Numerical Study on Seakeeping Performance of Hydrofoil Assisted Submersible Trimaran (HAST)

By

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ABSTRACT

Numerical Study on Seakeeping Performance of Hydrofoil Assisted Submersible Trimaran (HAST)

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The present study focuses on the seakeeping characteristics of Hydrofoil Assisted Submersible Trimaran (HAST). The seakeeping performance is an important part in ship design because it effects the comfort, control, resistance and safety. The motions in wave will reduce the performance of ship which is designed and optimized in calm water. The heave and pitch motions of HAST in calm water and regular waves have been investigated by STAR CCM+, a CFD analysis program. The theory of numerical simulation based on STAR CCM+ has been presented.

The basic theory of seakeeping performance studies has been discussed. A series of simulations have been undertaken using STAR-CCM+ for various speed regime. The simulation setting is presented in this thesis. Overset mesh is applied to increase the accuracy and efficiency of numerical simulation. In addition simulation data of a trimaran of similar dimensions has been used for comparative analysis. The mean value of trim angle, sinkage in each working conditions are given in this thesis. The oscillation amplitude of heave and pitch is also collected and shown to describe the seakeeping performance of HAST. The hypothesis was that a non-conventional hull form such as HAST would
provide reasonably better seakeeping performance than a traditional trimaran hull form.

The result indicated so far shows that HAST has better heave response when compared against a traditional trimaran hull form at high-speed. However, the pitch response of HAST appears to be a lot worse than a traditional trimaran.
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Nomenclature

$A_k$ Wave Steepness

$F_n$ Froude Number

$F_{nv}$ Froude Volume Number

$k$ Wave Number

$L$ Wave Length

$V$ Velocity

$z$ Heave Height

$z_a$ Heave Amplitude

$\theta$ Trim/Pitch Angle

$\theta_a$ Pitch Amplitude

$\zeta$ Wave Amplitude

$\rho$ Density

$\omega$ Frequency
## Acronyms

<table>
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<th>Acronym</th>
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<tr>
<td>6-DOF</td>
<td>Six Degrees of Freedom</td>
</tr>
<tr>
<td>AMECRC</td>
<td>Australian Maritime Engineering Cooperative Research Centre</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics</td>
</tr>
<tr>
<td>COG</td>
<td>Center of Gravity</td>
</tr>
<tr>
<td>DFBI</td>
<td>Dynamic Fluid Body Interaction</td>
</tr>
<tr>
<td>HAST</td>
<td>Hydrofoil Assisted Submersible Trimaran</td>
</tr>
<tr>
<td>ITTC</td>
<td>International Towing Tank Conference</td>
</tr>
<tr>
<td>RANS</td>
<td>Reynolds-Averaged Navier–Stokes</td>
</tr>
<tr>
<td>VOA</td>
<td>Volume of Air</td>
</tr>
<tr>
<td>VOF</td>
<td>Volume of Fluid</td>
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Dedication

To my Mother, Father and Grandmother.
Chapter 1

Introduction

1.1 Hydrofoil Assisted Submersible Trimaran

This thesis is focused on the seakeeping performance of a Hydrofoil Assisted Submersible Trimaran (HAST) in head seas.

In this thesis, the model that has been studied is named as Hydrofoil Assisted Submersible Trimaran (HAST) which has been presently patented by Pan (2015) CN104787275 (A) and approved in 2017. HAST is a special kind of marine vehicle whose displacement can vary between 200 and 800 tonnes. It is able to cruise as a trimaran on surface or as an underwater submersible craft by changing the configuration of the outriggers. HAST is equipped with both of gas and electric propulsion systems which will enable the craft achieve a speed of more than 50 knots when cruising on surface. The electric propulsion system also allowed to generate low noise propulsion underwater. The resistance and seakeeping performance of HAST in calm water has been investigated by Pan (2017) but the performance in head seas is not done yet. The motions in wave may reduce the performance of crafts so that this thesis focus on HAST motions in head seas.

The structural details of HAST is shown in Figure 1. The various components of the structural configuration are listed in Table 1. HAST can be transformed
by opening and closing of the side hull outriggers into a trimaran or submarine. This thesis is primarily focused on seakeeping characteristics in afloat condition when side hulls are open with hydrofoils deployed.

