the moving hulls towards the negative x -axis. At each time step, a column of wake panels are created adjacent to the stern, and the grid size in the x -direction is equal to the distance traveled by the hull in one time step as shown in Fig. 3.5. At time step k , the strength of the newly-created column of wake panels at the trailing edge, $\Delta\phi_j^k(z)$ (see Fig. 3.5), is given by

$$\Delta\phi_j^k(z) = \Gamma_j(z, t^k), \quad j = 1, 2, \quad (3.18)$$

where t^k is the time at step k , and Γ_j is given by Eq. (2.8). The derivation of Eq. (3.18) and the implementations of the two-dimensional shedding condition are briefly outlined below.

We model the shedding of vorticity at the stern of a moving hull in a quasi-two-dimensional manner, by applying a two-dimensional shedding condition at each horizontal hull slice of constant z -coordinate. Consider a two-dimensional lifting section at a vertical position z traveling in the positive x -direction at constant speed U . In a moving frame of reference

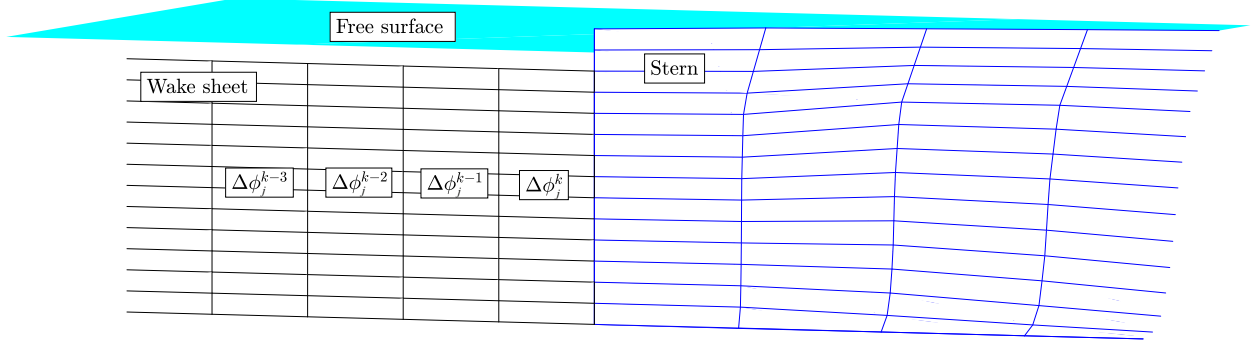


Figure 3.4: Schematic of hull mesh at stern beneath calm-water surface and wake sheet.

Oxy attached to the section, the trailing edge of the lifting section is fixed at $x = x_{TE}$ and $y = 0$.

The rate of change of the bound circulation $\Gamma(z, t)$ of the moving body section at z and the strength of the shed vortex wake sheet $\gamma_w(x, z, t)$, located along $y = 0$, should satisfy [71]

$$\frac{d\Gamma(z, t)}{dt} = -U\gamma_w(x_{TE}, z, t). \quad (3.19)$$

where $\gamma_w(x, z, t)$ also gives the tangential velocity jump in the x -direction across the wake sheet:

$$\gamma_w(x, z, t) = \frac{\partial\phi^-}{\partial x} - \frac{\partial\phi^+}{\partial x} = -\frac{\partial(\Delta\phi)}{\partial x}. \quad (3.20)$$

where ϕ^+ and ϕ^- denote the potential values acquired with $y = 0^+$ and $y = 0^-$, i.e. on the port side and starboard side of the wake sheet, respectively. In addition, to ensure that the linearized pressure is continuous, the potential jump across the trailing edge and the wake sheet $\Delta\phi(x, z, t)$ should satisfy

$$\frac{\partial(\Delta\phi)}{\partial t} - U\frac{\partial(\Delta\phi)}{\partial x} = 0 \quad (3.21)$$

which also implies that the wake sheet is advected downstream at constant velocity U . Consequently, Eqs. (3.19), (3.20), and (3.21) lead to

$$\frac{d\Gamma(z, t)}{dt} = \frac{\partial\Delta\phi(x_{TE}, z, t)}{\partial t}. \quad (3.22)$$

Integrating both sides of the above equation with respect to time, one will get

$$\Gamma(z, t) - \Gamma(z, 0) = \Delta\phi(x_{TE}, z, t) - \Delta\phi(x_{TE}, z, 0). \quad (3.23)$$

Furthermore, if initially (at $t = 0$) the bound circulation of the body section at z is assumed to be zero, and no wake sheet is present, i.e. $\Gamma(z, 0) = 0$ and $\Delta\phi(x_{TE}, z, 0) = 0$, we get

$$\Gamma(z, t) = \Delta\phi(x_{TE}, z, t). \quad (3.24)$$

Therefore, at each time instance, the dipole strength of the wake panel aft of the trailing edge is seen to be the bound circulation of the body section at that moment, which is given by the potential jump across the trailing edge. The unsteady shedding condition of Eq. (3.18) is readily obtained from Eq. (3.24) with the assumption that any girth-wise component of vorticity being negligible. Similar relations were also given by Sears [71] and had been previously applied by Yeung & Tan [33].

In addition, the pressure continuity condition across the wake sheet dictates that its potential jump should stay temporally constant once shed in the earth-fixed coordinate system. Therefore, even though the circulation of the moving hull is time-dependent, the dipole strength of the wake panels, once created, remains constant. In addition, a small gap is left between the top rows of the wake sheets and the free surface because vortices would quickly dissipate on the free surface [72].

3.4 Implementation of GPU-accelerated computation

Potential-flow panel method computation usually requires much smaller computing cost than its viscous-fluid CFD counterparts. However, in order to develop a global understanding of the performances of various multi-hull systems, considerable computing power and efficient implementation are still very much necessary since the configurations to be cover are generally numerous.

Table 3.1: Specifics of the workstation PC and GPU used in the present study.

CPU	AMD Threadripper 1950X, 16 cores @ 3.4GHz
GPU	Nvidia GTX Titan Black
PC (host) memory size	64 Gigabytes
GPU (device) memory size	6 Gigabytes
GPU processing power (double precision)	1707 GFLOPS

Therefore, in the present study, the MSPM computation is accelerated with the parallel computing capability of a GPU. While parallelism on CPUs are limited to a few cores optimized for sequential serial processing, GPU adopts massively parallel architecture consisting of thousands of smaller, more efficient threads designed for handling multiple tasks simultaneously [73]. Threads are grouped into several blocks (usually referred to as “thread block”) and communicate with each other in the same block with fast-accessing shared memory. Eventually, the massive parallelism of a GPU is achieved by performing computation on multiple thread blocks simultaneously.

In the MSPM computation, the generation of influence matrices in Eqs. (3.5) and (3.7) can be substantially accelerated with the help of a GPU, since the computation of influence coefficients is strictly independent between any pair of source panel and field point. In

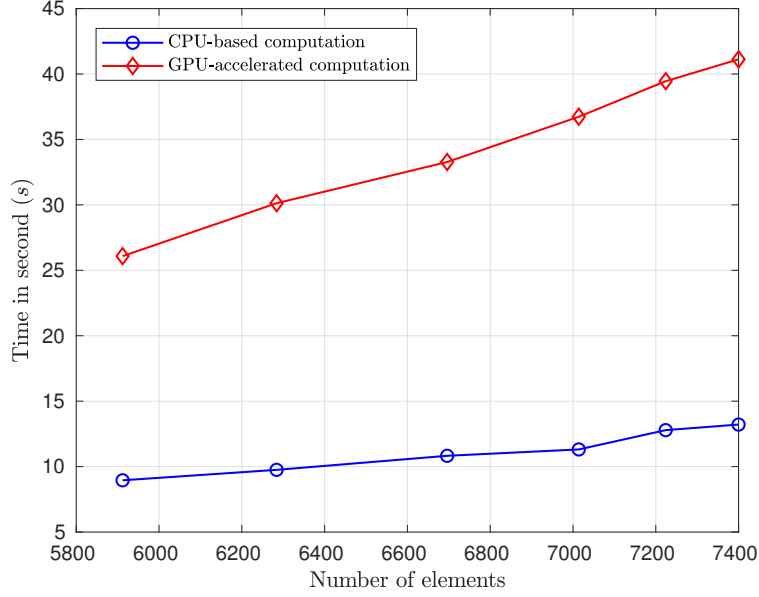


Figure 3.5: Comparison of computing time with different hardware implementations of the MSPM computation.

addition, since the computation of influence coefficients usually takes up a large portion of the total computing time of panel-method computation [40], the entire MSPM computing time can be significantly reduced with the help of a GPU, and a large amount of multi-hull configurations can be evaluated in little time.

In the present study, the MSPM computation was carried out on a workstation PC with its specifics listed in Table 3.1. Compared with the pure CPU-based implementation, three to five times of acceleration is achieved with the GPU-assisted computation, as shown in Fig. 3.5. Fast computation is crucial for the present study: first of all, multiple time-domain simulations of the moderate-speed overtaking maneuvers can be quickly carried out to reveal the influence of speed differences between hulls in the overtaking process, which will be further examined in Chapter 6; secondly, large amounts of computational results on the steady-state calm-water performances of trimarans are required for the training and testing of a Deep Neural Network (DNN) for the optimization of trimaran configurations, which will be discussed in Chapter 7.

Chapter 4

Experimental Investigations by Towing-tank Tests

The computational results were assessed by comparing against experimental measurements by towing-tank tests. In addition, the physical observations from model-scale testing also serve as the basis for some of the assumptions made in Chapter 2. In this chapter, the general set-up of the experimental wave-resistance measurement of steady-state di-hull systems and the tests on the moderate-speed overtaking maneuvers are described. The equipment set-up, testing matrix, experimental methodologies and a novel hybrid approach which combines experimental results with analytic formulations are illustrated and discussed.

4.1 Hardware design and set-up

The experiments were conducted in the Physical-testing Facility of Richmond-Field Station, University of California at Berkeley. The tank measures 80m, 2.55m and 1.8m in length, width and water depth, respectively. A flap-type wave-maker is located at one end of the tank, while the other end is installed with a wave-dissipating beach. In the experiments, the model hulls were towed by a carriage mounted on the rails resting on the two side walls of the tank, which is powered by an electrical motor with a digital controller, and capable of operating up to a speed of 1.8 m/s under manual or remote control.



Figure 4.1: Towing tank and the carriage of Richmond-Field Station.

Adjustable towing frame system

In order to attach the hull models to the carriage with adjustable hull spacing, a special towing frame was designed and fabricated using mostly aluminum bars and plates, as shown in Fig. 4.2.

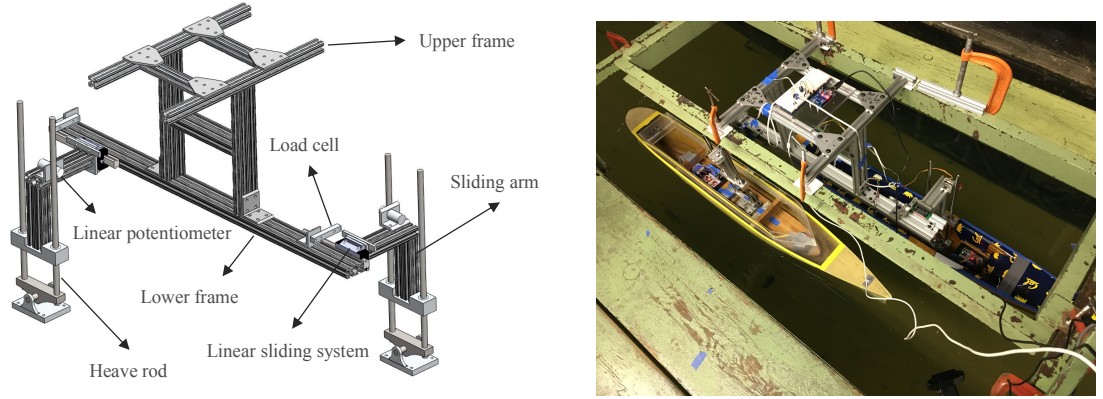


Figure 4.2: Solidworks drawing and photo of the towing-frame design and installation.

The adjustable towing frame used in the present study were an upgraded from its first-generation model described by Yu [22]. The main body of the towing frame is made out of aluminum 80/20 bars. The upper frame is clamped down on the carriage to sustain the weight of the entire frame, while the lower frame is made out of a 4-feet long double-extrusion 80/20 bar which serves as the backbone for hosting measuring devices. The resistance-measurement unit is constituted with a sliding system consisting of a precision rail and linear slider, a beam-shape strain gauge load cell and a sliding arm which holds the heave rods connected to the hull model beneath. During the experiments, the hull model was pushed backward together with the sliding arm by the resistance, and the slider on the precision rail then made contact with the load cell. The Wheatstone bridge on the load cell then transformed the deflection of the beam to voltage signals, which was then captured and amplified by a Phidgets bridge panel at a sampling rate of 100Hz. The processed signals were eventually transmitted to the connected laptop computer to output force readings. Two sets of such resistance-measurement units were installed on both sides of the lower frame; therefore, the resistance of each component hull in the di-hull system can be measured individually, allowing the study of resistance distribution among component hulls.

Before the experiments, the load cells on the towing frame were statically calibrated by applying loads of known magnitudes using electronic weight gauge at the bottom of the heave rod in the negative x -direction. The voltage readings at this moment were then collected and compared with the known applied loads, and the linear relation between the voltage and the force can be derived. However, it should be noted that the sliding system can only make contact with the load cell after sliding a small distance on the rail; therefore, the friction of the rail, though very small, will also affect the force measurement in the experiments. In this

regard, the friction of the rail was determined and compensated before deriving the linear force-voltage relation, and satisfactory linearity with a correlation coefficient of 0.999 was observed in the end.

Hull models and the normalization of hull-spacing parameters

A pair of identical Series 60 bare hull models [54] with 1/80 scale ratio were utilized in both the experimental and computational investigations of the present research. The detailed geometric features of the hull models are given in Table 4.1.

Table 4.1: Geometric parameters of the hull models

Block coefficient	$C_B = 0.6$
Length between perpendiculars	$L = 1.524$ m
Beam	$B = 0.2032$ m
Draft (at design load waterline)	$D = 0.0813$ m
Displacement	$\Delta = 13.8$ kg
Wetted surface area (of zero trim and sinkage)	$S = 0.396$ m^2
Scale ratio	1/80
Longitudinal center of flotation	$x_A = -1.6\%L$ from midship

One of the hull model is made of fiber-glass, while the other of wood, as shown in Fig. 4.2. Both models are coated with resin and protection paint to reduce erosion and frictional drag; therefore, the influence of surface roughness on ship resistance was neglected in the present study [48]. Both hull models were ballasted to the design waterline with zero sinkage and even keel. Turbulence-stimulators were installed at the first station from bows of both hull models to generate turbulent flow environments on the hull surfaces.

For the convenience of comparison and illustrations, the hull configuration parameters, stagger, separation and forward speed are all normalized in the following manner

$$\overline{st} = \frac{st}{L}, \quad \overline{sp} = \frac{sp}{L}, \quad F_n = \frac{U}{\sqrt{gL}}, \quad (4.1)$$

where g is the gravitational acceleration, taken as 9.81 m/s² in the present study, and F_n is the Froude number.

Sinkage and trim measuring device

The present ship-motion measurement system mainly consists of the Arduino micro-controller and compatible micro-electro-mechanical (MEMS) modules. The open-source Arduino micro-controller provide highly versatile functions in data collection, processing and transmission

with satisfactory sampling rate and very low costs. It can also be used to power other low-voltage electronic devices (up to 12V), pass on commands based on C-based computer code and read both digital and analog outputs from the measuring devices and interfaces with computers via USB connection. Besides, the very compact size, low power consumption, modularity, convenient programmability and durability make Arduino devices ideal for model-scale experiments.

The sinkage of each hull model was measured by the potentiometer mounted on the sliding arm. The upper part of the heave rod was in contact with a wheel connected with the potentiometer. When the hull model moved downward (in the negative z -direction), the connected heave rod would drive the potentiometer to rotate, and the resulting changes of voltage of the potentiometer was captured by a Arduino Uno controller and transmitted to the laptop to transform into sinkage readings. In addition, a layer of rubber was attached on each of the contact wheels of the potentiometers to prevent sliding of the heave rod on the contact surface.

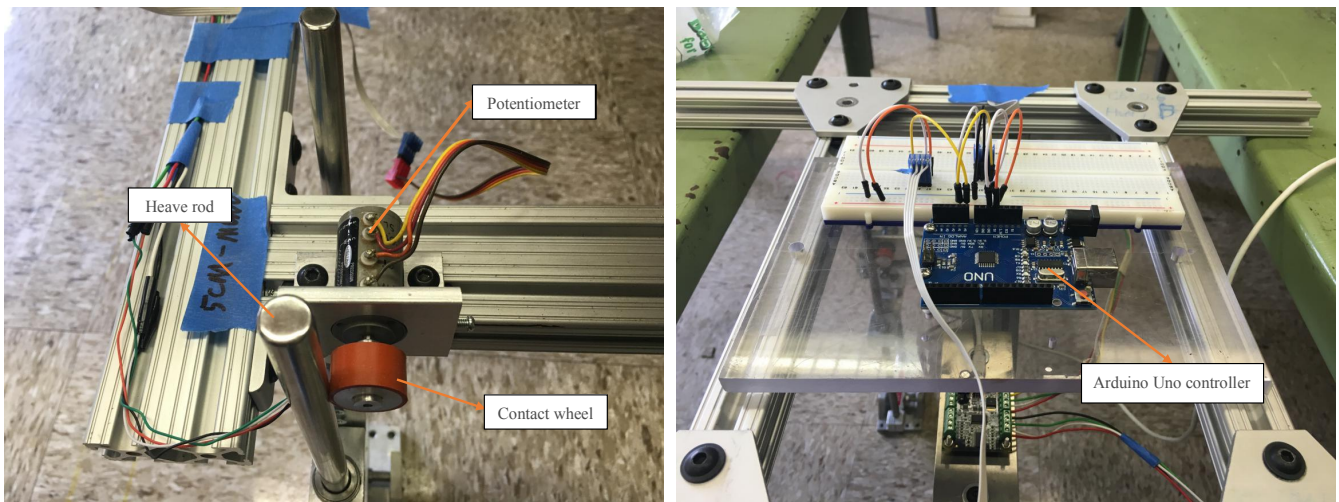


Figure 4.3: Installation of the potentiometer for sinkage measurements.

The measurement of trim of each hull model is less straightforward, since the conventional method of measuring the value of trim by taking the differences of draft between bow and stern is not feasible in the present di-hull experiment due to hardware limitations [74]. To this end, a novel wireless gyroscope/accelerometer system called the Ship Motion Analysis Radio Transmission (SMART) system is developed. The key measuring component for the SMART systems is the MPU-6050 sensor, which contains a MEMS accelerometer and a gyroscope on a single chip. It provides measurements of very high accuracy and sensitivity, with up to 131 least-significant-digit (LSB) per degree per second and a nonlinearity of 0.2%. It contains 16-bits analog-to-digital conversion hardware for each measuring channel and interfaces with Arduino controller with I2C-bus [75]. The MPU-6050 chip can capture signals in the x , y , and z direction at the same time and store the collected data in its 1024-byte first-in-first-out (FIFO) buffer to be read and processed by the controller. Another

important part of the SMART system is the nRF-2401 radio-transmission module. It is a low-voltage (around 1.9V) 2.4-2.5GHz ISM-band radio transmission chip with maximum data rate of 1000 kilobyte per second (kbps), with the maximum operating range up to 50m.

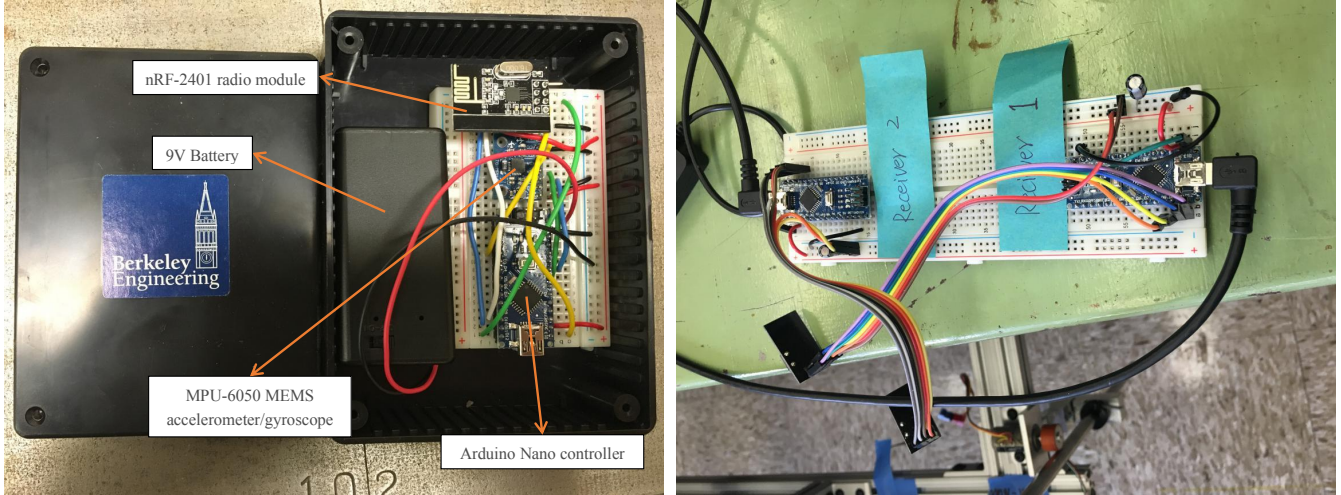


Figure 4.4: Onboard transmitting (left) and receiving (right) module of the SMART system.