Figure 1: HAST Structural Configuration (Pan, 2015)
The hydrofoil is thin sheets of material submerged in flowing water. It can be regarded as a “submerged airfoil”. The density of water is much higher than air so a foil in water will produce substantially more lift than in air.

Figure 2, the Sustension Triangle (Clark, Ellsworth and Meyer, 2004), provides a visual comparison of planning craft, multihulls, hydrofoil ships, hovercraft and hybrids about their buoyancy, powered lift force and dynamic lift force. The figure shows hydrofoil assisted crafts have high dynamic lift force which is different from normal hull forms. It means when this kind of crafts reach to high-speed the hulls are supported with the force generated with hydrofoil.
The first hydrofoil assisted passenger boat appeared in 1952 (SRI International, 1961). The boat sailed out of water with the supports of underwater "wings". Nowadays, the demand for providing smooth and comfortable ride in waves for high-speed passenger craft is increasing. There are many types of high-speed craft which have been tested but it is not easy to obtain a comfortable ride while keeping high-speed performance in waves. To solve this problem, a fully submerged hydrofoil is a good solution to meet such a requirement. (AYRS Member, 1970)

The development of hydrofoil assisted catamarans started in 1970’s. The early designs focused on small planing catamarans. The hydrofoils were assembled between the demi-hulls which can reduce the resistance by about 40% (Kihara et al., 1995). The hydrofoil also solved the problems of poor seakeeping and wet deck slamming because of hulls being lifted out of water. Since the early developments obtained good performance, the application of a
hydrofoil system has been used to improve seakeeping performance, ride control and resistance reduction. Over the last twenty years, the application of hydrofoil assistance in high-speed craft is not only getting popular but also being widely deployed. It has been used on variety of high-speed crafts including work boats, military crafts, pleasure crafts and high-speed ferries (Migeotte and Kornev, 2004).

1.2 Seakeeping Performance

The seakeeping performance of ships in a seaway is one of the most significant aspects in ship design effects comfort, safety, control, added resistance and some other characteristic. The ships are usually optimized for calm water but the performance in waves may be different. A large resonant motions may lead to reduction of stability and safety. Researchers have proposed a wide variety of arrangements to reduce ship motions in waves.

The seakeeping performance has generally been studied in a wave tank test using a scaled model or full scale seal trial test. Each of these investigative methods have advantages and disadvantages. Numerical methods are fast and efficient in quickly assessing the seakeeping characteristics of geometrically similar models and to estimate the initial seakeeping performance of a ship design.

The theory, which the numerical simulation usually based on is 2D strip theories has been developed more than 50 years ago (Migeotte and Kornev,
2004). The traditional strip theory such as Original Strip Method (Kroukovsky and Jacobs, 1957) and Rational Strip Theory (Ogilvie and Tuck, 1969) can predict the linear motions for low-speed vessels but are not applicable for high-speed ships. The high-speed slender body theory was developed by Chapman (1976). This 2.5D theory predict high-speed vessel motions in a wave and has been found to be more practical and efficient than 3D theories.
Chapter 2
Background

Trimaran vessels are currently of interest for many new high-speed ship projects due to their intrinsic hydrodynamic efficiency that can be achieved compared with mono-hull and catamaran hull forms. The resistance and seakeeping performance of high-speed trimaran interested researchers around the world, which is currently simulated numerically to test the performance of marine vessels in the absence of test facility or access to real data. For example, Wu et al. (2011) present a CFD computation of ship motion and added resistance in waves of a high-speed trimaran. The simulation used the finite volume method with basic theories based on governing equations, the Reynolds averaged Navier-Stokes equation, and continuity equations. In these equations, volume of fluid method deals with nonlinear free surface problem, where the incident waves are generated from the inflow boundary by prescribing a velocity profile resembling flexible flap wave maker motions, and the outgoing waves are dissipated inside an artificial damping zone located at the rear part of the wave tank. The results of numerical simulation are compared against the seakeeping experiment for the high-speed trimaran carried out in CSSRC towing tank. While, large-scale model tests has obvious advantages investigating ship hydrodynamics. Jiao et al. (2018) verified the possibility of reproduction the full-scale wave states in coastal sea areas. Significant wave height, mean period, spectral shape and directional spreading are considered to judge the similarity of wave. Various factors such as season, weather, scale of
model, space area and time are taken into account to determine if there are available waves for model measurements. The wave selection principle and methodology proposed in this paper are demonstrated through a case study using coastal wave data of the Puerto Rico, Virgin Islands and the Gulf of Maine (Yu, 2003).

The seakeeping characteristics of a Small Water-plane Area Twin Hull (SWATH) vehicle equipped with fixed stabilizing fins was investigated by experimental and numerical methods (Sun et al., 2017). The calculation methods range from viscous CFD simulation based on an unsteady RANS approach to Boundary Element Method (BEM) based on Three Dimensional Translating-pulsating Source Green Function (3DTP). Responses of ship motions in regular head waves and nonlinear effects on motion responses with increasing wave amplitude were analyzed. Numerical simulations have been validated by comparisons with experimental tests. The results indicate that the heave and pitch transfer functions depict two peaks with the increase of wave length. Comparisons against experimental data and different numerical calculations illustrate that the RANS method predicts ship motions with higher accuracy and allows the detection of nonlinear effects. The heave and pitch transfer functions see a downward trend with the increasing wave amplitude in the resonant zone at low-speed.

Muscat and La Rosa (2014) investigated the separation, stagger and draught layout effect on the resistance performance of a trimaran. The analysis is carried out by numerical simulation. In conclusion, larger separation, less draught and increased stagger was found by these researches to reduce
resistance.

Using numerical simulations, Narita and Kokan (1976) verified that linear theory to be an acceptable method in predicting wave resistance for trimarans and linear theory can also be used to optimize hull form. Narita and Kokan also found that wave resistance reduced when the displacement of side hull is a little larger than center hull.

Also using numerical simulation, Hafez and El-Kot (2012) investigated the influence of stagger variation of the outriggers on hydrodynamic interference of high-speed trimaran. The slender body theory and model were put into software, MAXSURF, obtain the resistance data for 3 different trimarans (Wigley-st, AME CRC -09, and NPL -4a) in calm water.

The seakeeping motions of catamarans and trimarans were numerically simulated using a time domain seakeeping method by Davis and Holloway (2007). The motions were accurately validated by testing over a range of Froude numbers (Fn) from 0.2 to 0.8. They determined that the time domain seakeeping computation based on the two-dimensional transient Green function solution for spatially fixed strips of water, gives accurate motion predictions over a wide range of Froude numbers. That showed that the effect of hull spacing for a catamaran in head seas is quite small. There is a strong increase in motions as the Froude number increases from 0.5 to 0.8, and this has direct consequences with regard to the comparative motion of large ferries of catamaran and trimaran design, since the Froude numbers vary within this range from one design to another. In head seas, the greater length of the trimaran reduces the maximum heave motions relative to wave height as a
consequence of the reduced length-based Froude number. However in quartering and beam seas, the trimaran loses its advantage and has a rolling motion that can be twice that of the catamaran, depending upon frequency.

Uzunoglu and Soares (2018) introduced a numerical 3D modelling method to develop quadrilateral meshes for panel method calculations with an integrated solution to obtain the mass properties from a grid. Their technique aims to offer a more precise and versatile alternative to the spreadsheet type approach repeatedly used in early design stages of marine structures.

Castiglione et al. (2011) also conducted a numerical study that evaluated the ability of the unsteady RANS code (Carrica et al., 2007) to predict the wave-resistance and seakeeping characteristics of high-speed multi-hull ships at high sea states. Numerical analysis includes the effect of wave steepness on ship response, the assessment of ship motion, the natural frequency of the catamaran and the additional resistance in waves. Castiglione’s study investigated the DELFT 372 catamaran use the URANS solver CFDSHIP-Iowa V.4 (Irvine et al., 2008). The code has been proven to have encouraging results for high-speeds (0.3 \( \leq F_n \leq 0.75 \)) and high amplitude (0.025 \( \leq A_k \leq 0.1 \)) (Simonsen et al. 2008). A comparison with the strip theory solution shows that the RANS method can more accurately predict ship motion and allow detection of nonlinear effects. For the selected model (DELFT catamaran model 372), the calculations shows the highest heave occurs at the resonance for all \( F_n \) which reaches an absolute maximum at \( F_n = 0.75 \). For each speed, the maximum pitch occurs at frequencies below resonance and the maximum value occurs at \( F_n = 0.6 \). The maximum additional resistance due to waves is
obtained at \( F_n = 0.45 \). Similar results were found in Simonsen et al. (2008) for KCS container ships tested for multiple hull geometries and higher speeds.