The onboard module of the SMART system consists of an MPU-6050 chip wired with an Arduino Nano controller and an nRF-2401 radio chip set, and the whole system was powered by a 9V battery and housed in a water-proof plastic case placed near the mid-ship section of each hull model. The collected data of the onboard module was then sent to the receiver end of another set of nRF-2401 module and an Arduino Nano controller connected to a laptop computer. With the application of wireless measuring and transmission system, the potential influences of the data cable in hindering the motion of hull models were eliminated, and the average sampling frequency was discovered to be around 30Hz.

The potentiometers and SMART system were also calibrated statically before the experiments. The potentiometers were calibrated by letting the heave rod slide a known distance and measuring the corresponding change of voltage signal, and the correlation coefficient was discovered to be 0.998. The SMART system was calibrated by first placing the device on an absolute flat, horizontal table in the machine shop, then the calibration script was run to collect the reading of each channel and automatically applied as offsets in the measurements later on.

4.2 Di-hull steady-state resistance towing-tank tests

Experimental set-up and test matrix

The first set of experiments conducted in the present study were the measurements of steady-state di-hull resistance in calm water. Previous studies have suggested that di-hull systems

consist of identical component hulls with a half-ship-length stagger traveling at Froude number $F_n = U/\sqrt{gL}$ around 0.4 should enjoy significant resistance reduction because of wave interference. The two sets of transverse ship waves of the two component hulls align peak to trough and achieve maximum wave cancellation [12, 57]. Therefore, the di-hull configurations with $\overline{st} = 0.5$ and two different separation values were experimentally evaluated, as shown in Table 4.2. Before conducting the di-hull tests, one of the component hull was selected for the conventional mono-hull towing-tank test for measuring mono-hull resistance in calm-water in isolation. Eleven tested forward speeds were selected between 1.006 and 1.600 m/s, and the corresponding Froude number were between 0.262 and 0.414.

Table 4.2: Hull configurations of the present towing-tank tests.

Test No.	st (m)	sp (m)	$\overline{st} = st/L$	$\overline{sp} = sp/L$
1	Mono-hull tests of the component hulls in isolation			
2	0.762	0.35	0.500	0.230
3	0.762	0.45	0.500	0.295

In addition, to mitigate the potential influence from the tank wall, in the mono-hull tests the hull model was towed along the centerline of the tank, while in the di-hull towing-tank tests the two hull models made equal lateral distance towards the centerline of the towing tank, as shown in Fig. 4.5a and 4.5b.

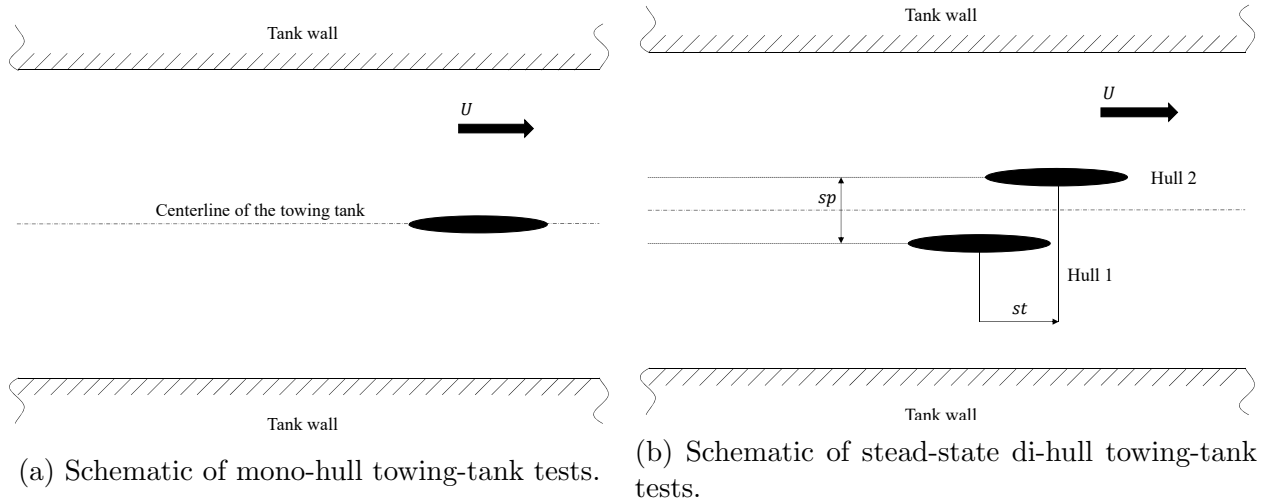


Figure 4.5: Set-up of the towing-tank experiment for the steady-state resistance measurements of isolated mono-hull and di-hull systems.

At the beginning of the tests, hull models were placed at one end of the towing tank and accelerated by the connected carriage until designated steady-state forward speeds were reached. Then the hull models kept traveling with a constant forward speeds for 20 to

30m before decelerating to a full stop and returning to the starting position. An interval of 15 to 20 minutes was taken between each test for the dissipation of disturbance on the water surface. The resistance data was recorded during the entire tests, and the steady-state sections of the captured signals were extracted and processed with a low-pass filter and averaged as the readings for calm-water resistances. The measurements of sinkage and trim follow similar procedures except that a MATLAB-based script was used instead to collect the voltage signal of the potentiometers and the SMART system.

Extraction of wave resistance from experimental measurements

The resistance measured directly in the towing tank tests can be decomposed into various components including skin friction, eddy resistance from flow separation, and wave-making resistance, etc.. Thus, measures must be taken to extract the wave-resistance component from the measured total resistance data. According to the recommendations of 1978 15th International Towing Tank Conference (ITTC), the three-component decomposition of calm-water ship resistance can be expressed as [23]:

$$R_t = (1 + k)R_f + R_w \quad (4.2)$$

where R_t and R_f are the total resistance and frictional drag, respectively. k is the form factor which accounts for form drag resulting from flow separation that stems from the curvature of the hull surface. To isolate the wave resistance of a tested hull model from the experimentally measured total resistance, flat-plate assumption was applied [23] that the frictional drag R_f was approximated by that of a flat plate with the same wetted surface area as the hull model, namely

$$R_w = R_t - (1 + k)R_f, \quad (4.3a)$$

$$R_f = \frac{1}{2}\rho U^2 S_H C_f. \quad (4.3b)$$

where the form factor, k , was assumed zero in the present study, since the hull models were small and slender without any appendages. The coefficient of frictional drag C_f was given by the International Towing Tank Conference (ITTC) line-correlation formula as

$$C_f = \frac{0.075}{(\log Re - 2)^2}. \quad (4.4)$$

where $Re = UL/\nu$ is the Reynolds number with ν being the kinematic viscosity of water.

It should be further stressed that in the present study, the wetted surface areas for the normalization of both experimental and numerical resistance data were corrected by the measured or computed sinkage and trim.

4.3 Hybridization of theory and experiments for di-hull wave-resistance prediction

Computation of amplitude function from experimental wave-cut measurements

As mentioned in Chapter 2, with the knowledge of the amplitude function $A(\theta)$ of each component hull, the di-hull wave resistance of any hull configurations can be derived. In addition, $A(\theta)$ can be directly computed from thin-ship approximation [11], or indirectly deduced from the measurements of free-surface elevations along a longitudinal cut of the mono-hull wave pattern [26]. Therefore, the experimentally obtained $A(\theta)$ can be substituted into Eq. (2.18) for the computation of wave resistance. Such a method, which is called the longitudinal wave-cut method (LCM) has been widely utilized for predicting wave resistances of vessels with symmetric wave-making patterns, such as the conventional mono-hull vessels [76], catamarans [19] and trimarans [77]. However, in the present study, with the help of Eqs. (2.16) and (2.19), the conventional LCM can be further extended to predict the wave resistance of di-hull systems with any hull configuration.

The procedure for experimentally acquiring the amplitude function of each hull closely follow the ones introduced in Eggers et al. [26]. As $A(\theta)$ is discovered to be highly oscillatory when θ approaches $\pm\pi/2$, for convenience and accuracy of integration of resistance integrals of Eq. (2.18) and (2.19), $A(\theta)$ is transformed into refined amplitude functions $G(u)$ and $F(u)$ by

$$G(u) + iF(u) = \frac{4k_0\pi \cos^3 \theta}{1 + \sin^2 \theta} A(\theta) \quad (4.5)$$

where u is called the longitudinal wave numbers defined as

$$u = \sec \theta \tan \theta. \quad (4.6)$$

Accordingly, by utilizing refined amplitude functions $G(u)$ and $F(u)$ acquired from LCM, the mono-hull wave resistance integral Eq. (2.18) is re-written as

$$R_{w_j} = \frac{\rho U^2}{8\pi k_0^2} \int_0^\infty \{F_j^2(u) + G_j^2(u)\} \frac{\sqrt{1+4u^2}}{1 + \sqrt{1+4u^2}} du \quad j = 1, 2 \quad (4.7)$$

and Eq. (2.19) for interference resistance can be re-written as

$$\begin{aligned} R_{w_{1\Rightarrow 2}} = & \frac{\rho U^2}{4\pi k_0^2} \int_0^\infty du \frac{\sqrt{1+4u^2}}{1 + \sqrt{1+4u^2}} \cos(k_0 s p u) \\ & \times \{(G_1 G_2 + F_1 F_2) \cos(k_0 s t s(u)) \\ & + (F_1 G_2 - G_1 F_2) \sin(k_0 s t s(u))\} \end{aligned} \quad (4.8)$$

Some interesting conclusions can be drawn from the above mathematical derivations. First of all, for di-hull systems made out of identical mono-hulls, the last parts of the integral

of Eq. (4.8) vanishes, and the expressions for $R_{w1\rightleftharpoons 2}$ and α are symmetric about $st = 0$; Secondly, although the amplitude functions $G(u)$ and $F(u)$ are only determined by the geometric shape of the hull, they are also functions of wave angle θ and Froude number F_n . However, for di-hull systems made out of mono-hulls with identical shape, maximum wave interference effect, be it beneficial or detrimental in terms of wave resistance reduction, can be expected when two hulls are of the same dimension. Last but not least, for di-hull systems with zero stagger, the interference between pure transverse ship waves with $u = 0$ stays constant for a given forward speed, regardless of the separation value chosen. All of these observations were also made in Tuck et al. [10], where the wave amplitude functions of the two mono-hulls in the di-hull systems are directly superposed.

The experimental set-up for taking longitudinal wave-cuts of the mono-hull wave patterns will be introduced in the next section. Meanwhile, on obtaining $G(u)$ and $F(u)$ from experimental wave-cut measurements during mono-hull towing tank tests, one can refer to Eggers et al. [26], which is beyond the scope of the present study. The use of experimentally determined wave-making signatures should not only to a large degree eliminates the influence of thin-ship approximation in the direct computation [11], but also incorporate the influence of sinkage and trim of the hull in the experiments. In addition, after acquiring $G(u)$ and $F(u)$ of each mono-hull for different forward speeds from wave-cuts of mono-hull towing tank tests, the interference resistance as well as the interference parameter α of di-hull systems with any hull spacing and forward speeds can be quickly determined with the help of Eqs. (4.7) and (4.8), which provides a convenient tool for conducting fast, large scale search of optimal multi-hull configurations.

Wave-cut measurement in mono-hull towing-tank tests

Apart from resistance measurements, the free-surface elevation along two longitudinal cuts were also measured in the mono-hull towing tank tests. Two NHL capacitor-type wave probes were mounted on an aluminum boom cantilevered from one side of the towing tank; therefore, two wave cuts at different lateral locations on one side of the symmetric wave pattern of the mono-hull model were taken, as shown in Fig. 4.6. It should be mentioned that one wave-cut will be sufficient for the wave-pattern analysis; therefore, the measurements of extra wave cuts at another lateral location serve the purpose of cross-validation of the experimental measurements. Wave-cut records were truncated at the point where the reflective waves from the port side tank wall reach the wave probes, and the limited wave-cut length for analysis is called effective length L_{eff} . The lateral distances of wave-cuts with respect to the centerplane of hull model and the corresponding effective length are given in Table 4.3.

The NHL wave probes give out free-surface elevation measurements by converting the immersion length on the galvanized metal tubes to voltage signals. The signals were then amplified and processed with low-pass filters and transmitted to the Data Acquisition Unit (DAQ), at a sampling rate of 200Hz. The wave probes were also statically calibrated before testing, by drawing a linear relation between the immersion length and the voltage output. Satisfactory linearity was discovered, with the correlation coefficient being 0.999998.

Table 4.3: Lateral locations of wave-cuts and corresponding effective length

Cut No.	y (m)	$\bar{y} = y/L$	L_{eff} (m)
1	0.399	0.262	4.637
2	0.495	0.325	4.095

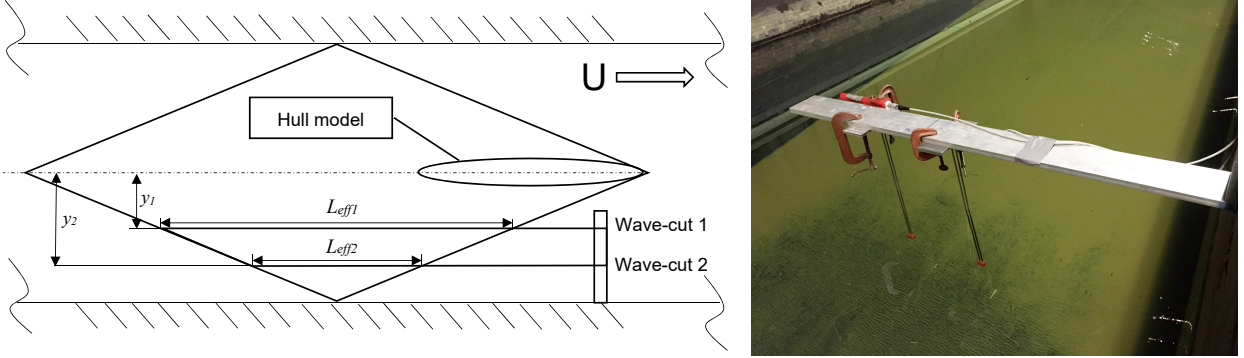


Figure 4.6: Schematic of the wave-cut measurement (left) and the wave probes set-up on the cantilevered boom.

4.4 Analysis of experimental uncertainties

To ensure the reliability of the experimental measurements, in accordance with the ITTC 2008 Guide in the Expression of Uncertainty in Experimental Hydrodynamics [78], the expanded uncertainties of measured total resistance and free-surface elevation in normalized forms were evaluated. The total uncertainty U_T is categorized as the sum of Type A and Type B uncertainties, where Type A uncertainty denotes standard deviations of at least ten measurements, and Type B uncertainty represents the bias of equipments and experimental procedures such as calibration of measuring devices. Each of the uncertainty component is assumed independent. The coverage factor k was taken as 2 for 95% confidence. Details regarding the procedures of uncertainty analysis are well documented in the literature [79, 80, 81], which will not be repeated in the present study.

Uncertainty in resistance measurements

The uncertainty in resistance measurements was evaluated by examining the measured coefficient of total resistance, defined as

$$C_T = \frac{R_t}{\frac{1}{2}\rho S_H U^2} \quad (4.9)$$

Because of the large number of tests needed for various speeds and hull configurations, it is impractical to evaluate the uncertainty of resistance measurements for each test case. Thus,

for the mono-hull tests, the uncertainty analysis was conducted for the case of $U = 1.183$ m/s ($F_n = 0.306$) because relatively large variance in measurements was discovered for this particular speed. Meanwhile, in order to have wider coverage of evaluated speeds, di-hull configurations with two moderate-to-high Froude numbers were chosen for the uncertainty evaluations. Twelve runs were repeated for each of the cases in the uncertainty tests.

In the meantime, the major sources of Type B uncertainty considered in the present study are:

- Errors caused by static calibration and least square curve fitting of load cell calibration data;
- Accuracy of load cell and data acquisition device;
- Error in towing carriage speed measurements, including the contribution from measuring the diameter of carriage wheel and the measurement of wheel angular velocity by a tachometer;
- Error introduced by load cell and model towing direction misalignment.

Water temperature was at 18 degree Celsius with a trivial amount of fluctuation during the entire testing period; therefore, water density ρ was assumed constant at 998.6 kg/m^3 . The errors caused by ship model offsets, surface roughness and appendages were all neglected. The total uncertainty for C_T are listed in Table 4.4:

Table 4.4: Total uncertainty of the measured resistances.

Hull configurations	F_n	Total uncertainty $U_T(\%)$	
Mono-hull	0.306	4.90	
Di-hull config. 1 ($\overline{st}, \overline{sp}$) = (0.5, 0.230)	0.352	Hull 1: 4.29	Hull 2: 3.38
Di-hull config. 2 ($\overline{st}, \overline{sp}$) = (0.5, 0.295)	0.398	Hull 1: 3.09	Hull 2: 2.37

The uncertainty levels of resistance measurements were discovered to be acceptable, especially for the cases with higher forward speeds. However, viscous effects such as the influences of boundary-layer separation, the change of form drag of each component hull in the di-hull systems [19] and the accuracy of the approximated frictional drag coefficient from the ITTC formula are not considered.

Uncertainty in wave-cut measurements

The uncertainty in the wave-elevation measurements was estimated by measuring and comparing the wave-cut data of twenty runs at $U = 1.183$ m/s ($F_n = 0.306$) in the mono-hull tests. The procedure of the present uncertainty analysis for wave-elevation measurements is similar to that introduced by Souto-Iglesias et al. [25].

For the evaluation of Type A uncertainty at different longitudinal locations along the wave-cut profile, around 3400 x -stations were evenly distributed between $x/L = -0.452$ and $x/L = 3.00$, with $x/L = 0$ and $x/L = 1.00$ representing stem and stern perpendiculars, respectively. Then, the standard deviations of wave-elevation measurements for every normalized x -station can be independently evaluated.

Meanwhile, the major sources of Type B uncertainty considered in the free-surface elevation measurements are

- Error caused by the calibration process of the least square fitting of the sampled data;
- The accuracy of wave probes and data acquisition devices;
- Errors of transverse locations of wave probes and the corresponding effective length L_{eff} , which is estimated to be around 5 mm. The error caused by the orientation of wave probes was discovered to have negligible effects on the experimental results and are thus ignored.

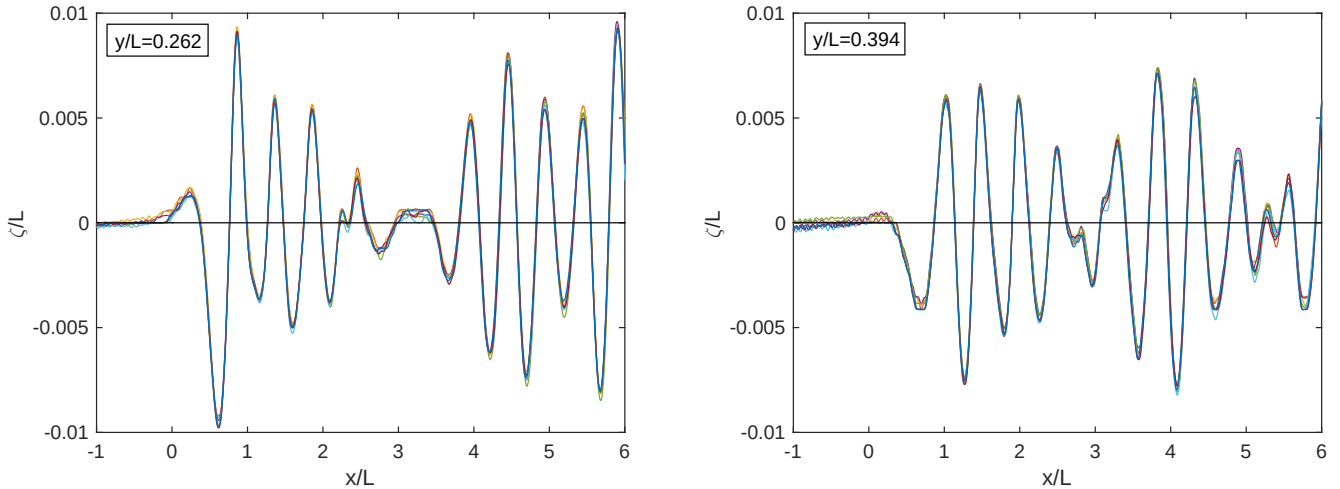


Figure 4.7: Plots of twenty wave-cut measurements taken at two different lateral locations with forward speed $U = 1.183$ m/s ($F_n = 0.306$)

The resulting wave-cut measurements at two different lateral locations of twenty overlapping runs are shown in Fig. 4.7. Very good repeatability is discovered, and the maximum total uncertainty of all x -station measurements was found to be below 5%.