Sahoo et al. (2008) investigated a systematic series of high-speed trimaran hull forms using numerical methods and experimental work. The wave resistance characteristics of the series of trimaran hull forms based on the AMECRC systematic has been tested. The experimental work of trimaran model TRI-9 from the systematic series was carried out. The wave resistance of each trimaran models are also collected by computational fluid dynamics suite, SHIPFLOW, and theoretical slender body theory. A wide variety of data was acquired within the parametric space with various side hull locations.

Ma et al. (2015) researched on the longitudinal position of center of gravity (COG) effect on the resistance of the trimaran planing boat and the motion responses of trimaran planing crafts in regular waves. The towing tank tests were performed on a trimaran planing hull in calm water which aim to verify the navigational properties with different displacements and center of gravity. The result shows that the resistance can be reduced by moving COG aft. But at the same time the longitudinal stability is also reduced. Another test is in regular wave to collect heaving and pitching data. The result shows that the trimaran planing boats’ motion response is similar to common high-speed vessels’ performance. The trimaran planing boats also have the largest motion amplitude when they meet the specific encounter frequency. The encounter frequency which causes the maximum response amplitude is greater at a higher velocity.

The dynamic behavior of a trimaran vessels is investigated by Vakilabadi et al.
The body of the trimaran is composed of a center hull with a quite slender wave piercing bow profile (a length to breadth ratio of 12.96) and two outriggers with Wigley mathematical body forms. Several seakeeping tests are conducted on the model of the trimaran vessel in a towing tank in order to study its heave and pitch motions at different Froude numbers of 0.2, 0.37, and 0.51. Regular wave with wave length changing from 0.6 m to 2.4 m are generated in towing tank in increments of 0.3 m. The amplitude of the waves is 35 mm. A resonance peak is detected on the curve of the heave response amplitude operator (RAO) against non-dimensional wave length at around Fn 1.0. Increasing the Froude number leads to rapidly descending post-resonance-peak regions of the heave RAO versus on-dimensional wave frequency diagrams. Wave length changes doesn’t affect the motion too much when Froude number is less than 0.8. Also, for the values greater than 2.5 to 2.8 of the non-dimensional wave frequency, the pitch RAO does not experience any significant changes as a result of variations in the Froude number or wave amplitude.

Lin et al. (2017) compared two computational methods to analyze the seakeeping performances of wave piercing high-speed catamaran (CAT-1). One of the method is potential flow. The other one is viscous flow RANS (Reynolds Averaged Navier-Stokes) method. The accuracy of two methods have been compared. The results from the two methods are also compared against experimental results to analyze the numerical errors and the differences of two methods. The numerical simulation results show a good agreement with the experimental results. In conclusion, the viscous flow RANS method can predict the seakeeping performance better.
For the CFD simulation, the potential flow methods and viscous flow RANS method are applied to predict the seakeeping characteristics of CAT-I in regular head waves by using the commercial CFD software “HYDROSTAR” and “STAR-CCM+”. For HYDROSTAR, there is a good correlation between numerical and experimental results except the one around the resonance region of pitch motions. The experimental measurements are lower than numerical predictions. For STAR-CCM+, there is satisfactory agreement among all results. The comparison between numerical and experimental results shows a good agreement.

In the meshing part, the automatic meshing facility in STAR CCM+ has been used. The boundary conditions have also been mentioned that the top, bottom and side boundaries can be set not only as velocity inlet but also as slip-wall faces, pressure outlet or symmetry plane. If the virtual towing tank is big enough which means the boundaries is far enough away from the hull faces, the boundary conditions for these three boundaries would not affect the final results.

Researchers found that hydrofoil can produce significant lift to boats lifting it out of water. Since then researchers have focused on the development of hydrofoil and try to use it on varies hull forms. Kihara et al. (1995) introduced the fully submerged hydrofoil catamaran called “Mitsubishi Super-Shuttle 400”. This catamaran can reach a speed of 40 knots with diesel engine. The capacity is 341 passengers which is higher than other similar forms at that time (around 280 passengers). The propulsion system has less initial and maintenance costs than gas turbine engines.
In this paper, the author mentions the concept of design which also point out the advantages of hydrofoil assisted crafts. The hydrofoil assisted catamaran has smaller resistance when the hull is in take-off condition and a soft wave impact. Also the catamaran has better performance than mono-hull with hydrofoil. This boat is the first craft of this type for commercial use.

Hoppe (1995) carried out a optimization method for hydrofoil supported planing catamarans. The early designs used the systematical model test to optimize the hull form. A theoretical method is presented as a computer program in his paper. This method has more accuracy. Hoppe’s computer program determined the hydrofoil strength and hull stability. Lift and drag force can also be determined by dead rise angle, wetted length and beam. The theoretical result’s validation is verified by model test data and prototype trial data comparison. The theoretical design predicts accurately the influence of the main parameters on the hydrodynamic performance. The results have been presented for a 22m hydrofoil supported catamaran.

Hoppe (2001) discussed the applications of hydrofoil supported catamarans.
Hydrofoil was firstly used on a police boat test model to reduce resistance in which a 40% reduction was achieved. This result was regarded to be suspicious and hence a larger model was created for verification. The model test again demonstrated a 40% resistance reduction at cruising speed compared with traditional hull form of similar size. The author mentions several hydrofoil supported catamarans which are designed and built in the UK, France and the US. The performance of these crafts were discussed. The results prove that the combination of catamaran and hydrofoil can improves the hull efficiency. Not only the resistance but also the stability, seakeeping performance, lifting capacity and floating attitude were also improved.

Reichel and Bednarek (2007) presented the investigation of hydrofoil resistance with experiments. The model test was conducted in large towing tank of Ship Hydromechanics Division. The model is 4 m length with two foils. The results are matched with mathematical model and validated against computer simulation results.

Migeotte and Kornev (2004) discussed the resistance and ride control performance for several kinds of hydrofoil assisted multi-hull forms. The most common hydrofoil assisted multi-hulls can be classified as hydrofoil catamaran in which hulls were totally or partially lifted out of water, hydrofoil assisted trimaran and SWATH type hydrofoil assisted catamaran. The history of hydrofoil assisted multi-hulls development from 1980 to 2003 and the performance of different hydrofoil systems has been introduced in this paper. The achievement of hydrofoil now days are also mentioned. It has been applied to 50 m length trimaran which is built by North West Bay Ships and a craft
built by Almaz which achieved well over 50 knots. Due to these improved developments, the hydrofoil can lead to reduction in resistance and better seakeeping and ride control performance.

Prastowo et al. (2016) optimized the type of hydrofoil for a 25m length hydrofoil supported catamaran to obtain the maximum performance by CFD method. The research focus on increasing the efficiency of hydrofoil. Several combination of foil and catamaran hull forms were presented in this paper. The angle of attack, lift force and drag force of foils were also considered.
Chapter 3
CFD and Seakeeping Theory

3.1 Governing Equations

Numerical simulations in this thesis are carried out based on a numerically simulated wave tank. The nonlinear free surface flow is treated as two-phase flow: water and air. The surface is the interface between air and water. Density of water and air are assumed to be constant which means both water and air are incompressible.

Starting with the continuity equation:

\[ \nabla \cdot \vec{V} = 0 \]  \hspace{1cm} (1)

where \( \vec{V} \) is the velocity vector, and because of the fluid is assumed to be incompressible, the conservation equation of water (or air) in the two-phase flow of waves can be written in the form of volume fraction:

\[ \frac{\partial \alpha_w}{\partial t} + \vec{V} \cdot \nabla \alpha_w = 0 \]  \hspace{1cm} (2)

In every control volumes, the summation of volume fraction of water and air should be one:

\[ \alpha_w + \alpha_a = 1 \]  \hspace{1cm} (3)
where $\alpha$ is volume fraction, the subscripts $a$ and $w$ means air and water.