4.5 Unsteady di-hull towing-tank tests of moderate-speed overtaking maneuvers

The unsteady interaction between vessels in moderate-speed overtaking maneuvers was also studied with towing-tank experiments, in which one of the hull model was set to overtake its stationary counterpart with constant forward speed.

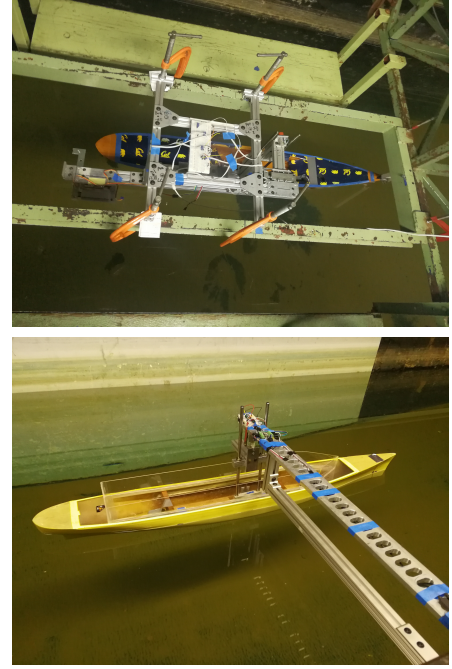
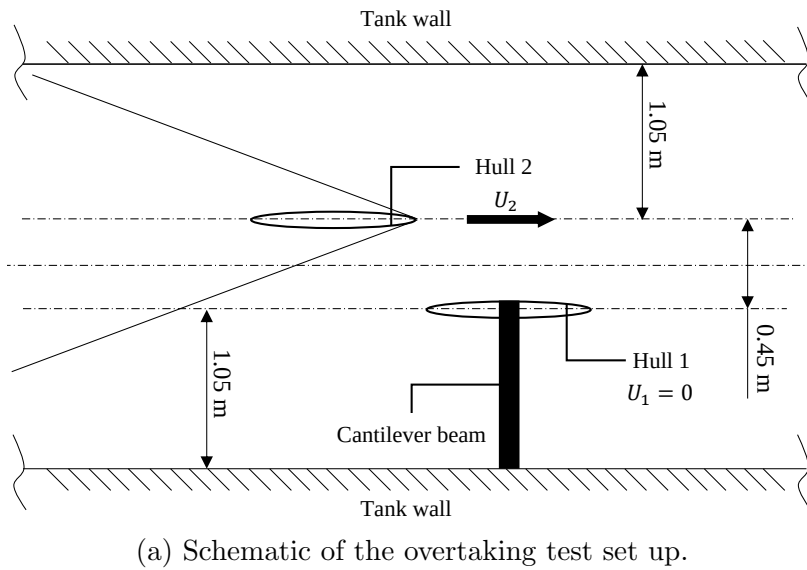


Figure 4.8: Set-up of the towing-tank experiment of moderate-speed overtaking maneuvers. The schematic of the experiment is shown in (a), while photos of the set-up of hulls and mounting structures are shown in (b).

The equipment used in the steady-state di-hull experiments was re-purposed for the unsteady tests. With slight modifications, the cantilevered arm used for holding wave probes was used to keep hull 1 at the middle of the towing tank. Meanwhile, hull 2 was still connected to the towing frame and driven by the carriage to overtake hull 1. A beam-type load cell was installed at the longitudinal center of gravity of hull 1 to measure the sway force at a frequency of 125 Hz, while another pair of load cells were mounted on the carriage to measure the drag and lateral force of the overtaking hull 2. The recorded signals on load cells were processed by a low-pass filter with 5 Hz cut-off frequency. Tested overtaking speeds for hull 2 were chosen between 1.01 and 1.24 m/s, and the corresponding Froude number $F_n = U/\sqrt{gL}$ are 0.260 and 0.321, respectively. The lateral distance between hulls, i.e. the

separation sp was set to be 0.45m, and both model hulls are placed with equal distance towards the centerline of the towing tank to minimize the influence from tank walls.

During the tests, hull 2 was first accelerated by the carriage until the designated overtaking speed was reached, then it went pass hull 1 with constant speed, while the load cells took measurements of the entire process. In order to synchronize the instantaneous stagger between hulls with force measurements, an HC-SR04 ultrasonic probe controlled by an Arduino Uno controller was also installed on the cantilevered arm aligned with the mid-ship section of the overtaken hull, while a light-weight foam bar was attached to the bow of hull 2. During the overtaking experiment, the ultrasonic probe gave out distance measurements at 30Hz as hull 2 overtook hull 1, and the blockage of foam bar would create a sudden change in distance measurement, which marks the moment of the bow of hull 2 passes the mid-ship section of hull 1. Therefore, with the known length of the hull models, the absolute time stamp of the moment when hull 2 went pass the midship of hull 1, and the carriage velocity, the entire time-history of measurements can be reconstructed, and the measurements of loads on both hull models can be synchronized.

Chapter 5

Results and Discussions on the Wave Resistances of Steady-state Di-hull Systems

In this chapter, the experimental and computational results on the wave resistances of steady-state di-hull systems are analyzed. The influences of varying stagger, separation and forward speed on the wave resistances of the di-hull systems are discussed. In addition, based on the wave-pattern analysis introduced in Chapter 2, a novel approach called the wave-pattern alignment method (WAM) is introduced to study the relation between the wave resistance of a di-hull system and the wave amplitude function of its component hulls.

5.1 Experimental results and the validations of MSPM computation

Wave resistance coefficient C_w , sinkage h and trim φ

The experimental results of C_w for the mono-hull tests and the di-hull tests of different hull configurations are plotted against their computational counterparts from the MSPM computation in Fig. 5.2(a) and 5.3, and satisfactory agreement is observed. In addition, the numerical and experimental results of the present study also align well with the published data from [54] and [20]. Meanwhile, the variations of wetted surface area brought by the changing sinkage and trim of each component hull are plotted against increasing Froude numbers in Fig. 5.1. It is also shown that for the tested di-hull configurations, hull 1 traveling at the rear of the formation experiences smaller wave resistance than its counterpart at the front, and the difference becomes more significant as forward speed increases.

The results for sinkage and trim for different hull configurations are also well captured by the MSPM computations, except that computational results mildly underestimate the sinkage while overestimate the trim angles, as shown in Fig. 5.2(b) and 5.4. The results in [82] acquired by finite element method is also provided for comparison. Both component hulls in the tested di-hull systems have similar sinkage. However, at a higher forward speed, the trim angle of each component hull diverges from each other, with hull 2 trimming by the stern like the isolated component hull and hull 1 trimming by the bow. Moreover, both experimental and computational results show larger differences in wave resistances and trim angles between the component hulls for smaller $\overline{sp}$ at higher Froude numbers, indicating that stronger wave interference may occur when the two component hulls are placed closer to each other.

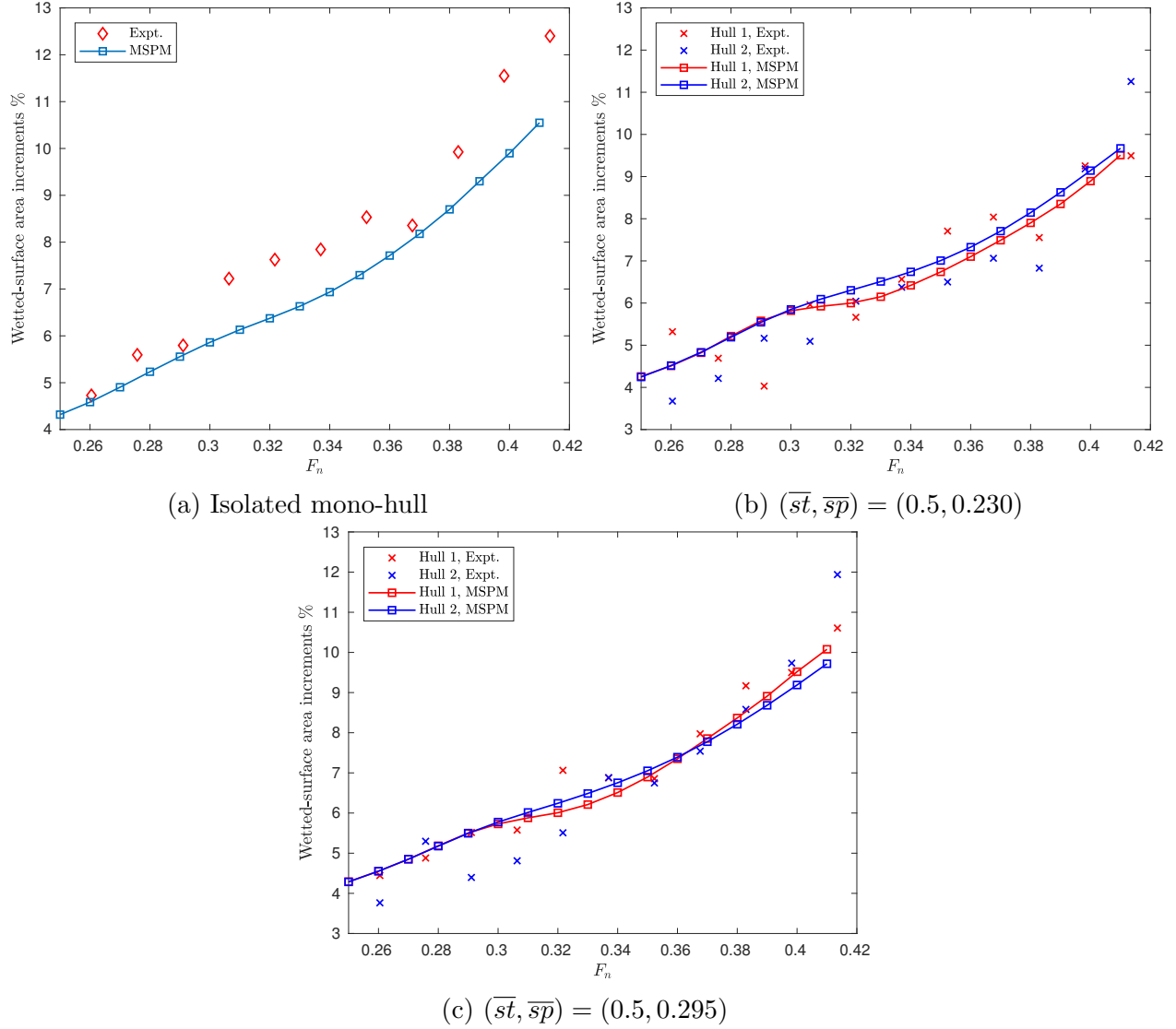


Figure 5.1: Corrections of wetted-surface area of each hull brought by changing sinkage and trim.

5.2 Results on α_1 , α_2 and α_d

Lastly, the results of α_1 , α_2 and α_d as defined by Eq. (2.20) in Chapter 2 are as shown in Fig. 5.6. The computational results of α_d given by the thin-ship computation CATRES as introduced in [11] are also shown in Fig. 5.6(e) and (f) for reference. Good agreement is in general achieved between the results of MSPM computation and the experimental measurements. Values of α_1 are found to be mostly negative over the range of the evaluated Froude number, while the opposite is true for α_2 . For the MSPM computation and the experimental measurements, negative values of α_d are observed for $F_n > 0.35$, with the most

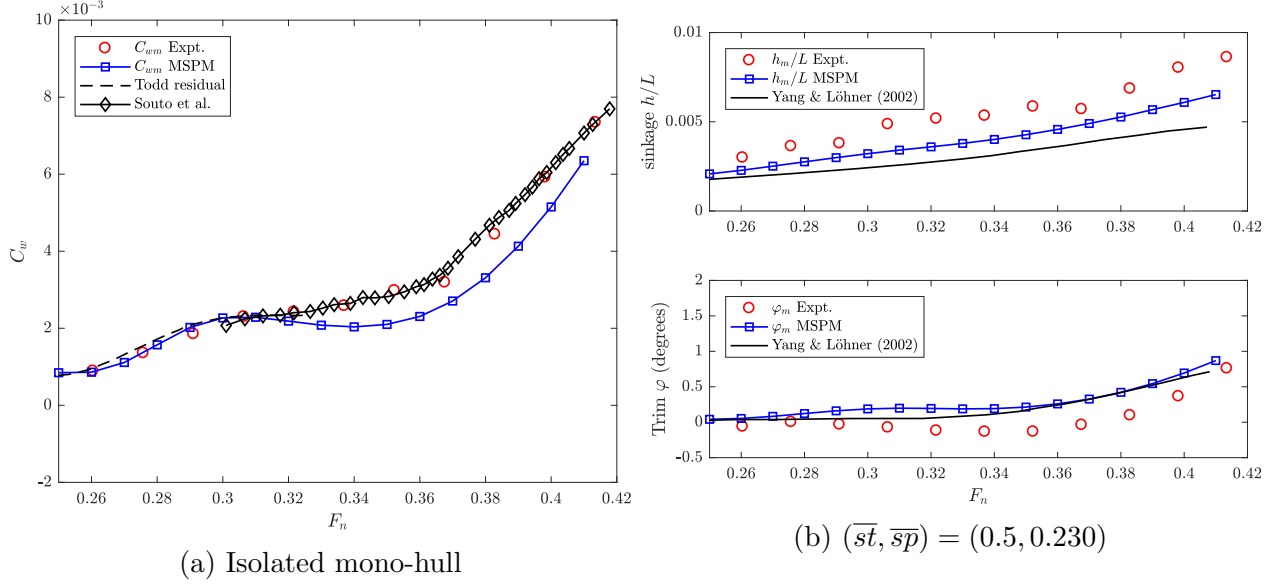


Figure 5.2: Wave resistance coefficient C_w , trim and sinkage of an isolated Series 60 hull model.

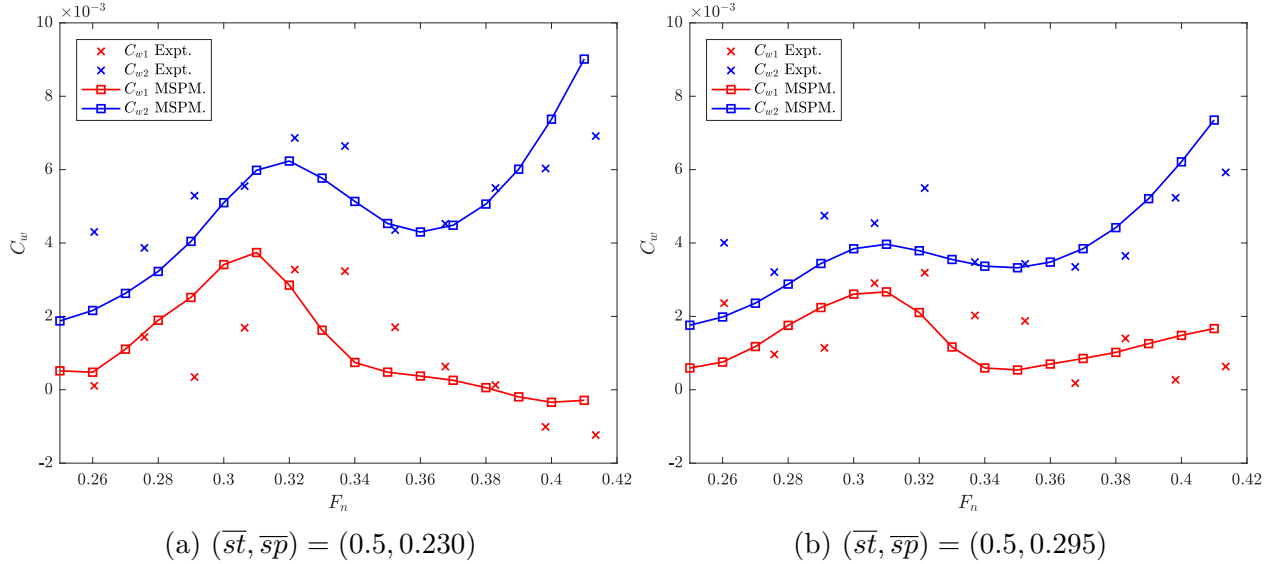


Figure 5.3: Wave resistance coefficient C_w of each hull in different di-hull configurations made by a pair of identical Series 60 hull models.

negative one around -0.3 for $F_n \approx 0.4$, which confirms the theoretical predictions made in the previous study [12] and represents a dramatic 60% total wave-resistance reduction.

It is discovered that for the evaluated hull configurations, most of the wave-resistance reduction takes place at hull 1 at the rear of the formation, which also accounts for the

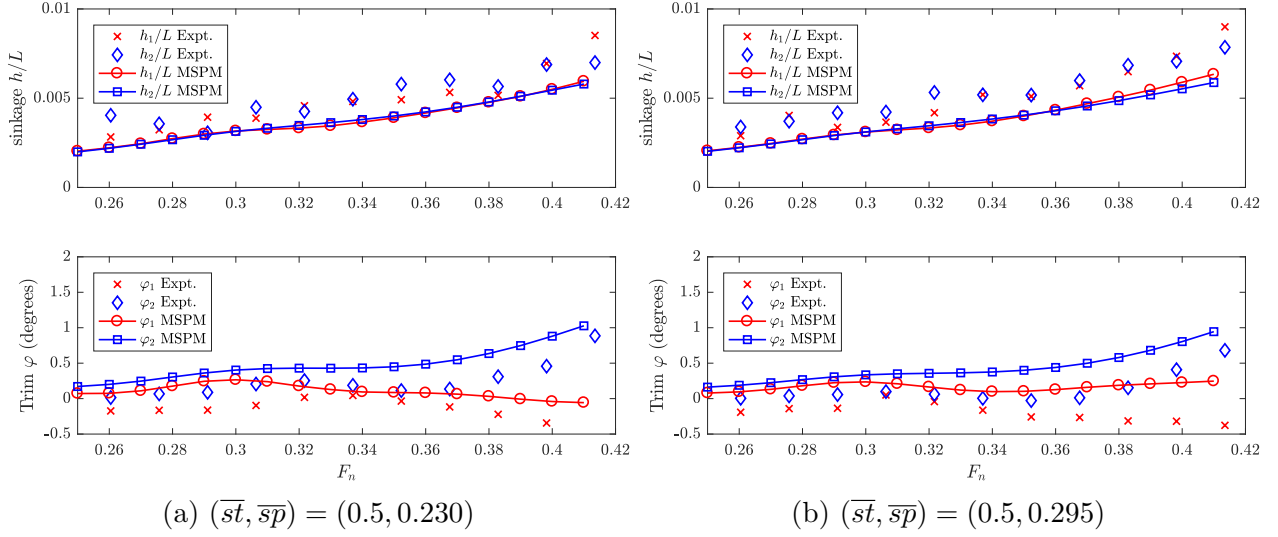


Figure 5.4: Sinkage and trim of both component hulls in di-hull systems with various configurations.