How using the conservation equation of momentum for the two-phase flow we obtain:

$$\frac{\partial}{\partial t}(\rho \mathbf{V}) + \nabla \cdot (\rho \mathbf{V} \mathbf{V}) = -\nabla p + \nabla \cdot \left[ \mu \left( \nabla \mathbf{V} + \nabla \mathbf{V}^T \right) \right] + \rho \mathbf{g}$$  \hspace{1cm} (4)

Where $\rho$ is the density, $\mu$ is the viscosity, $g$ is the acceleration due to gravity, $p$ is pressure. The density of the two-phase flow $\rho$ is given by,

$$\rho = \alpha_w \rho_w + \alpha_a \rho_a$$  \hspace{1cm} (5)

### 3.2 Wave Generation

According to the linear wave theory in water of infinite depth, the free surface elevation of regular wave would be,

$$\eta = \zeta \cos(kx - \omega t)$$  \hspace{1cm} (6)

Consequently, the velocity field of regular wave flow will be,

$$u = \omega \zeta e^{kz} \cos(kx - \omega t)$$
$$v = 0$$
$$w = \omega \zeta e^{kz} \sin(kx - \omega t)$$  \hspace{1cm} (7)

Where $\zeta$ is wave amplitude, $k$ is wave number, $\omega$ is wave frequency.
3.3 Ship motions

Generally, ship motion is analyzed in the global coordinate system. There are 6 degrees of freedom of a rigid body floating in sea waves. This thesis only focus on heave and pitch motions.

Heave is the linear vertical motion of a floating body in z-direction which can be shown as

\[ a\ddot{z} + b\dot{z} + cz = F_0 \cos(\omega_e t) \tag{8} \]

\[ z = z_a \cos(\omega_e t) \tag{9} \]

Where \( a \) is virtual mass, \( \ddot{z} \) is vertical acceleration, \( b \) is damping constant, \( \dot{z} \) is velocity, \( c \) is restoring force constant, \( z \) is translation of center of gravity, \( F \) is amplitude of exciting force, \( \omega_e \) is encountering frequency, \( z_a \) is the dynamic amplitude of heave motion.

For pitching, it can be shown as

\[ a\frac{d^2\theta}{dt^2} + b\frac{d\theta}{dt} + c\theta = M \cos(\omega_e t) \tag{10} \]

\[ \theta = \theta_a \cos(\omega_e t) + \epsilon \tag{11} \]

where \( a \) is virtual mass moment of inertia, \( \theta \) is pitching angle, \( b \) is damping moment coefficient, \( c \) is restoring moment coefficient, \( \epsilon \) is phase angle.

The strip theory is usually used in these ship motion studies. Strip theory assumes that the hydrodynamic properties of a vessel such as the added mass,
damping and stiffness may be calculated by integrating a series of two-dimensional strips over the length of the vessel. Base on the strip theory, the equation of heave and pitch motion can be modified to:

Equation of heave motion:

\[ a_z \ddot{z} + b_z \dot{z} + c_z z = F_o \cos(\omega_o t) \]  

(12)

\( a_z \ddot{z} \) is inertial force in heave. The virtual mass, \( a_z \), can be presented as:

\[ a_z = \int_{-L/2}^{L/2} a_x \, dx \]  

(13)

\[ a_x = C \frac{\rho \pi B_x^2}{8} \]  

(14)

(Bx is the waterline breadth at that section and C can be obtained from Lewis form section curves)

\( b_z \dot{z} \) is damping force in heave. The damping constant, \( b_z \), can be presented as:

\[ b_z = \int_{-L/2}^{L/2} b_n \, dx \]  

(15)

\[ b_n = \frac{\rho g^2 A^2}{\omega^3} \]  

(16)

\[ A = \frac{z_a}{z_a} \]  

(17)
\[
\frac{\xi_a}{z_a} = \frac{\text{Amplitude of the radiated wave}}{\text{Amplitude of the heaving motion}}
\]  
(18)

\(b_n\) is the sectional damping moment coefficient.