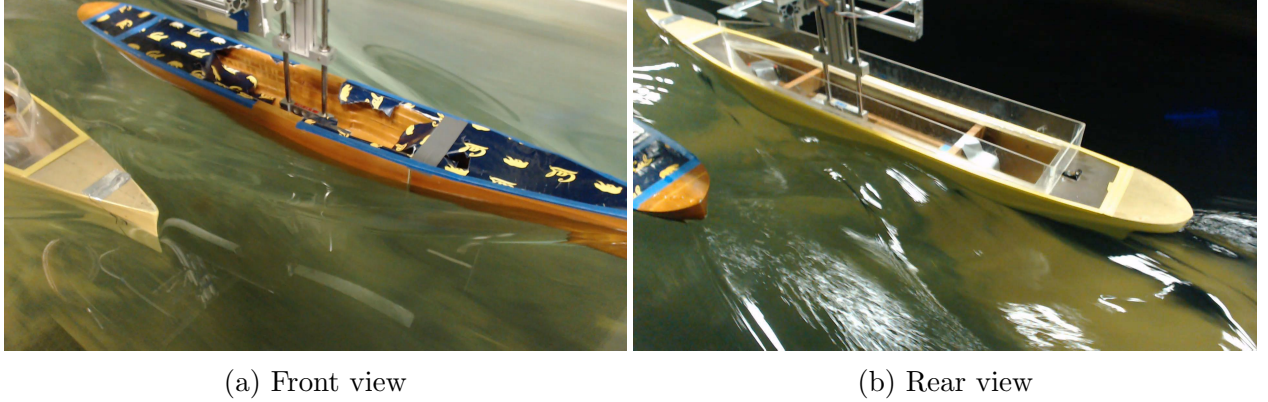


Figure 5.5: Photos of the towing-tank tests of hull configuration $(\overline{st}, \overline{sp}) = (0.5, 0.295)$ with $F_n = 0.306$.

most part of the total wave-resistance reduction of the di-hull system. In addition, hull 1 is in fact traveling within the wedge-like wave patterns (known as the “Kelvin Wedge” [52]) generated by hull 2 and reflected a large portion of its port-side divergent waves (Fig. 5.5(a)); meanwhile, similar blockage effect by hull 2 on the wave patterns of hull 1 was not as significant, since hull 2 travelled outside the divergent waves of hull 1 (Fig. 5.5(b)). Therefore, apart from the interference between the far-field ship waves generated by each hull, the wave patterns of hull 2 in the di-hull system were also interfering with the diffracted waves from hull 1. This additional interference may be responsible for the positive values of α_2 and α_d for Froude number ranges from 0.3 to 0.35 in the MSPM computation and the experimental measurements. One may find support for such claim in the less satisfactory prediction of the

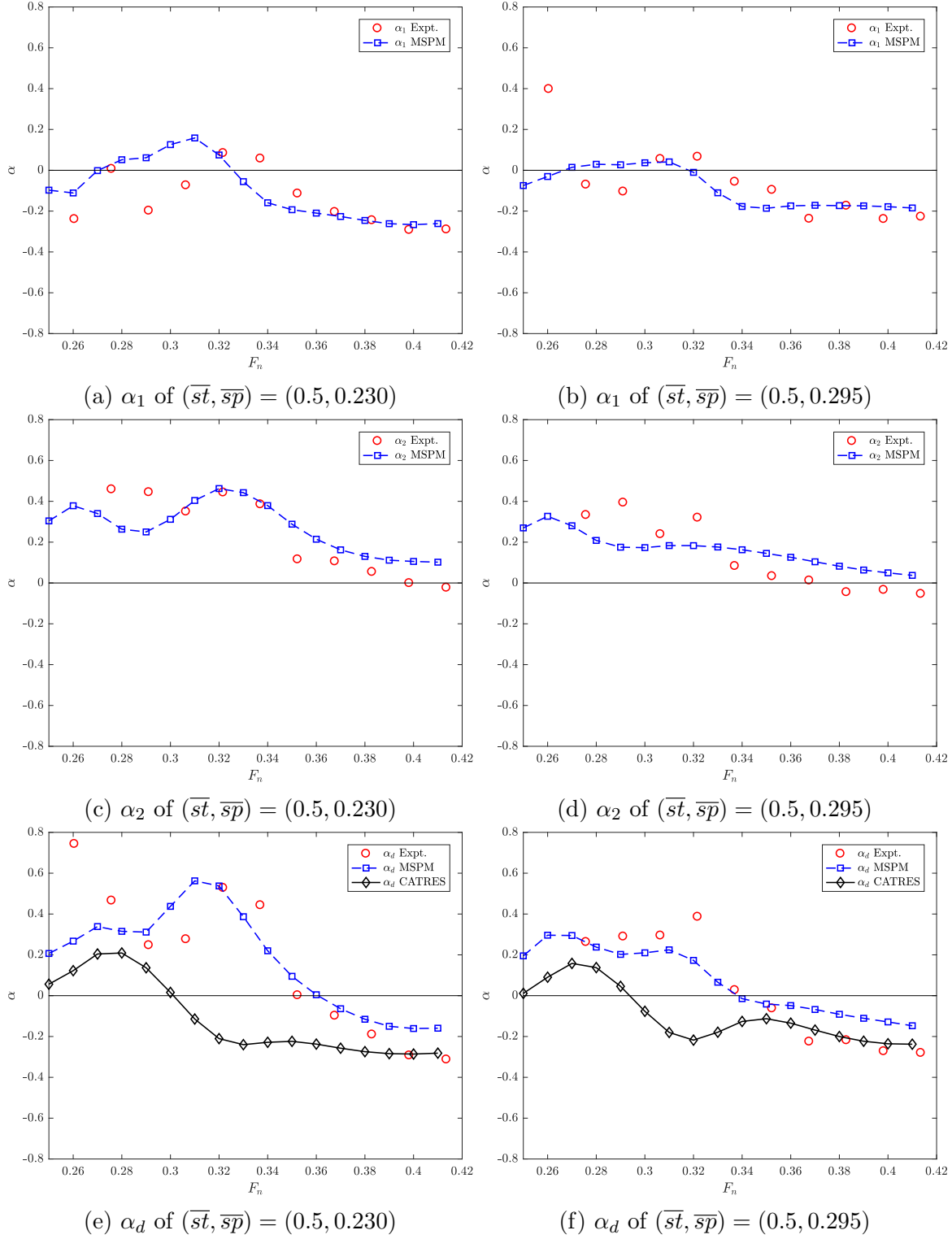


Figure 5.6: Interference parameter α given by di-hull towing-tank tests, in comparison with numerical results from MSPM and CATRES.

CATRES computation for the same Froude number range, since apart from the thin-ship approximation, the near-field interaction including the wave diffraction of hull bodies is also neglected in the CATRES computation in the direct superposition of far-field wave patterns of the isolated component hulls, that each of the component hull in fact does not “feel” the existence of its nearby neighbour [3]. In short, in addition to the far-field wave interference, the near-field interaction, especially the wave diffraction of hull bodies also has non-negligible influence over the wave resistance of a di-hull system with component hulls closely spaced.

5.3 Di-hull wave interference and the wave-angle alignment method (WAM)

It is shown in the far-field wave-pattern analysis introduced in Section 2.3 that with the appropriate hull configurations, negative interference resistance $R_{w_{1\rightleftharpoons 2}}$ may be achieved, and the interference of ship waves can reduced the total wave resistance of the di-hull system. Such an idea can be further pursued that a two-dimensional di-hull system with a stagger measuring one half wavelength of the waves it generates can enjoy zero total wave resistance, as the two sets of ship waves align peak-to-trough, resulting in the complete cancellation of ship waves [57]. Similar idea was also adopted in Wang & Yeung [83] for designing a two-dimensional pressure patch with zero wave resistance.

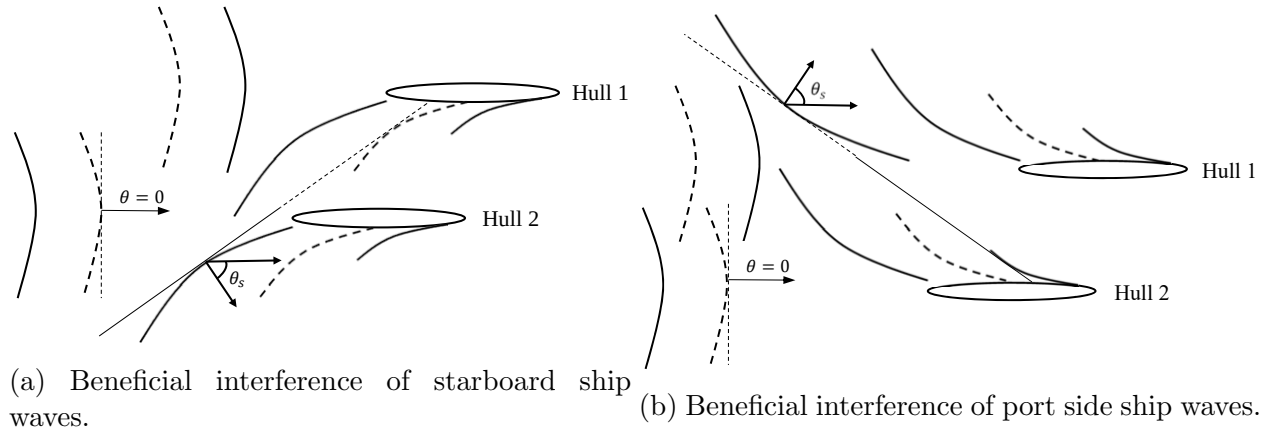


Figure 5.7: Schematics of three-dimensional wave interference. The peaks and troughs of the three-dimensional ship waves are denoted by — and — —, respectively. By overlapping wave patterns of isolated mono-hulls with proper hull spacing, the ship wave of θ_s are reduced for both starboard and portside.

However, while the half-wavelength phase shift for far-field wave-cancellation is still valid for the wave-resistance reduction of three-dimensional multi-hull systems [10, 12], its effectiveness may be substantially impaired by the fact that three-dimensional ship waves propagate with various wave angles $\theta \in (-\pi/2, \pi/2)$ as shown in Fig. 2.3; therefore, it is

obvious that the complete cancellation of ship waves in all directions is impossible for a three-dimensional multi-hull system.

Nevertheless, previous studies [13, 28, 27] have shown that far-field ship waves propagating in various directions contribute differently towards the wave resistance of a hull. Thus, significant wave-resistance reduction for a di-hull system may still be achieved by reducing the waves with the largest contributions to the wave resistance of its component hulls. In this regard, similar to the LCM introduced in Section 4.3, the transverse wave-cut method [26, 55](TWM) is applied on the MSPM-computed wave patterns of the isolated component hull, and the free-wave spectrum [28] derived from a transverse wave cut taken at a given x -position is given by

$$H(u; x) = 2[C(x, u) + i\frac{C_x(x, u)}{v}]e^{ik_0vx} \quad (5.1)$$

where

$$\begin{aligned} C(x, u) &= \int_{-\infty}^{\infty} \zeta(x, y)e^{iuy}du \\ C_x(x, u) &= \int_{-\infty}^{\infty} \zeta_x(x, y)e^{iuy}du \end{aligned} \quad (5.2)$$

and u and v are transverse and longitudinal wave numbers described in Eq. (4.6).

In the present study, the transverse wave cuts for deriving $H(u; x)$ of different Froude numbers are taken at a constant x position, which is located at one ship length downstream of the stern of the hull. Therefore, in the following illustration, with the dependency on x dropped, $H(u; x)$ is written as $H(u)$ for brevity. It is worth noting that $H(u)$ is directly associated with the refined amplitude functions $G(u)$ and $F(u)$ introduced in Section 4.3 by

$$H(u) = \frac{1}{k_0^2}(G(u) + iF(u)). \quad (5.3)$$

Therefore, LCM can also be utilized to derive the value of $H(u)$ with the computational wave-cut data. However, TWM is favored in the present computational investigation since it is relatively easier to implemented on the computational wave pattern by simply drawing a transverse cut at a given x location across the wedge-like wave pattern.

Eventually, similar to Eq. (4.7), the mono-hull wave resistance of each of the component hull is given by

$$R_{w_{mj}} = \frac{\rho U^2}{8\pi} \int_0^\infty |H_j(u)|^2 \frac{\sqrt{1+4u^2}}{1+\sqrt{1+4u^2}} du, \quad j = 1, 2 \quad (5.4)$$

which in fact associates the wave resistance of a mono-hull vessel and its free-wave spectral density $|H(u)|$. Thus, if one assumes that wave interference only happens at far-field, then by simply overlapping two sets of mono-hull wave patterns with the appropriate hull spacing, a di-hull system may be properly configured to create a half-wavelength phase difference for reducing waves in the direction of maximum $|H(u)|^2$ to achieve significant wave-resistance

reductions. The u and θ associated with maximum $|H(u)|$ are termed u_s and θ_s , respectively, and such a method is named the Wave-angle Alignment Method (WAM). By simple geometric analysis, the relation between $\overline{st}$ and $\overline{sp}$ of the di-hull system configured to reduce waves of a given θ_s angle is easily acquired as

$$\overline{st} \pm \overline{sp} \tan \theta_s = n\pi F_n^2 \cos \theta_s, \quad n = 1, 3, 5... \quad (5.5)$$

i.e. a set of straight lines with slopes and intercepts determined by Froude number and θ_s on the $(\overline{st}, \overline{sp})$ plane. Conversely, if n of Eq. (5.5) is even, constructive interference of waves with angle θ_s occurs, leading to significantly increased wave resistance. In the following illustrations, the hull configurations following Eq. (5.5) with n being odd and even integers are termed as “beneficial configurations” and “detrimental configurations,” respectively. In the special case of $\theta_s = 0^\circ$ when pure transverse ship waves contribute the most towards wave resistance, the change of separation value has no effects on the cancellation of pure transverse waves, and the beneficial configurations become

$$\overline{st} = \frac{n\pi}{k_0 L}, \quad n = 1, 3, 5... \quad (5.6)$$

while n being even integers will lead to detrimental configurations, and the conclusions made in Tuck et al. [10] for canceling pure transverse ship waves propagating with $\theta = 0^\circ$ is recovered by the present WAM analysis. For example, as shown in Fig. 5.7, by overlapping two sets of mono-hull wave patterns with proper selections of st , sp , the waves of θ_s can be reduced for both starboard and port side. Meanwhile, the change of separation has no influence over the interference of transverse ship waves traveling with $\theta = 0$.

As shown in Fig. 5.8(a), the values of $|H(u)|^2$ with $F_n = 0.34$ peak at $u_s = 0.892$ or $\theta_s = 33.89^\circ$, and the straight lines representing beneficial and detrimental configurations given by the WAM analysis of Eq. (5.5) are drawn on top of the contour plot of α_d versus $\overline{st}$ and $\overline{sp}$ provided by the MSPM computation in Fig. 5.8(b). The predictions of WAM analysis are found to generally align well with the MSPM-computed α_d in both signs and orientations, except for the configurations with small $\overline{sp}$ when the near-field interaction between component hulls gains more significance as discussed in the previous section. Along the straight-line predictions of WAM, two hull configurations A and B with extremely positive and negative α_d are marked, and the associated di-hull wave patterns are shown in Fig. 5.8(c) and (d) with the crest-lines of the waves of θ_s indicated by red lines. It is easily observed that the wave heights along $\theta_s = 33.89^\circ$ are greatly reduced in configuration B with significant di-hull total wave-resistance reduction, while the constructive interference of the waves of $|H(u)|_{max}$ seen with configuration A brings about exceptional increase in both wave elevations and wave resistances.

Similar observations can be made with a higher Froude number $F_n = 0.41$ from Fig. 5.9, where the pure transverse waves at $u_s = 0$ or $\theta_s = 0^\circ$ carry the highest free-wave spectral density. Once again, significant wave-resistance reduction is achieved by cancelling waves with the highest free-wave spectral density. Negative α_d values are discovered with $\overline{st} \approx 0.5$ by both MSPM computation and WAM analysis, and the interference of waves of θ_s brings

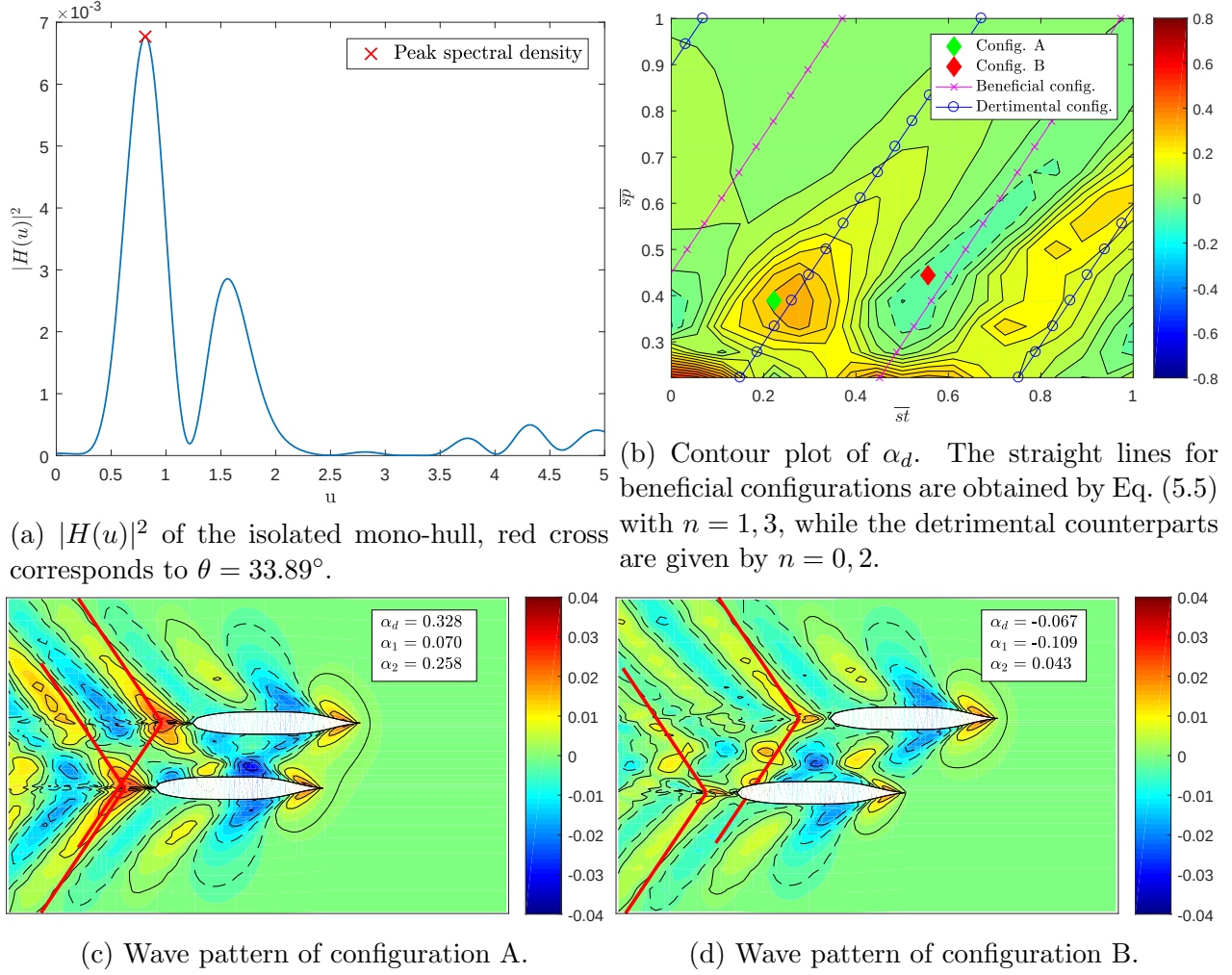


Figure 5.8: Mono-hull wave-angle spectrum and contour plot of α of $F_n = 0.34$. Red lines of (c) and (d) represent the crest-line of waves with $\theta = 33.89^\circ$.

about significant changes in the wave resistances of di-hull systems, as shown by Fig. 5.9(c) and (d). Further, since the change of separation values has little effect on the interference of waves at $\theta \approx 0^\circ$ [10], the contours of α_d stretch almost vertically with respect to the $\overline{st}$ axis, and the predictions of the WAM analysis align well with the contours as vertical lines. In addition, with higher Froude number, the agreement between the results of WAM analysis and MSPM computation improves even for cases with small $\overline{sp}$, probably due to the predominant far-field wave interference at high forward speeds [52, 84] which renders the near-field interaction relatively unimportant. Furthermore, the dramatic wave-resistance reduction of di-hull systems with $\overline{st} \approx 0.5$ at $F_n \approx 0.4$ is consistent with the discoveries of published studies [12, 3], which further substantiate the reliability of the present analysis.

In short, good agreement is generally achieved between the results of WAM analysis and

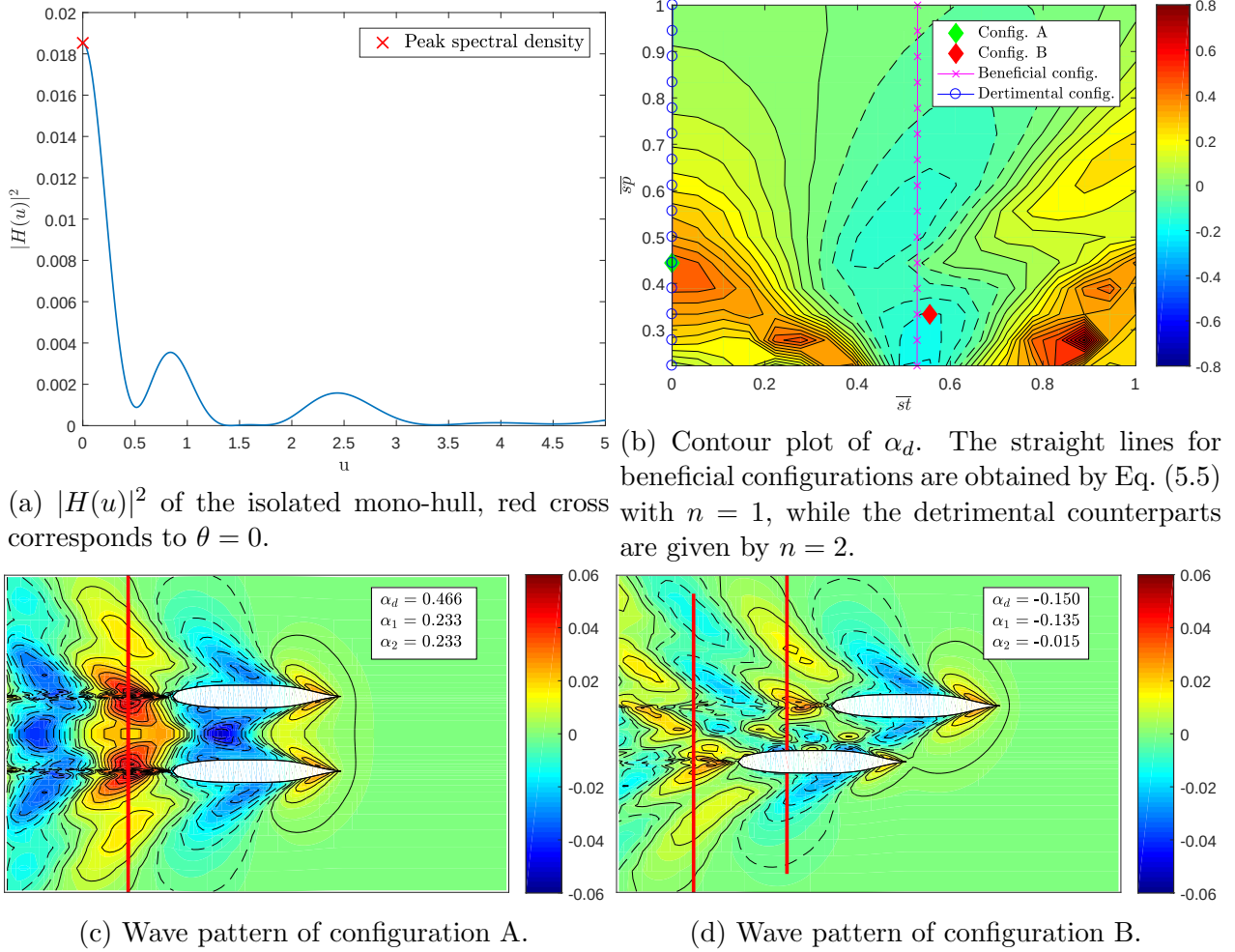


Figure 5.9: Mono-hull wave-angle spectrum and contour plot of α of $F_n = 0.41$. Red lines of (c) and (d) represent the crest-line of waves with $\theta = 0^\circ$.

the experimentally-validated MSPM computation, especially for the hull configurations with large hull spacing and higher forward speeds. Straight-line predictions of Eq. (5.5) not only go pass the hull configurations with extreme wave-resistance reduction or increase on the contour plots, but also align well with the contours of α_d for different signs and orientations. Therefore, a qualitative understanding of di-hull wave interference can be easily acquired with the knowledge of free-wave spectrum of each component hull and the simple geometric analysis of Eq. (5.5).

Meanwhile, just like its counterparts of wave-pattern analysis and hybridization method introduced in Chapter 4, the far-field analysis of WAM inevitably neglect the near-field contributions as introduced in Section 5.3, and the non-negligible influence of the near-field interaction manifests in the less satisfactory agreement between the results of WAM analysis and MSPM computation, when the two component hulls are placed closer to each other

and travel at a lower forward speeds. In particular, the wave pattern and consequently, the free-wave spectrum of a component hull in the di-hull system are different from those of an isolated mono-hull; meanwhile, reducing hull spacing may further increase such discrepancy by introducing stronger wave diffraction effect. Therefore, with the absence of near-field effects, the WAM analysis yields less satisfactory predictions when the two component hulls are closely spaced.

Chapter 6

Results and Discussions of the Moderate-speed Overtaking Maneuvers

In this chapter, the experimental and the computational results of an unsteady di-hull system made by two hulls during the moderate-speed overtaking maneuvers with $U_1 \neq U_2$ are discussed. Compared to the steady-state di-hull systems, the hydrodynamic interactions are further complicated by the relative motions between component hulls and the unsteadiness of the wave-making patterns. Therefore, the influences of varying overtaking speed, the speed difference between component hulls, and the associated free-surface effect on the loads on both component hulls in the overtaking process are of particular interests. In this regard, the hull-speed ratio γ which represents the speed difference between component hulls is defined as:

$$\gamma = \frac{F_{n1}}{F_{n2}}. \quad (6.1)$$

Therefore, $\gamma = 0$ represents hull 2 overtaking a stationary hull 1, and speed difference between hulls shrinks with increasing value of γ until $\gamma = 1$ is reached, when the two hulls move with the same forward speed and form a steady-state di-hull systems as the ones demonstrated in Chapter 4.

6.1 Overtaking maneuvers with zero γ

The plots for normalized longitudinal force, lateral force and yaw moment, denoted as C_{F_x} , C_{F_y} and C_{M_z} on the two hulls with different grid sizes and time intervals for two hulls are shown in Fig .6.1. h is the uniform grid length in the x_j directions for both patches, and the normalized time step is given as $\Delta\bar{t} = U_2\Delta t/h$, where Δt is the dimensional time step and h is a typical grid size on the free surface in x direction, respectively. Satisfactory convergence of results is observed for the computed loads on both hulls, and all of the computational results in the rest of the paper are computed with the scheme of $h = \frac{1}{36}L$ and $\bar{t} = 0.009$. Each free surface patch has 288 and 40 panels in x and y directions, respectively, and the surface of each hull is discretized into 1080 (36 in the longitudinal and 30 in the girth-wise direction) panels.

The normalized longitudinal forces on both hulls are plotted with respect to stagger in Fig. 6.2a for four different Froude numbers of hull 2 and $F_{n1} = 0$. Then, the maximum computed longitudinal forces are given against increasing F_{n2} in Fig. 6.2b. As hull 2 approaches, hull 1 begins to experience negative longitudinal forces in the opposite directions of U_2 , before switching sign at near-zero stagger. For hull 2, the experimentally measured

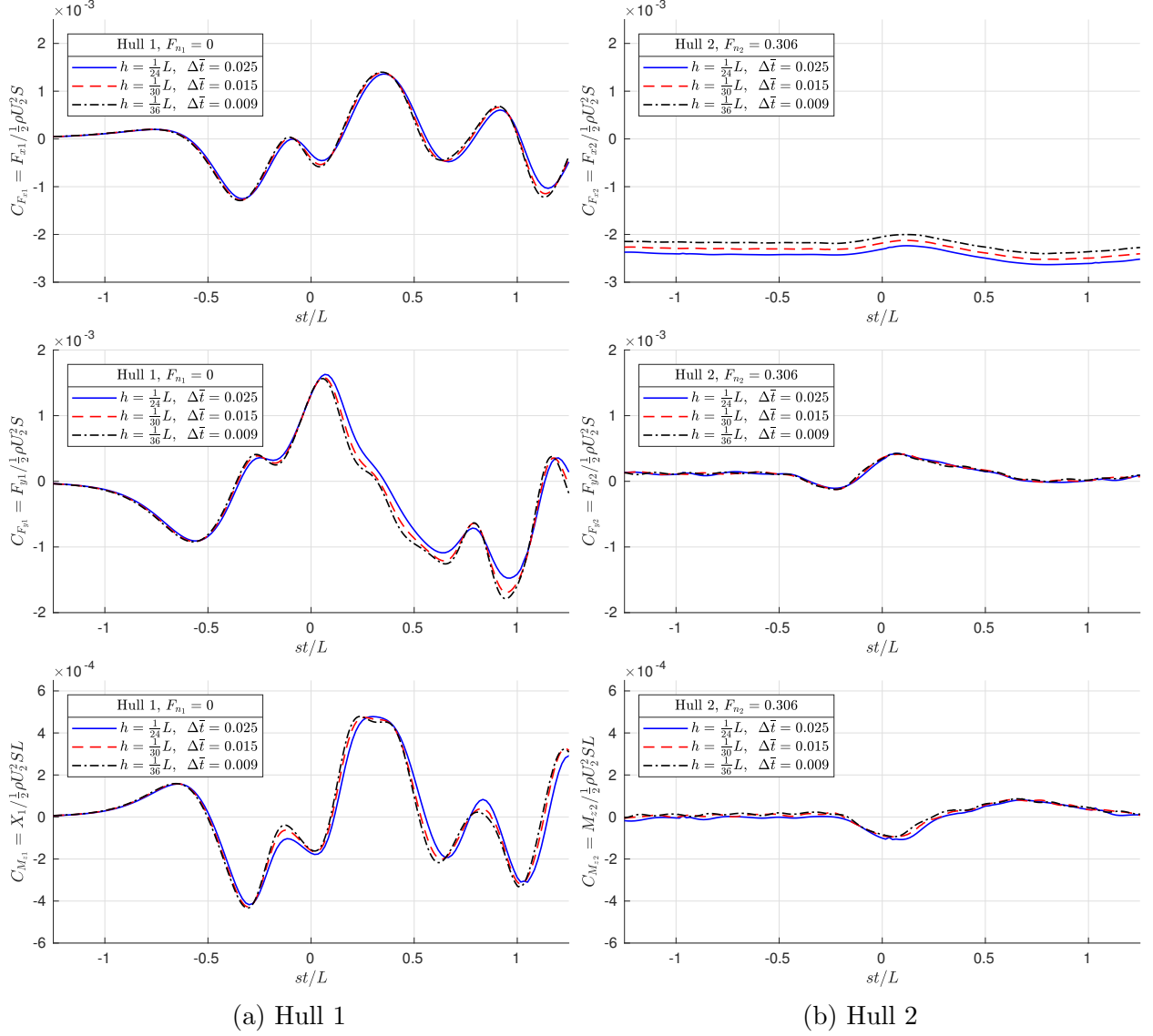
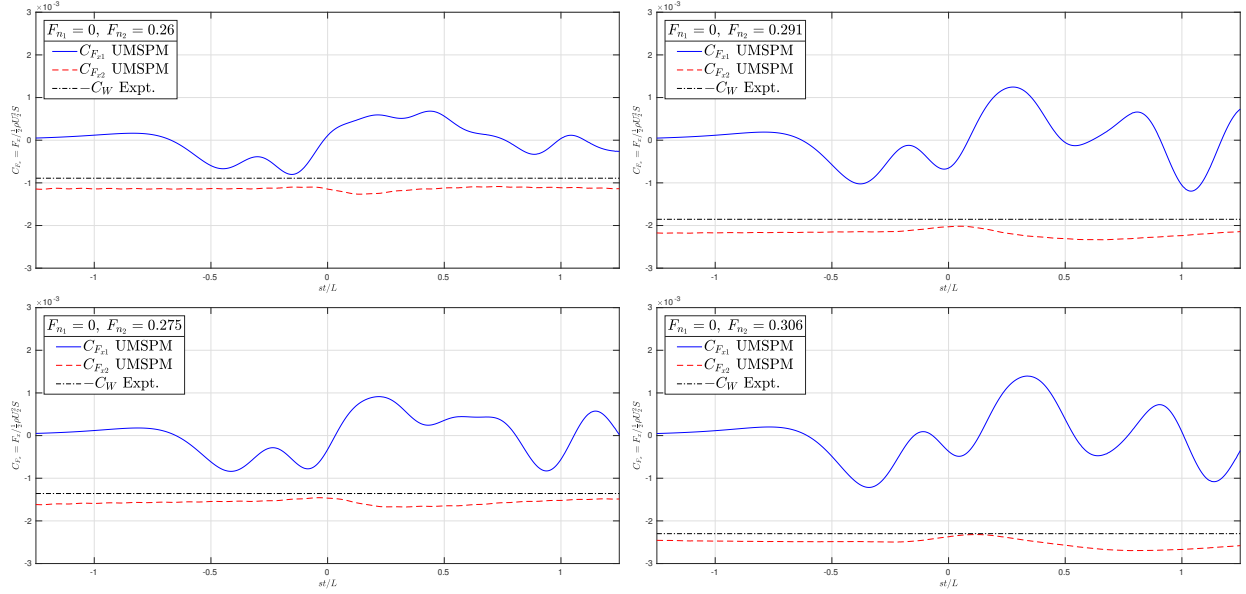


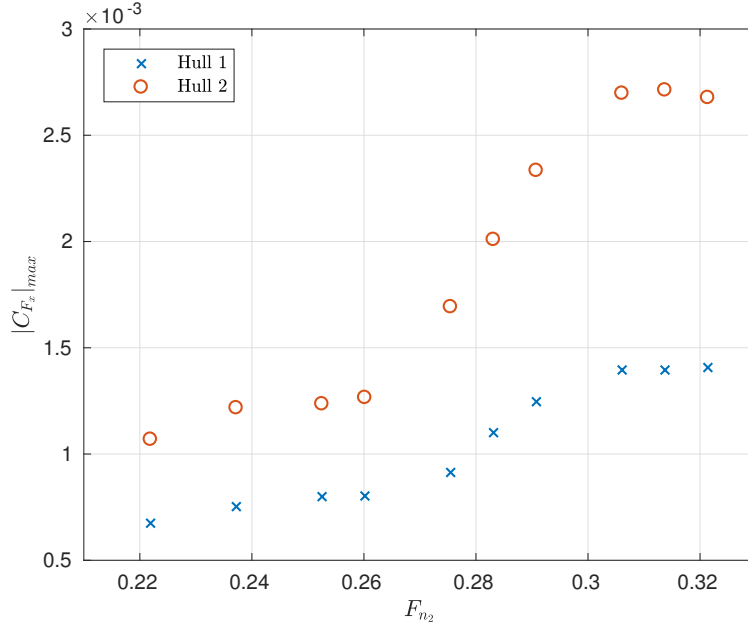
Figure 6.1: Computational results for hull 2 passing stationary hull 1 at $F_{n2} = 0.306$ of different grid sizes and time step.

unsteady longitudinal forces shows almost negligible variations from the steady-state wave resistance of the same model as reported in Yu et al., [22] that the differences are too small to be captured with sufficient confidence. [80] Therefore, the experimental results of wave resistance coefficient C_W are shown instead for reference. The computed unsteady longitudinal forces of hull 2 also show negligible variations about a constant value, which is very similar with the experimentally-measured wave resistance mentioned above. Meanwhile, as the overtaking hull experiences increasingly substantial wave resistances with higher forward speeds, the longitudinal forces on hull 2 increase more significantly with respect to speed

than those of hull 1, especially for $F_{n2} > 0.26$.



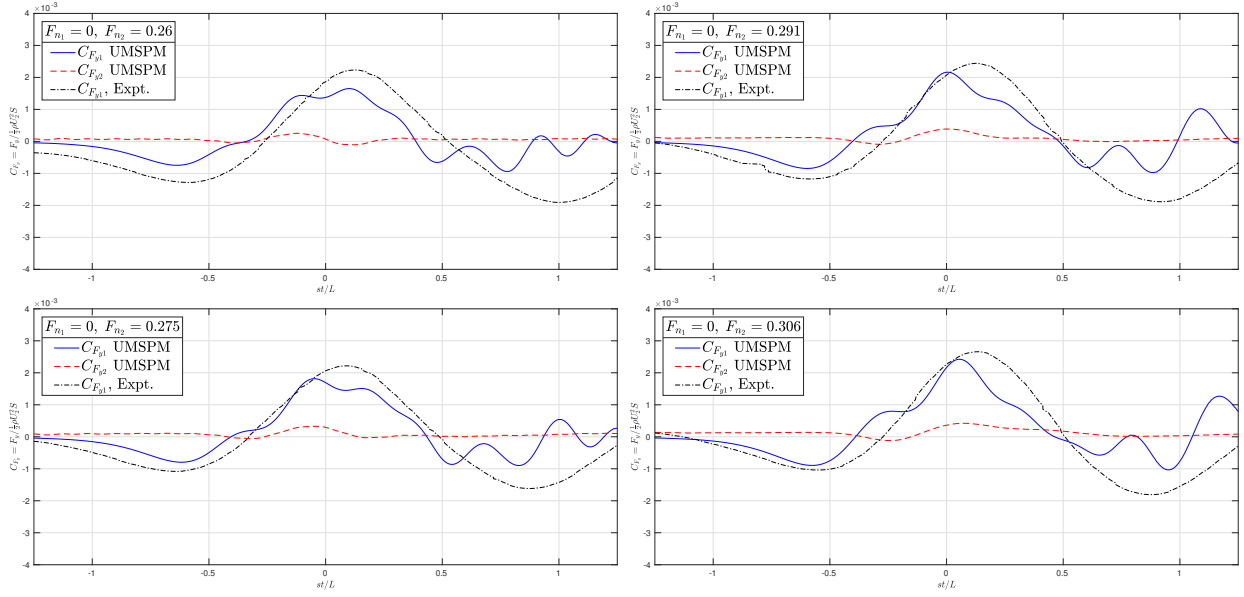
(a) Computational and experimental results of C_{F_x} versus st/L of selected F_{n2}



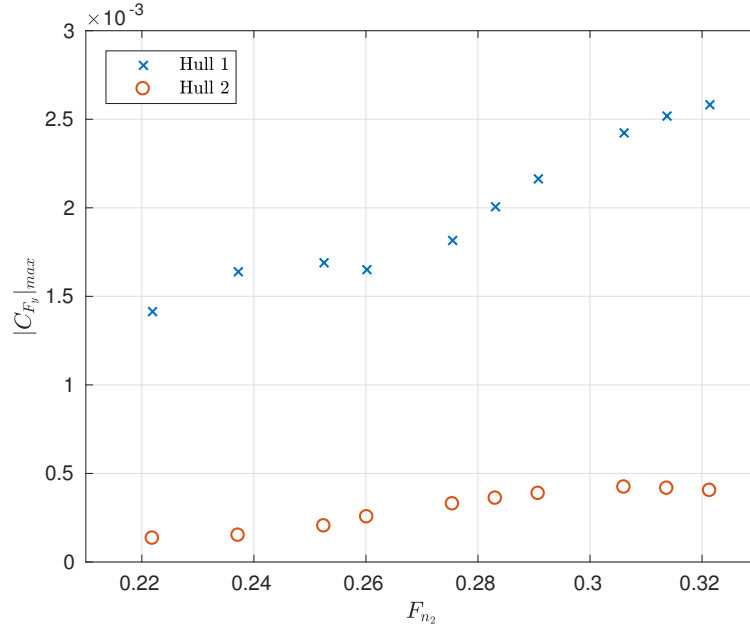
(b) Computational results of $|C_{F_x}|_{max}$ versus F_{n2}

Figure 6.2: Normalized longitudinal forces plotted against normalized stagger (left) and the Froude number of hull 2 (right) at $\gamma = 0$.

A similar set of plots for lateral forces are shown in Fig. 6.3a and 6.3b. The computational results of $C_{F_{y1}}$ show satisfactory agreement with the experimental measurements, especially



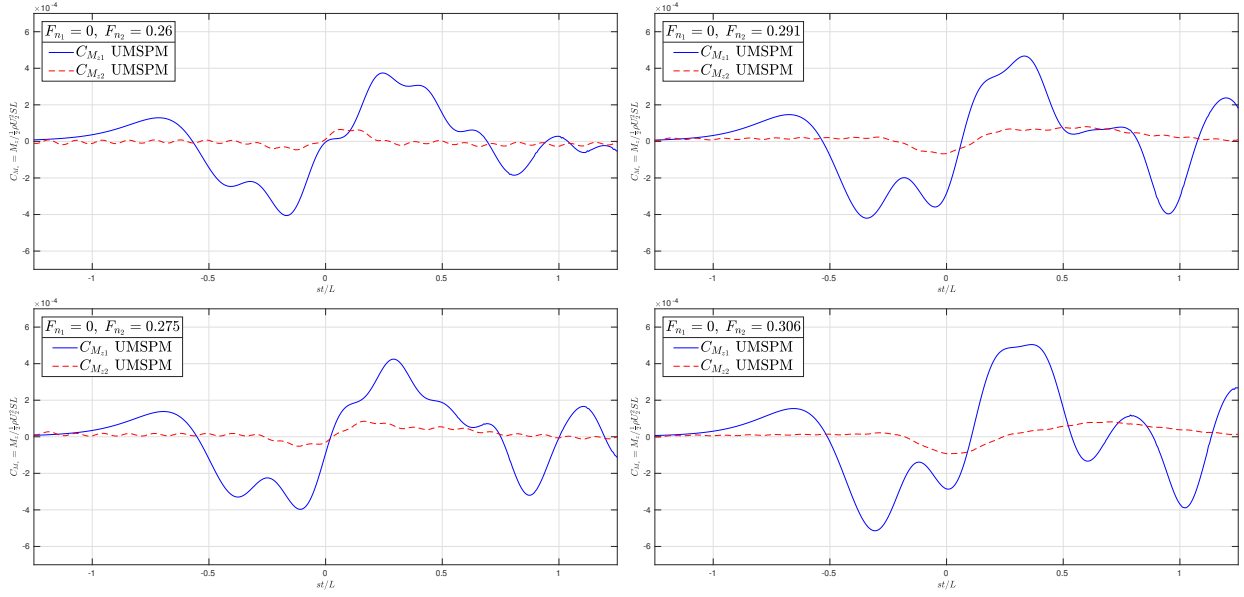
(a) Computational and experimental results of C_{F_y} versus st/L of selected F_{n_2}



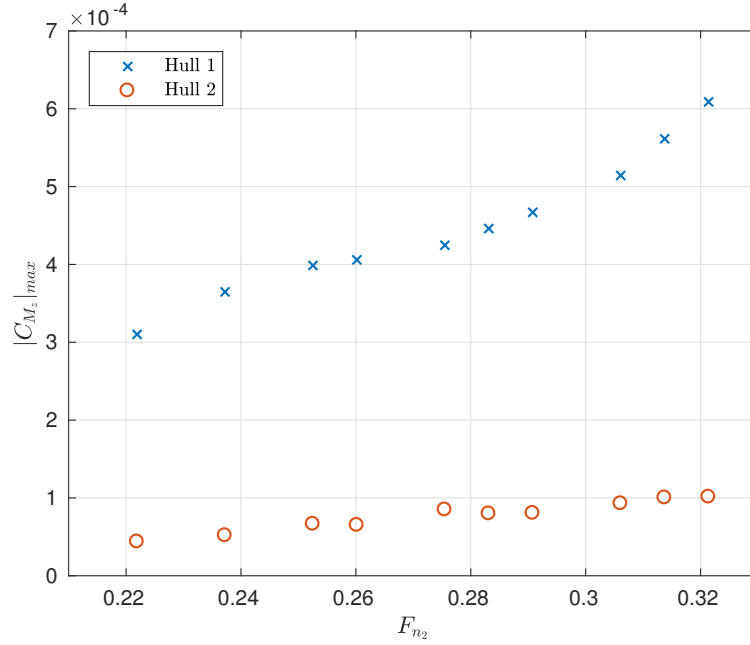
(b) Computational results of $|C_{F_y}|_{max}$ versus F_{n_2}

Figure 6.3: Normalized lateral forces plotted against normalized instantaneous stagger (a) and the Froude number of hull 2 (b) at $\gamma = 0$.

for the predictions of peak positive lateral forces at near-zero stagger. As hull 2 approaches, repulsive force acts on hull 1 first, which turns attractive later with $st/L \approx -0.25$. The



(a) C_{M_z} versus st/L of selected F_{n_2}



(b) Computational results of $|C_{M_z}|_{max}$ versus F_{n_2}

Figure 6.4: Normalized yaw moments plotted against normalized stagger (left) and the Froude number of hull 2 (right) at $\gamma = 0$.

experimental and computational results start to diverge from each other with $st/L > 0.5$, when negative lateral forces are observed in the experiments but are underestimated by the computations. Meanwhile, the computed results of $C_{F_{y_2}}$ of the overtaking hull still exhibit

only minor variations around zero in all tested cases.

The corresponding plots for instantaneous yaw moments are shown in Fig. 6.4. It is discovered that small positive bow-in moments (the bow of a hull has the tendency of turning towards another hull) act on hull 1 first as hull 2 approaches, followed by more significant negative bow-out moments with $st/L > -0.6$. Then, the values of $C_{M_{z1}}$ turn positive again near $st/L = 0$ and reach peak values around $st/L = 0.25$. Meanwhile, the yaw moment of hull 2 shows minor deviation from zero.

The results of zero γ overtaking show that the loads in the lateral direction (lateral force and yaw moment) of the stationary overtaken hull are much greater than those on its identical overtaking counterpart in both magnitudes and variations. Besides, as the overtaking speed increases, hull 1 will experience loads with larger magnitudes, especially for lateral force and yaw moment. Meanwhile, hull 2 does not seem to be much affected by the stationary hull 1 and behaves like a steady-state mono-hull moving in calm water, with almost constant longitudinal forces and near-zero lateral force and yaw moment. Furthermore, the simple potential-flow computation has been shown to provide satisfactory results of hydrodynamic loads on both hulls during the overtaking process with acceptable accuracy.

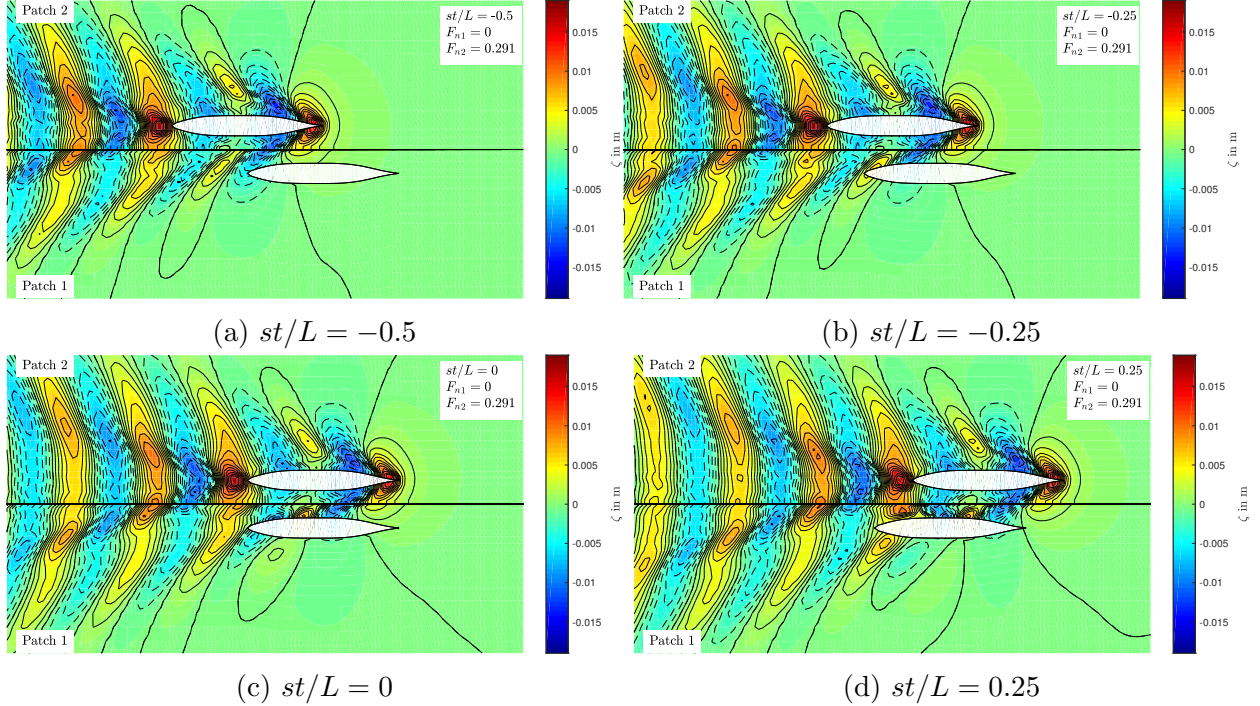


Figure 6.5: Contour plots of instantaneous wave patterns during overtaking process with $F_{n1} = 0$ and $F_{n2} = 0.291$.

The computed instantaneous wave patterns with $\gamma = 0$ and $F_{n2} = 0.291$ at four different values of stagger are shown in Fig. 6.5. It is shown that hull 1 effectively blocks the divergent waves [14, 57] of hull 2 during the overtaking process, that the starboard free surface of hull

1 is less disturbed than that of the portside as hull 2 passes by. Meanwhile, the transverse ship waves which propagate with near-zero wave angle [52] are less obstructed.

It is also noted that the computational results exhibit more short-period variations than the experimental measurements, and such observation is especially pronounced with $st/L > 0.5$, when hull 1 is becoming more exposed to the waves generated by the passing hull 2. Two major factors may contribute to such inconsistency between experimental and computational results. First of all, the absence of nonlinear terms, especially the missing refraction effect among waves traveling with various directions in the linearized free-surface modeling usually results in noticeable cumulative phase error in computing downstream ship waves [40], which eventually evolves into underestimated periods for ship wave and wave-induced loads [55, 85]. Secondly, the cantilever beam and the connecting mechanism for holding hull 1 in the experiments are inevitably flexible, and the overtaken hull is observed to have small pitch and heave motions in the experiment. Meanwhile, in the overtaking computation of $\gamma = 0$, hull 1 is completely stationary. Therefore, the non-zero small body accelerations of hull 1 will result in slight differences between the measurements and the actual external forces, which contributes to the discrepancies between experimental and computational results.

6.2 Overtaking maneuvers with distinct non-zero γ

To investigate the effect of changing γ , computations of overtaking maneuvers are carried out with F_{n_2} kept at 0.3, while the value of F_{n_1} changes between 0 and 0.3.

The normalized longitudinal forces of two hulls versus increasing instantaneous stagger of various γ are shown in Fig. 6.6a and 6.6b. It is shown that the computed $C_{F_{x1}}$ becomes generally more negative with larger values of F_{n_1} and γ . Meanwhile, the longitudinal forces of hull 2 in overtaking a moving hull 1 exhibit much more variations than the cases with $\gamma = 0$, when hull 1 is stationary.

Fig. 6.6c and 6.6d shows the normalized lateral forces for both hulls of various γ . The magnitudes of lateral forces of hull 1 generally decrease with larger γ , especially at near-zero stagger. Meanwhile, the normalized lateral forces of hull 2 of non-zero γ deviate considerably from that of $\gamma = 0$ with more variations.

The computational results of normalized yaw moment C_{M_z} of different values of γ are depicted in Fig. 6.6e and 6.6f. The yaw moments of hull 1 of different values of γ show insignificant differences, while those of hull 2 demonstrate larger deviations from zero with increasing values of γ .

The computed loads at $st/L = 0$ with various values of γ are shown on Fig. 6.7. When $\gamma = 1$ and $st/L = 0$, two hulls in fact constitute a symmetric catamaran [20] made by identical demi-hulls moving with constant forward speeds. Therefore, the loads at this moment are independent of time, and the longitudinal forces of both hulls are identical and equivalent to the calm-water wave-making resistance [10] of the demi-hull, while the lateral forces and yaw moments of both hulls are of same magnitudes but opposite directions. Meanwhile, the lateral force of hull 1 reduces in magnitudes with larger values of γ , while

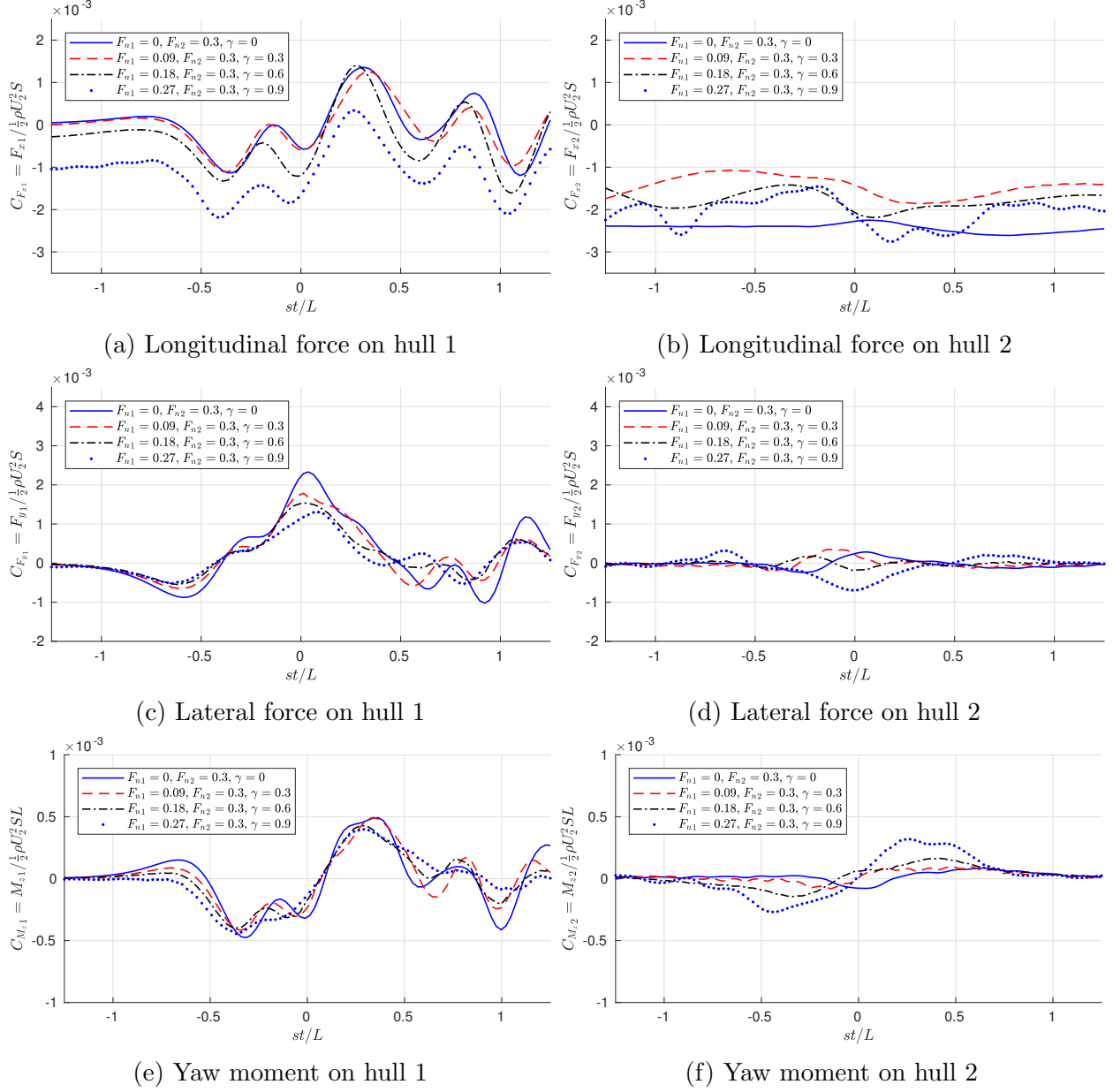


Figure 6.6: Computed normalized longitudinal, lateral forces and yaw moments of the hull 1 (left) and hull 2 (right) at different values of γ .

its hull 2 counterpart becomes more negative as γ increases, and both results approach to the corresponding steady-state lateral forces at higher values of γ . On the other hand, the yaw moment of both hulls at $st/L = 0$ are generally small and do not change significantly with different values of γ . In general, with reducing speed differences between hulls, it is observed that the unsteady loads of both hulls at an instantaneous stagger becomes more

and more similar to those of a steady-state di-hull system of the same hull spacing and forward speed [11].

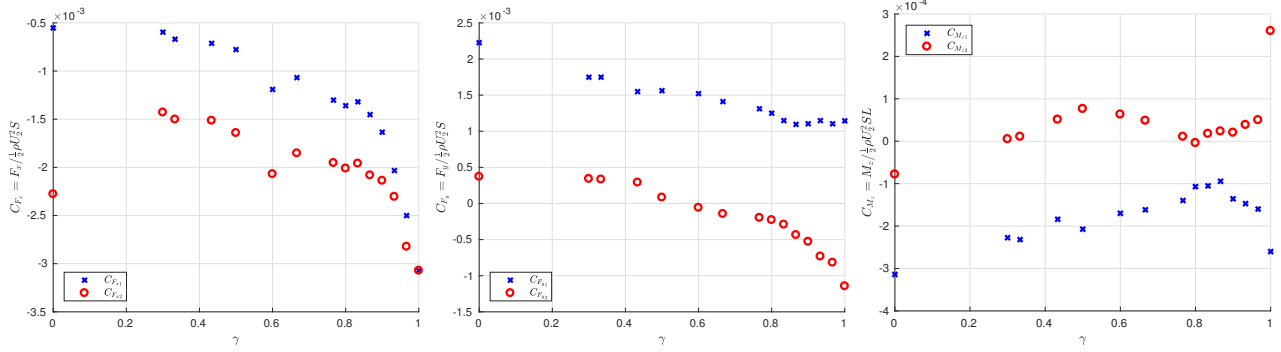


Figure 6.7: Normalized longitudinal, lateral forces and yaw moment taken at $st/L = 0$ for different values of γ .

The computed instantaneous wave patterns of the overtaking process with $\gamma = 0.3, 0.6$ and 0.9 at various values of instantaneous stagger are shown on Fig. 6.8. With smaller γ as shown in Fig. 6.8a, hull 1 moves at much slower speeds than the overtaking hull, and generates insignificant waves; therefore, its wave patterns are almost imperceivable in comparison with those of hull 2. As the value of γ approaches one, two hulls generate waves with comparable amplitudes and wavelengths, therefore more significant wave interference may be achieved which may considerably change the wave elevations of the flow field [11, 10, 22]. For example, as shown in Fig. 6.8c, the transverse ship waves of each hull are of similar wavelengths and heights, and significant increase of wave elevations is observed in-between hulls and behind sterns at near-zero stagger where two sets of ship waves align peak-to-peak. However, regardless of the value of γ , hull 1 still effectively block the divergent waves of hull 2 in a similar manner as in the cases of $\gamma = 0$ discussed in Section III B, and the free surface on the starboard of hull 1 is less disturbed by the influence of hull 2 during the overtaking process.

6.3 Analysis of the free-surface effect

Compared with the studies of slow-speed ship-to-ship interactions [86, 38, 87] in which the free-surface effect is ignored, above observations of moderate-speed overtaking indicate strong influence from the deformed free surface. In particular, wave-making of the moving hulls and the interference of ship waves are discovered to be the two major aspects of the free-surface effect, which substantially affect the loads on each hull in the overtaking process.

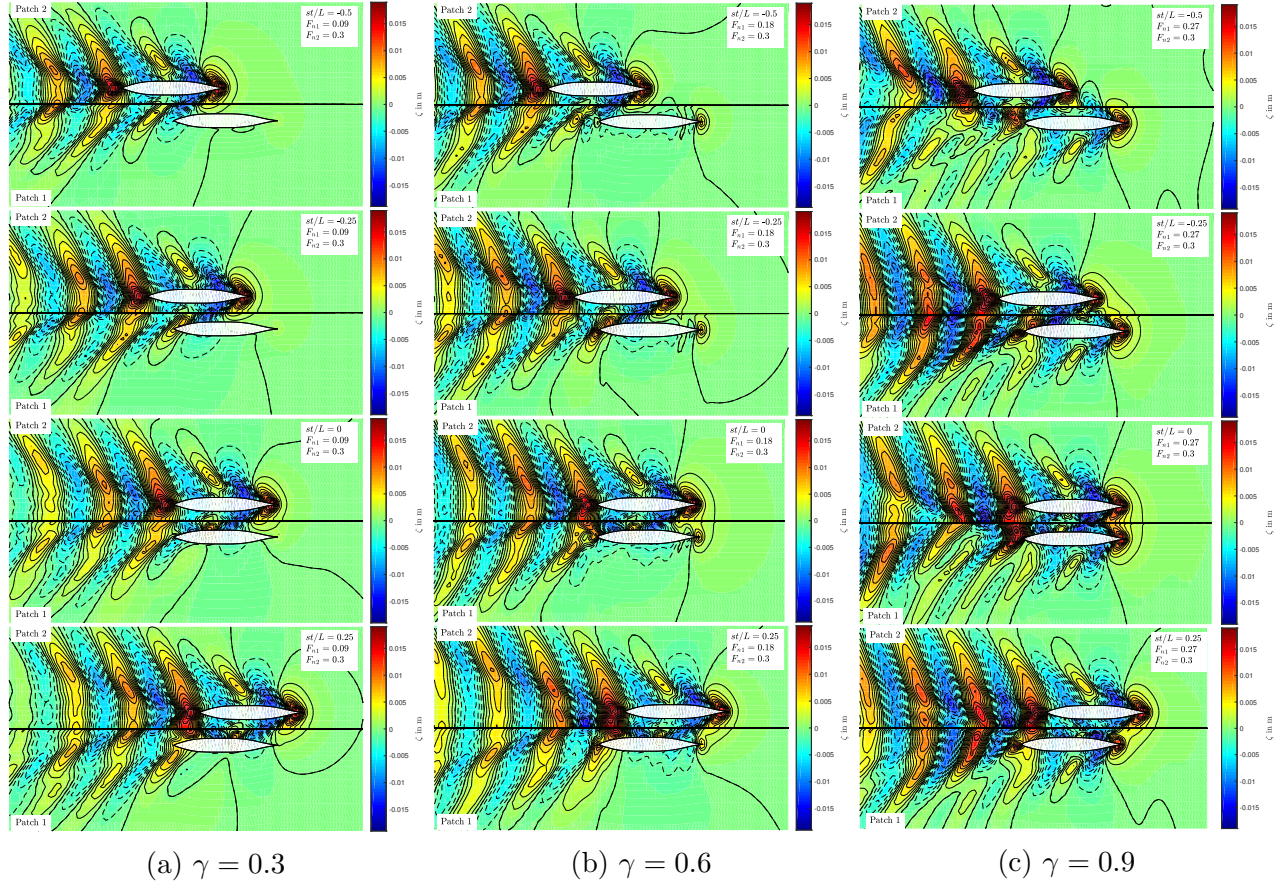


Figure 6.8: Contour plots of instantaneous wave patterns during overtaking process of various values of γ and $st/L = -0.5, -0.25, 0$ and 0.25 from top to bottom, respectively.

Wave-making of hulls

Wave-making resistance becomes more significant for ships traveling at higher forward speeds. Therefore, if the longitudinal force is considered as the instantaneous wave resistance of a moving hull, it is not hard to further deduce that the wave-making of each moving hull also accounts for a considerable portion of its longitudinal force in the moderate-speed overtaking maneuver. Such claim can be easily substantiated by the discovery that the longitudinal force on hull 2 in the process of overtaking a stationary hull is almost identical with its wave resistance in isolation. In the meantime, the loads on the stationary hull 1 are also influenced by the uneven hydrodynamic pressure distribution associated with the deformed free surface. For instance, as shown on Fig. 6.3 and 6.5, compared to the relatively calm starboard free surface of hull 1 caused by the blockage effect of hull body, positive pressure of portside wave peak (Fig. 6.5a) and the negative pressure of portside wave trough (Fig. 6.5b and 6.5c) are found to be associated with negative and positive lateral forces on the overtaken hull, respectively.

Interference of ship waves

In the overtaking maneuvers of non-zero γ , the interference between two sets of ship waves becomes another important aspect of the free-surface effect. In a steady-state di-hull system, the in-phase (peak-to-peak alignment of waves) and out-of-phase (peak-trough alignment of waves) interference between two sets of ship waves can significantly increase or decrease the free-surface elevations and wave resistances of the two hulls. Similarly, in the unsteady interactions between two hulls in the overtaking maneuvers, the phase difference between two sets of ship waves varies with respect to instantaneous stagger, and the resulting interference substantially changes the free-surface elevation of the fluid domain and the loads on both hulls in a fluctuating manner. For instance, as shown on Fig. 6.6a, as a result of traveling in the wave field generated by the overtaken hull, longitudinal force of hull 2 exhibits much greater variations in the overtaking maneuvers of non-zero γ .

In addition, in the overtaking maneuvers with larger γ as shown on Fig. 6.8c, a deep wave trough is formed in between two hulls due to in-phase wave interference at near-zero instantaneous stagger. For hull 1, since its starboard free surface is also already considerably disturbed by its own wave-making, the uneven pressure distribution across both sides of the hull as in the case of zero γ overtaking is alleviated, which results in reduced positive peak lateral forces as shown on Fig. 6.6b. On the other hand, wave pattern of the hull 2 is made less symmetric due to the existence of large, deep wave trough on its starboard, and the induced negative pressure zone causes the overtaking hull to sway in the negative direction.

Interestingly, above wave interference effect on loads is discovered to be more significant with $\gamma > 0.8$, since the interference between ship waves is stronger with two sets of waves of comparable magnitudes and frequencies. The longitudinal forces of both hulls converge more rapidly towards the demi-hull wave resistance of a steady-state catamaran. Meanwhile, the variance of lateral force of hull 1 comes to a stasis with $\gamma > 0.8$, while that of hull 2 becomes more negative a faster rate. Conversely, in the overtaking maneuvers with smaller γ , the insubstantial waves generated by the slow-moving overtaken hull 1 exert insignificant influence on the wave pattern and loads of the overtaking hull, in such context hull 2 behaves more similarly as a mono-hull traveling in isolation.

Chapter 7

Optimization of Trimaran Configurations with Deep Neural Network (DNN)

The design and optimization of multi-hull systems usually require exhausting a large quantity of multi-hull configurations, which makes the direct computational and experimental approaches too expensive. However, an efficient solution is provided with the advent of the “Big Data” era and the recent progress of machine-learning technology: instead of conducting time-consuming multi-hull towing-tank experiments, or the complicated computations, existing data on multi-hull systems can be used to “train” a statistical learning algorithm to correctly establish the mapping between the design parameters of multi-hull configurations and the associated hydrodynamic performances of interest. In this way, large quantities of multi-hull configurations can be evaluated conveniently with the fast-inferential capability of the well-trained learning algorithm.

In this chapter, a data-assisted approach for the evaluation and optimization of trimaran configuration is discussed. The trimaran configurations studied in this chapter are steady-state multi-hull systems consisting of three component hulls, with a main hull in the center and the two outriggers placed symmetrically about its centerline on the port side and starboard side. Based on the computational results provided by the MSPM computation explained in Chapter 3, a Deep Neural Network (DNN) is trained to approximate the relation between the design parameters and the associated calm-water transportation efficiency of different trimaran configurations. Then, with the help of the fast, parallel forward-propagation of the DNN, large amount of trimaran configurations can be simultaneously and accurately evaluated, and the optimal configuration with the best calm-water efficiency within given operational requirements can be quickly and accurately determined.

7.1 Brief introduction of DNN and its application to optimizing trimaran configurations

Loosely inspired by the perceptions of animal brains, the Artificial Neural Networks (ANNs) is made out of a collection of connected computing units called the artificial neurons (or “neurons” for short in the following discussions) to allow information flow in the network [88]. It should be mentioned that in the present study, the usage of ANN is restricted in the area of supervised learning, that the network is trained with pre-labeled instances to infer a complex mapping function between a set of input and the expected output. A typical sequential, multilayer perceptron (MLP) ANN [89] consists of three major parts, namely

- Input layer: the layer to receive input parameters.
- Hidden layer: layers populated with artificial neurons to process the values passed from the previous layer and pass the results onto the next layer.
- Output layer: the last layer of the network which gives out the output results.

Data moves from the input layer to the output layer during forward propagation, while the flow in the opposite direction is called backward propagation. In the forward propagation, a neuron firstly receives a vector of values $\mathbf{x}$ from its counterparts on the previous hidden layer and converts them into z score by a linear transformation

$$z = \mathbf{w}^T \mathbf{x} + b, \quad (7.1)$$

where $\mathbf{w}$ and b are the weight and bias of the neuron, respectively. Then, the z score is further processed by a nonlinear function, which is frequently termed as the activation function, to introduce complexity in the data fitting and separate ANN from simple linear regression. A typical example for the activation function, which is also utilized in the current study is the Rectified Linear Unit [90] (ReLU), defined as

$$f(z) = \begin{cases} z, & \text{if } z \geq 0 \\ 0, & \text{otherwise} \end{cases} \quad (7.2)$$

The outcome given by the activation function is then passed onto neurons of the next layer for the same treatment as explained above until the output layer is reached.

With the help of today’s high-performance computing (HPC), especially the massively parallel computing power provided by the GPUs, ANN can be constructed with much more hidden layers and neurons to approximate very complicated mapping functions, and the Deep Neural Networks (DNNs) are, therefore, invented. A schematic of the DNN used in the present study is shown in Fig. 7.1, which will be further explained in Section 7.4.

Before being put to use, DNN “learns” to approximate a mapping function by training with a pre-labeled training set. The inputs of the instances in the training set are first submitted at the input layer for forward propagation on the network. The outcomes, i.e. the network predictions acquired at the output layer, are later compared with the expected outcomes of the instances, and the training errors, i.e. the differences between the network predictions and the expected outcomes, are computed by a loss function. Then, the weights and bias of neurons on the hidden layers are adjusted during backward propagation in the direction of minimizing the training error with gradient-descent based methods [88]. Each training cycle on all instances in a training set is called an epoch, and the process of training becomes feeding the network with training sets for sufficient number of epochs until the training error is reduced under a desirable threshold.

The performance of the trained DNN is evaluated on the test set, which is a set of pre-labeled instances unseen by the DNN in the training process. To estimate the performance

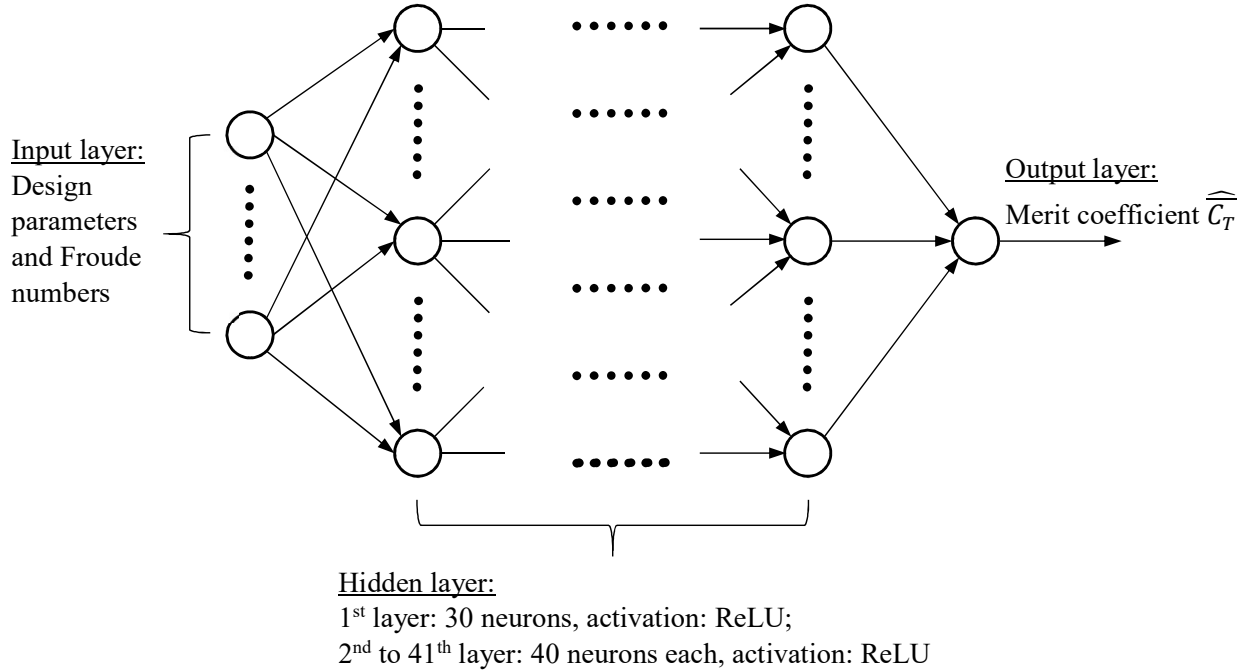


Figure 7.1: Scheme of the structure of the DNN of the present study.

of a DNN on real-world datasets, instances of the test set are submitted to the DNN for forward propagation, then the network predictions are compared with the expected values of the test set, and the error acquired is called the test error. The test error of a DNN should be minimized in the training process, and a well-trained DNN should perform with similar accuracy across the training set and the test set, so that it does not overfit the training instances and provide reliable results on the real-world data. Further, it should be mentioned that another set of pre-labeled data called the validation set, which is not implemented in the present study, is usually deployed for tuning the hyperparameters of the DNN to facilitate the training process and achieve better accuracy.

Most noticeably, since the forward and back propagation of each instance on the network is independent, the flow of information on a DNN can be significantly accelerated with parallel computation. Modern popular Deep Learning framework such as Tensorflow [46] utilize modern GPUs [91] to facilitate DNN-related tasks, and DNNs with complex structures can be quickly trained and tested to accurately fit a complicated data pattern. As a result, a large amount of instances can be accurately and quickly evaluated on a well-trained DNN, which makes the large-scale search for optimal solutions efficient and convenient.

In the present study, DNN as introduced above is utilized to facilitate the evaluation and optimization of calm-water transportation efficiency of trimaran configurations. Major design parameters which represent the geometries and arrangements of the component hulls, and the forward speed (Froude number) of the trimaran configuration are taken as the input of the DNN. Meanwhile, the output of the network is the total calm-water resistance of the

trimaran configuration, which is normalized by the sum of the displacements of all component hulls to represent the cost-to-transport ratio. Therefore, with sufficient computational results provided by the MSPM computation, the DNN can be trained to accurately establish the mapping function between design parameters and the calm-water performance of different trimaran designs and configurations. The DNN is then utilized to evaluate a large amount of trimaran configurations simultaneously with parallel forward propagation, and the optimal configuration with the best overall calm-water transportation efficiency can be quickly and accurately determined.

7.2 Scope of the present optimization of trimaran configurations

Major design parameters

It is usually infeasible to include every detail, especially those of the complex hull geometry in the ship design process. Therefore, several key design parameters which are discovered to have substantial influences on the calm-water performances of mono-hull [47] and multi-hull [92] vessels are selected for the present evaluation, they are

- The length-to-beam and beam-to-draft ratios, L/B and B/T , of each component hull.
- Hull-spacing parameters: normalized stagger $\overline{st}$ and separation $\overline{sp}$.
- The block coefficients C_B and the prismatic coefficient C_P of each component hull, defined as:

$$C_B = \frac{\nabla}{LBT}, \quad C_P = \frac{\nabla}{A_M L}, \quad (7.3)$$

where A_M is the mid-section area of the hull. Block coefficient C_B is an indicator of the fullness of the hull, while prismatic coefficient is related to the displacement distribution along the hull and fullness of the bow and stern. A ship with fuller hull form and more evenly distributed displacement along the length of the hull will usually possess larger values of C_B and C_P . The parameters associated with the main hull and outrigger are denoted with subscripts m and o , respectively.

- The length ratio between the outrigger and the main hull L_o/L_m .

Evaluation metric and the optimization objective

The evaluation metric, i.e. the calm-water transportation cost of a trimaran is represented by the merit coefficient defined as

$$\overline{C_T} = \frac{R_T}{\rho g (\nabla_m + 2\nabla_o)}, \quad (7.4)$$

where ∇_m and ∇_o represent the displaced volume of the main hull and the outrigger, respectively. Thus, for a given forward speed, vessels with smaller values of $\overline{C_T}$ achieve better transportation efficiency by requiring less power to deliver the same amount of payload.

However, in practice ships do not travel with a fixed forward speed. In addition, it has been shown in Chapter 5 that effect of the wave interference for a multi-hull system is substantially influenced by its forward speed, and the wave interference for one speed may be beneficial and turn detrimental for another. Therefore, a more realistic indicator on the transportation efficiency of a trimaran configuration should consider its operational profile [93] by incorporating the operating speed ranges and their associated distribution of likelihood. In this regard, the objective of the following optimization on trimaran configurations is given by the Aggregate Power Consumption Index β defined as

$$\beta = \int_{F_{n_l}}^{F_{n_u}} w(F_n) \overline{C_T} F_n dF_n, \quad (7.5)$$

where F_{n_l} and F_{n_u} are the lower and upper bounds of the target Froude number range, and w is the weight function specifying the probability distribution of F_n in the operating speed range. Therefore, trimarans with small β enjoy better overall calm-water transportation efficiency for the specified operational profile. Eventually, the present optimization of trimaran configurations can be summarized as finding a set of design parameters as introduced in Section 7.2 within a given design space, such that the value of β as defined in Eq. (7.5) is minimized for a target Froude number range and distribution.

7.3 Trimaran design space and the preparation of training and test sets

In the present study, the data for the training and testing of the DNN for the evaluation and optimization of trimaran configurations is provided by the computational results of the MSPM introduced in Chapter 3. For this purpose, the MSPM computation is slightly modified to accommodate the three component hulls with symmetric hull arrangements port and starboard. In addition, the three component hulls experience coupled sinkage and trim as one rigid body rather than having independent values like the case in Chapters 3 and 4.

Furthermore, since the MSPM computation can only supply the inviscid resistance component, i.e. the wave resistance in the present study, the frictional drag is provided by the ITTC line-correlation formula of Eq. (4.4), and the total resistance of a trimaran is computed by Eq. (4.3), with the form factor k assumed zero.

Moreover, the training instances for the DNN should come with adequate variations, so that the resulting DNN can give accurate predictions for trimaran configurations with various sets of design parameters. However, detailed information on existing trimaran designs is scarce, and the offset tables for the component hulls are usually unavailable. Therefore, in the present study, the shapes of the component hulls are generated based on the classic

Wigley hulls [94] with slight modifications, and the hull-surface function [48] of which is given by

$$y_h = \pm \frac{B}{2} \left(1 - \left|\frac{2x_h}{L}\right|^a\right) \left(1 - \left|\frac{z_h}{T}\right|^b\right), \quad a, b > 0, \quad (7.6)$$

where (x_h, y_h, z_h) describe the hull surface. a and b are two parameters controlling the fullness of the waterplane and cross-section of the hull, respectively. In addition, by the definition of Eq. (7.3), the block coefficient and prismatic coefficient of a modified Wigley hull can be analytically computed as

$$C_B = \left(1 - \frac{1}{1+a}\right) \left(1 - \frac{1}{1+b}\right), \quad C_P = \left(1 - \frac{1}{1+b}\right). \quad (7.7)$$

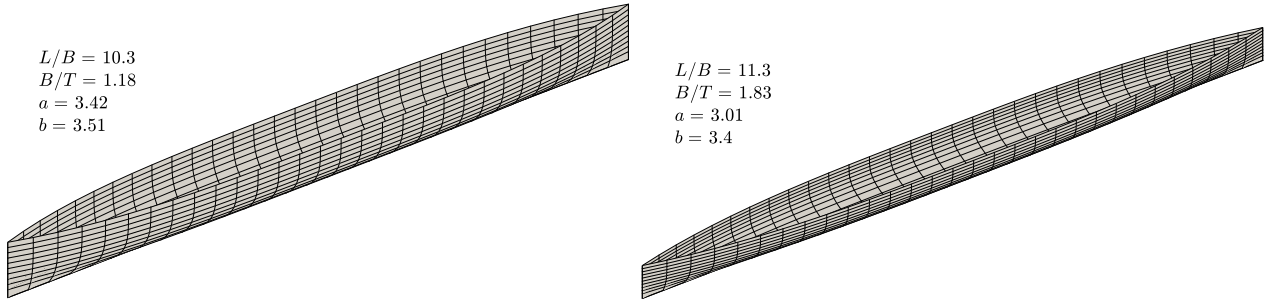


Figure 7.2: Three-dimensional views of modified Wigley hulls of selected parameters.

Consequently, by choosing different sets of values for a , b , L/B and B/T , component hulls with different shapes can be easily generated with the help of Eq. (7.6). Together with the length ratio between the main hull and outrigger L_o/L_m and hull-spacing parameters $\overline{st}$ and $\overline{sp}$, trimaran configurations with various design parameters as introduced in Section 7.2 can be conveniently generated. In this manner, in the present study 3,000 different trimaran configurations are randomly generated, and each of the design parameters are randomly selected within the range specified in Table 7.1, while the length of the main hull is fixed at 10m. Then, the generated trimaran configurations are evaluated by the MSPM computation for their values of $\overline{C}_T$ for twenty-one Froude numbers between 0.25 and 0.5.

Table 7.1: Ranges for the trimaran design parameters.

Parameters	$\overline{st}$	$\overline{sp}$	L_o/L_m	L_m/B_m	B_m/T_m	L_o/B_o	B_o/T_o	C_{B_m}	C_{P_m}	C_{B_o}	C_{P_o}
Minimum	0	0.15	0.3	12.003	0.749	12.003	0.749	0.366	0.600	0.366	0.600
Maximum	1	0.55	1	19.997	2.251	19.997	2.251	0.637	0.799	0.637	0.799

7.4 DNN implementations

Training and testing of the DNN

Based on the computational results introduced in the previous section, the DNN is trained to carry out the same task as the MSPM computation, which is to accurately predict the value of $\overline{C_T}$ based on the input variables consisting of the design parameters listed in Table 7.1 and a Froude number between 0.25 and 0.5.

The resulting DNN consists of 41 hidden layers, with 30 neurons placed on the first hidden layer, and the rests are populated with 40 neurons each, as shown in Fig. 7.1. All neurons are activated by the ReLU function of Eq. (7.2), and the network is initialized with the method introduced in He [95] and optimized with Adaptive Moment Estimation (Adam) to minimize the mean absolute percentage error (MAPE) δ [96] defined as:

$$\delta = \frac{1}{m} \sum_{i=1}^m \left| \frac{\overline{C_{T_i}} - \hat{\overline{C_{T_i}}}}{\overline{C_{T_i}}} \right| \times 100\%, \quad (7.8)$$

where $\hat{\overline{C_T}}$ is the DNN prediction of merit coefficient $\overline{C_T}$, and m is the number of training instances provided by the MSPM computation. The hyper-parameters for the Adam optimizer are kept the same as those in Kingma et al. [97] except for the learning rate (i.e. the step size in the gradient descent), which is chosen as 0.00049. Out of the 3,000 computationally evaluated trimaran configurations, 2700 are randomly chosen as the training set and the rests are taken as the test set. A training/test instance consists of a set of design parameters and one Froude number as input, and a corresponding value of $\overline{C_T}$ as output, and each of the trimaran configuration is evaluated for twenty-one Froude numbers in the MSPM computation. Consequently, there are a total of 56,700 and 6,300 instances in the training set and test set, respectively. The TensorFlow [98] and Keras deep-learning frameworks are utilized for the building, training and testing of the DNN as well as the following DNN-based evaluation of trimaran configurations. The network is trained with 8,000 epochs and a batch size of 2,000 on a GTX 1080 Ti GPU. Each epoch of training took about 0.2 seconds, and the entire training process is completed within 30 minutes. The ultimate training and test error measured in MAPE are 2.13% and 2.99%, respectively. Consequently, on the trained DNN, more than 5,000 trimaran configurations can be simultaneously evaluated with the parallel forward propagation in less than half a second.

Since the DNN is trained on data provided by the MSPM computation, the accuracy of the DNN predictions is mainly determined by two factors: the capability of the DNN in fitting and approximating the MSPM computational results, and the accuracy of the MSPM computation itself in the evaluation of trimaran calm-water performance. While the former task is accomplished with sufficient training and properly-designed network structure, which is reflected by the small training and test error, the latter task calls for reliable third-party validations. To this end, the computational results of $\overline{C_T}$ given by the MSPM computation and the DNN predictions are compared with recent published numerical and experimental

results in Wang and Lu [1], in which the model-scale towing-tank tests and the fully nonlinear potential-flow panel-method computation were performed. The design parameters of the Wigley trimaran investigated in Wang and Lu [1] are listed in Table 7.2, and the plot comparing results given by the different approaches are given in Fig. 7.3.

Table 7.2: Configurations of the trimaran studied in [1].

$\overline{st}$	$\overline{sp}$	L_o/L_m	L_m/B_m	B_m/T_m	L_o/B_o	B_o/T_o	C_{B_m}	C_{P_m}	C_{B_o}	C_{P_o}
0.3	0.15	0.368	12.5	2.25	12.5	2.25	0.444	0.667	0.444	0.667

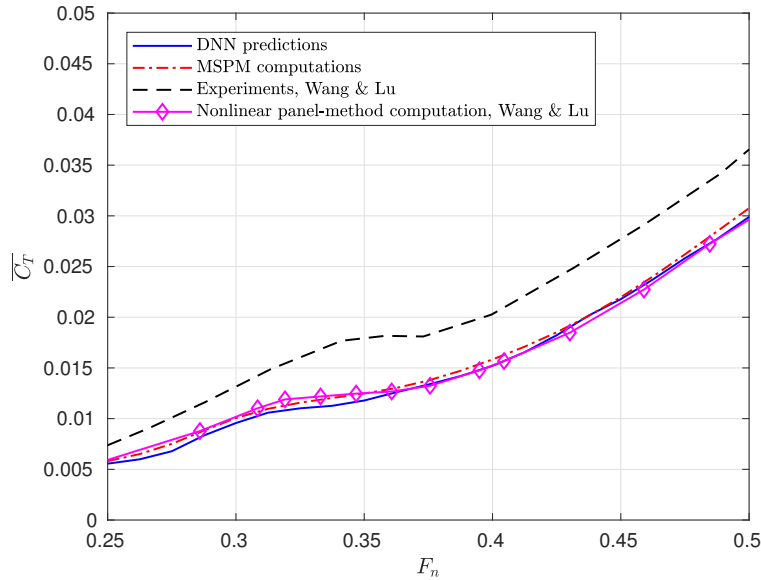


Figure 7.3: Results from MSPM computation and DNN predictions in comparison with results in Wang and Lu [1].

Good agreements are found among the results of the MSPM computation, DNN predictions and the published computational results, but they are all almost consistently smaller than the experimental measurements. Nevertheless, for an optimization task, the accurate prediction of the variation trends is more crucial than capturing the absolute performances of a trimaran configuration. In short, it is concluded that the MSPM computational results on the trimaran calm-water efficiency come with satisfactory accuracy, and very similar performance can be expected from the DNN predictions. Therefore, the trained DNN can be utilized instead of the time-consuming direct hydrodynamic computation to accurately and quickly evaluate a large amount of trimaran configurations, and the large-scale search for the optimal configuration can be efficiently carried out.

DNN-assisted optimization of trimaran configurations

One difficulty in the optimization of multi-hull configurations, which was alluded to in Chapter 5, is caused by the periodic variation in wave resistance with stagger as a result of the phase differences between the two sets of ship waves. As a result, it may be hard for conventional optimization approaches such as gradient descent and clustering [99] to reach the global optimum in practice.

Nevertheless, since the DNN can evaluate thousands of trimaran configurations in less than a second, a straightforward Monte-Carlo type search can be conducted to find the configuration with minimal β , and the procedure of which is described as below:

1. Specify a search range by placing a limit on each of the design parameter and the Froude number. The search range should not exceed the range listed in Table 7.1. In addition, the number of divisions between F_{n_l} and F_{n_u} , n_U , is also specified for the computation of β by Eq. (7.7).
2. Randomly generate n_1 trimaran configurations within the search range defined in the previous step in the manner introduced in Section 7.3.
3. Evaluate the generated trimaran configurations on the DNN for their values of $\overline{C_T}$ with the n_U Froude numbers, and compute the value of β for each configuration by Eq. (7.5).
4. Within the evaluated configurations, identify the best trimaran configuration with the β_{min} and the associated set of design parameters.
5. Repeat Step 2 to 4 for k times until the result on the optimal configuration converges. In the iteration process the search range for each of the design parameters can be further shrank down based on the discovered optimal configuration in the last iteration to facilitate the search for the global optima.

In addition, to achieve higher confidence in the discovered global optima, a larger n_1 can be adopted to distribute more configurations in the defined search range, and the entire Monte-Carlo search can be repeated multiple times. Moreover, the optimal configuration given by the above process can be further refined by an additional search targeting each of the design parameters, and the steps of which are provided below:

1. Select one of the design parameter to continuously vary within a search range of interest, while the rest of the design parameters are kept the same as those of the optimal configuration of the previous iteration. In this process n_2 trimaran configurations are generated.
2. In the same fashion like Step 3 of the Monte-Carlo search, the newly-generated configurations are evaluated for their associated values of β . Update the value of β_{min} and the optimal configuration if needed.

3. Pick another design parameter and repeat Steps 1 and 2, until the search is completed for all design parameters.

In practice, the above refined search can also be repeated several times to validate the convergence of the optimal configuration. In addition, the sequence of parameters being optimized can also be randomized between iteration. The above optimization searches can be completed in very little time with the help of the fast, parallel DNN forward propagation. For instance, the search with $n_1 = n_2 = 3000$, $k = 12$ and $n_U = 11$ takes less than 20 seconds on a GTX 1080 Ti GPU.

7.5 Case study - the optimal configuration for high-speed trimaran design

For demonstration purposes, a search for the optimal trimaran configuration with the aforementioned DNN is conducted. The search range for each design parameter is identical to the one provided for the MSPM computation introduced in Table 7.1. The target Froude number range is set between $F_{n_l} = 0.4$ and $F_{n_u} = 0.45$ based on the common operating speeds for high-speed ferries [100], and the weight function w is taken as $1/F_n$ for the evaluating speed range.

The resulting optimal configurations and the associated design parameters and the values of β are listed in Table 7.3. The results are acquired with $n_1 = n_2 = 3000$, $k = 12$ and $n_U = 11$. Furthermore, the trimaran configuration as well as the value of β for the trimaran introduced in Wang et al. [1] are also listed for comparison.

Table 7.3: Design parameters for the optimal trimaran configurations and the associated values of β for $F_n \in [0.4, 0.45]$.

Design parameters and β	$\overline{st}$	$\overline{sp}$	$\frac{L_o}{L_m}$	$\frac{L_m}{B_m}$	$\frac{B_m}{T_m}$	$\frac{L_o}{B_o}$	$\frac{B_o}{T_o}$	C_{B_m}	C_{P_m}	C_{B_o}	C_{P_o}	$\beta \times 10^4$
After the Monte-Carlo search	0.502	0.187	0.970	12.810	1.619	14.085	1.883	0.478	0.718	0.522	0.766	6.204
After the refined parameter-based search	0.477	0.150	1.0	12.5	1.5	13.644	1.974	0.455	0.770	0.519	0.770	5.574
Wang & Lu (2011)	0.3	0.15	0.368	12.5	2.25	12.5	2.25	0.444	0.667	0.444	0.667	7.559

It is discovered that the refined parameter-based search makes slight adjustment to the configuration given by the Monte-Carlo search, and further decreases the value of β by around 10%, which is more than 26% lower than that of the trimaran configuration introduced in Wang et al. [1].

Furthermore, to reveal the influence of each design parameter on the overall transportation efficiency of trimaran configurations, each of the eleven design parameters are varied

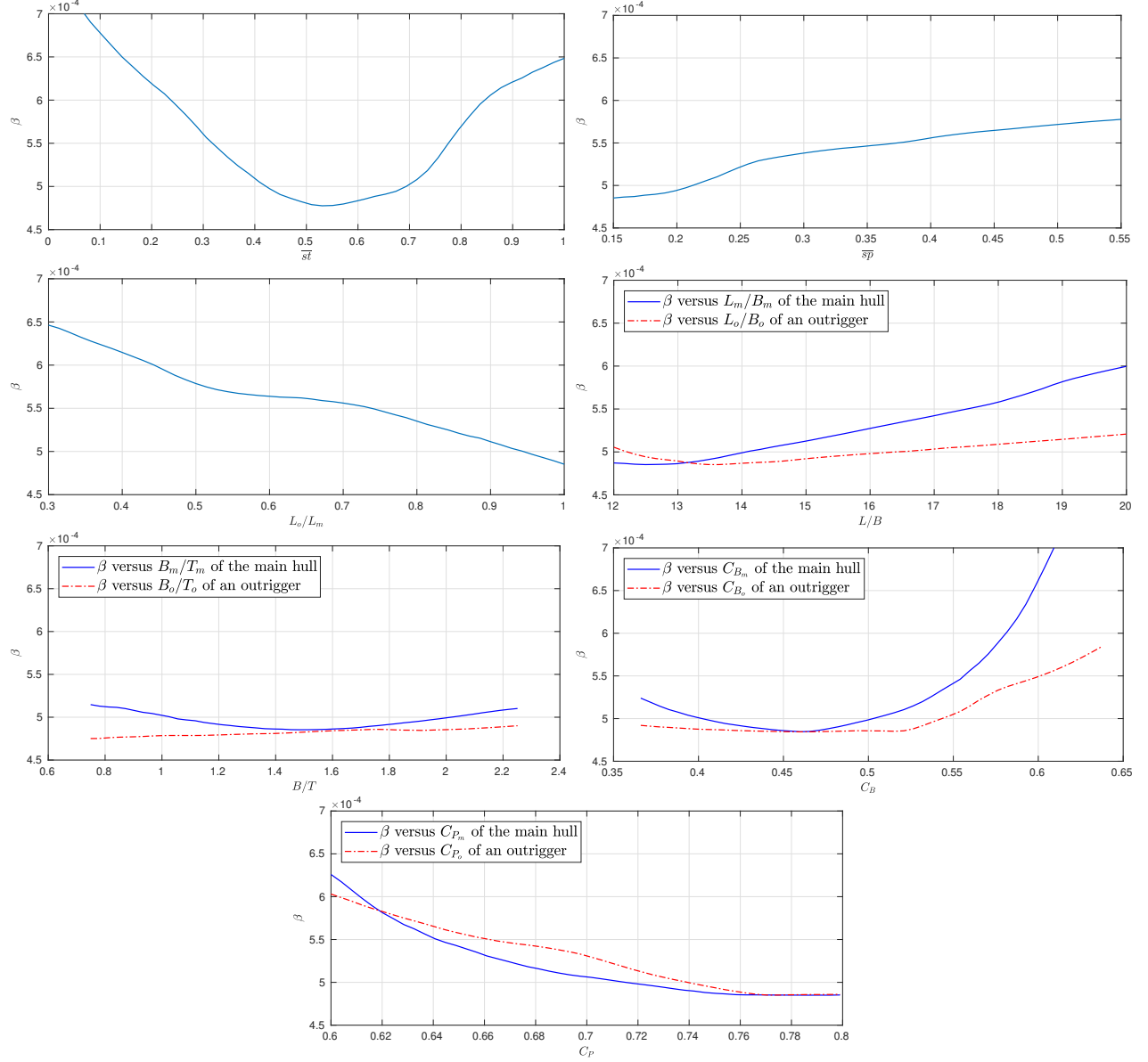


Figure 7.4: Variations of β with each of the design parameter around the optimal configuration for F_n between 0.4 and 0.45.

individually around the optimal configuration, and the values of β of the associated configurations are plotted in Fig. 7.4.

Previous studies [57, 10] have shown that the wave-interference effect is the most significant for multi-hull systems consisting of identical component hulls, and a half-ship length stagger is associated with the largest wave-resistance reduction [12, 3] for $F_n \approx 0.4$. Interestingly, the optimal trimaran configuration given by the present optimization search exhibits very similar characteristics, that the main hull and outriggers are almost identical in size

and shape. In addition, the normalized stagger $\overline{st}$ is discovered to substantially affect the value of β , and β_{min} occurs with $\overline{st} \approx 0.5$

Furthermore, having larger outriggers with similar size and shape like the main hull also introduce larger wetted-surface area and, therefore, higher frictional drag [100]. However, it is shown that such increase in resistance is largely outweighed by the associated wave-resistance reduction brought by the strong beneficial wave interference [22].

Moreover, modern popular trimaran designs usually adopt very slender component hulls ($L/B > 15$) to achieve exceptional resistance performances [101, 102]. However, the discovered optimal configuration shows that a smaller length-to-beam ratio and slightly fuller hull forms with larger C_B and C_P are in fact more desirable for better transportation efficiency.

Table 7.4: Optimal configurations for different Froude number ranges.

F_{nl}, F_{nu}	$\overline{st}$	$\overline{sp}$	$\frac{L_o}{L_m}$	$\frac{L_m}{B_m}$	$\frac{B_m}{T_m}$	$\frac{L_o}{B_o}$	$\frac{B_o}{T_o}$	C_{B_m}	C_{P_m}	C_{B_o}	C_{P_o}	$\beta \times 10^4$
0.35, 0.4	0.378	0.15	1.0	12.5	1.5	12.5	2.0	0.414	0.762	0.468	0.750	3.978
0.4, 0.45	0.477	0.15	1.0	12.5	1.5	13.644	1.974	0.455	0.770	0.519	0.770	5.574
0.45, 0.5	0.572	0.15	1.0	12.778	1.5	15.322	1.682	0.475	0.770	0.540	0.768	7.316

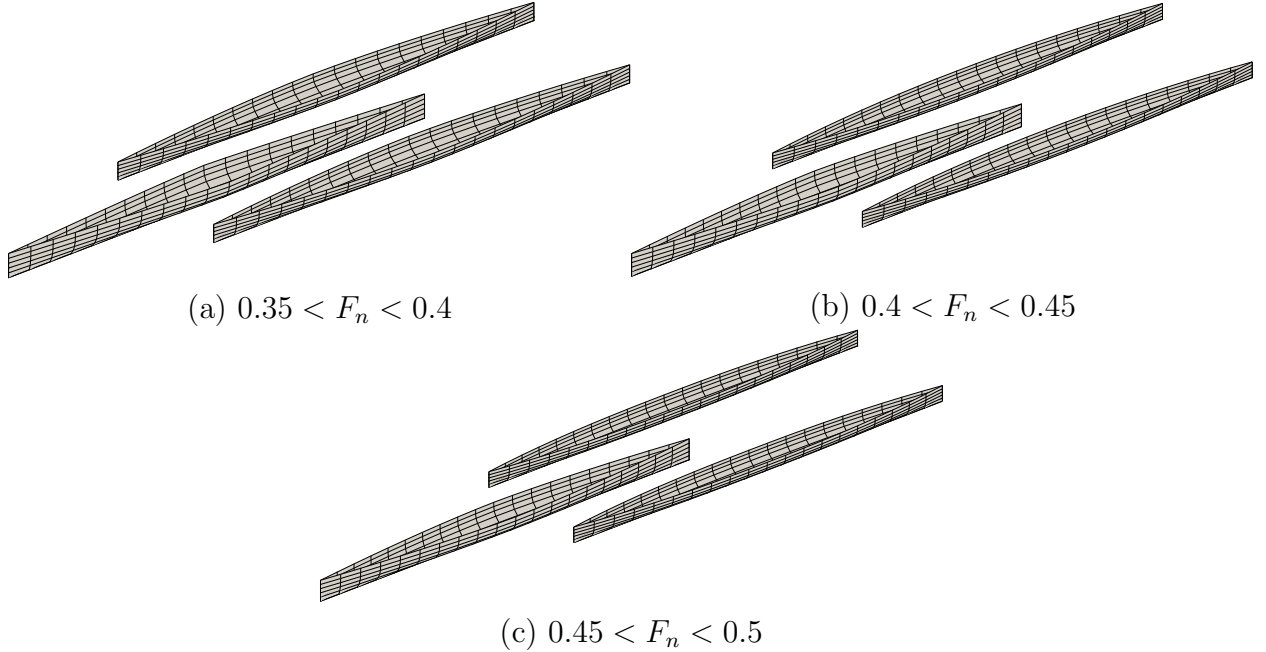


Figure 7.5: Three-dimensional views of the optimal configurations for different Froude number ranges. Grid density is reduced for illustrative purpose.

Furthermore, to investigate the differences among optimal trimaran configurations for different forward speed ranges, the present DNN is also utilized to conduct similar optimization searches for other Froude number ranges, and the results are listed in Table 7.4, and the

associated three-dimensional views of the optimal trimarans are shown in Fig. 7.5. It is observed that the optimal configurations of higher forward speeds prefer larger stagger values, since vessels traveling at higher forward speeds generate waves of greater wavelength [57]; therefore, greater longitudinal hull spacing is needed to achieve beneficial wave interference. In the meantime, design features like small separation and component hulls with similar sizes and shapes are still preferred for the optimal configurations. In a word, the desirable design features of trimarans based on hydrodynamic analysis and engineering practices are indeed recovered by the DNN based optimization, and its effectiveness in quickly and accurately locating the optimal configuration within a predefined search range is validated.

To conclude, in the process of optimizing a trimaran configurations for better calm-water transportation efficiency, strong beneficial wave interference for wave-resistance reduction plays a key role. As a result, the optimized trimaran should be made out of component hulls of similar dimensions and small separation to obtain stronger wave interference in general, while the stagger should be chosen according to the target forward speed range to achieve out-of-phase interference for wave-resistance reduction, which has also been demonstrated in detail in Chapter 5. Lastly, with proper training, the DNN has proven to be capable to accurately and efficiently give out the optimal configurations with very little computing cost, which can be very helpful in accelerating the designing iterations of trimarans.

Chapter 8

Conclusions and Future Works

The focus of this thesis, its major findings, and future prospects are summarized in the section below.

8.1 Wave resistance and wave pattern of steady-state di-hull systems

The wave resistance and wave pattern of a di-hull system are shown to be substantially affected by the wave interference between the two component hulls. With proper configurations, the far-field wave interference can significantly reduce the wave elevation and wave resistance of the di-hull system. In particular, it is shown by both experimental and computational investigations that wave-resistance reduction mostly takes place for the component hull traveling at the rear of the formation.

The Multi-hull Simple-source Panel Method (MSPM) introduced in this thesis is validated against experimental measurements to provide reliable predictions on the wave resistance, sinkage and trim of di-hull systems with various hull configurations. In addition, it is discovered that the wave resistance and wave pattern of a di-hull system are strongly influenced by the interference of waves with the strongest free-wave spectral density of its component hulls in isolation; moreover, the Wave-angle Alignment Method (WAM), which aims at canceling waves in the direction of the highest spectral density through simple peak-to-trough alignment, is validated as a convenient and efficient alternative for qualitative wave-resistance optimization of di-hull systems.

Nevertheless, the near-field interaction between component hulls, especially the wave-diffraction effect of the hull bodies show non-negligible influence on the wave resistance and wave pattern of a di-hull system, which may impair the accuracy of predicting wave-resistance reduction by just using far-field-based methods.

To draw a more complete understanding of the steady-state di-hull interactions, further study can be pursued in the following directions:

- Study di-hull systems made out of distinct hulls and more variations of stagger, separation and forward speeds;
- Conduct high-fidelity computation of di-hull systems with viscous-fluid simulations such as RANS and LES, so that effects of viscosity and near-field interactions can be better understood;

8.2 Unsteady multi-hull system: ship-to-ship hydrodynamic interactions in moderate-speed overtaking maneuver

The interactions between two hulls during calm-water overtaking maneuvers in a deep, infinite fluid domain is studied both computationally by a linearized potential-flow panel method (MSPM), and experimentally by towing-tank tests. The results of the potential-flow panel method is validated by the experimental measurements; however, the application of linearized free-surface condition likely caused excessive fluctuations of loads, especially with positive instantaneous stagger between hulls.

When the speed difference between hulls is large, it is shown that the slower vessel experiences loads with significantly larger magnitudes and stronger fluctuations, while the loads on the overtaking vessel resemble those of a mono-hull moving steadily in isolation in calm water. It is also observed that with shrinking speed differences between vessels, the loads on both hulls approach those of a steady-state di-hull system of the same hull configuration.

The unsteady free surface around hulls is found to have non-negligible influences on the loads of both hulls. The free-surface elevations as well as the unsteadiness of the wave patterns are responsible for the different loading conditions on the two hulls during the overtaking process. In addition, the interference between two sets of unsteady ship waves is also found to have non-negligible influence on the surge force of the overtaking hull.

To build a more complete knowledge of moderate-speed ship-to-ship interaction, present study can be further expanded in the following directions:

- Conduct similar research on encountering maneuvers, where two ships move towards each other in parallel courses;
- Conduct CFD simulations on ship-to-ship interactions to better capture the influence of fluid viscosity;
- Conduct research on close-quarter maneuvering of vessels with nonparallel courses, the results of which will be of paramount importance for collision avoidance.

8.3 Optimization of trimaran configurations with DNN

To optimize trimaran-type vessels for better calm-water transporting efficiency, a Deep Neural Network (DNN) is trained on the computational results provided by the steady-state MSPM to accurately establish the mapping between key design parameters (hull spacing, length ratio and the geometric coefficients of component hulls) of a trimaran configuration and its calm-water performance. With the parallel forward propagation on the DNN, a large

amount of different trimaran configurations can be quickly evaluated with similar accuracy as the MSPM computation itself. Then, using the trained DNN, the optimal configuration with the minimum transporting cost over target speed range can be easily determined. In this manner, the design and optimization process of trimaran vessels can be greatly accelerated with satisfactory accuracy and efficiency.

Moreover, the optimization results for selected test cases show that for higher forward speed ranges, an efficient trimaran should be designed to best harness the beneficial wave interference for wave-resistance reduction. Therefore, configurations with closely-spaced component hulls of similar size and shapes are preferred. In particular, the value of stagger between the main hull and the outriggers, which is discovered to exert substantial influence on the calm-water resistance of a trimaran, should be carefully chosen according to the forward speeds to achieve beneficial wave interference.

The present study can be expended in the following meaningful directions:

- Training and testing of the DNN with data provided by high-fidelity computational and experimental results. Since the accuracy of the DNN predictions is ultimately limited by the quality of the training and testing data, the DNN trained with data from high-fidelity approaches, such as the towing-tank experiments [103] and CFD computations [17] should be able to provide predictions with greater accuracy.
- Incorporate more geometric features of the component hulls, especially those of the existing trimaran hull forms from other research and engineering practices. In this way, the DNN may be trained to provide more sophisticated results.
- Investigate other design aspects of trimarans. In a similar manner, the DNN can be implemented to investigate other important hydrodynamic or structural aspects of trimarans, such as seakeeping performances, structural reliability, and maneuverability.

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