\(c_z\) is the restoring force in heave. The restoring constant, \(c_z\), can be presented as \(c_z = \rho g A_w\).

Or

\[
c_z = \rho g \int_{-L/2}^{L/2} 2y(x)dx
\]  
(19)

Equation of pitch motion:

\[
a_\theta \ddot{\theta} + b_\theta \dot{\theta} + c_\theta \theta = M \cos \omega t
\]  
(20)

Where, \(a_\theta = I_{yy}'\)

\(I_{yy}'\) is virtual mass moment of inertia in pitch.

\(\ddot{\theta}\) is angular acceleration in pitch.

\(\dot{\theta}\) is angular velocity in pitch.

\(b_\theta\) is damping moment coefficient.

\(\theta\) is angular displacement in pitch.

\(c_\theta\) is restoring moment coefficient.
Linear strip theory assumes the vessel’s motions are linear and harmonic, in which case the vessel responses in heave and pitch, for a given wave frequency, will be proportional to the wave amplitude.

Other integral assumptions behind linear strip theory, indicated by Lloyd (1989), are:

- The fluid in inviscid. The viscous damping is ignored, along with demi-hull interference.

- The ship is slender (the length is much greater than the beam or the draft, and the beam is much less than the wave length).

- The hull is rigid so that no flexure of the structure occurs.

- The speed is moderate so there is no appreciable planing lift.

- The motions are small (or at least linear with wave amplitude).

- The ship hull sections are wall-sided.

- The water depth is much greater than the wave length so that deep water approximations may be applied. The presence of the hull has no effect on the waves (Froude-Krylov hypothesis).

### 3.4 Theory of Numerical Simulation

Seakeeping analysis has progressed from the linear frequency-domain 2D strip method to the nonlinear time domain 3D panel method. The numerical schemes of STAR CCM+ are presented by Ferziger and Peric (2003).
Kim (2011) validate the CFD method which can be used as a seakeeping tool for ship design considering fully nonlinear three-dimensional slamming and green water on deck. The structural loads on a large container carrier were calculated from the CFD analysis and validated with segmented model test measurements. Figures 4 and 5 are representative of the mesh configuration.

![Mesh Scene](image1)

*Figure 4: Mesh Scene (Kim, 2011)*

![Wave Scene](image2)

*Figure 5: Wave Scene (Kim, 2011)*

The unsteady ship viscous flows are assumed to be governed by the RANS
equations. The equations for continuity, momentum and turbulence properties are generally given in integral forms and transformed into a system of algebraic equations by the discretization schemes such as finite-volume method (FVM).

A turbulence model such as eddy-viscosity model ($k-\varepsilon$ or $k-\omega$ model) may be used for ship viscous flows. A specific numerical algorithm such as SIMPLE algorithm (Patankar, 1980) may be used for pressure calculation by solving the pressure correction equation derived from the continuity equation.

For the modeling of violent free surface flows with possible air trapping, a two-phase fluid model is to be considered. In order to effectively capture the interface between the water and air, the volume of fluid (VOF) method may be used by solving an additional equation for the volume fraction of air.

The equations of ship motion in waves are derived by considering all the inertial and hydrodynamic forces and moments acting on the rigid body. The 6-DOF equations of motions are to be solved by time integration methods such as a second-order predictor-corrector scheme. Since the surge-sway-yaw motions in horizontal plane have no hydrostatic restoring mechanism, a numerical course-keeping model would be required to avoid unrealistic drift motions during the motion simulation. Otherwise, only the heavy-roll-pitch motions in vertical plane may be considered as a simplified approach.

For the accuracy in conducting the simulation for incident waves, the 5th-order Stokes waves are recommended by STAR CCM+ tutorial. The incident waves are required as an initial condition as well as an inlet boundary condition. The incident waves, given at the inlet boundary, propagate inside the computational domain. At the outlet boundary, an artificial damping beach or
specific outlet boundary condition is to be considered.

The computational domain size and artificial damping beach zone is to be large enough so as not to affect the flows near the ship. During the CFD simulation, due consideration is to be paid to the numerical damping, which primarily depends on mesh size and time step. The numerical damping can be reduced by providing a fair resolution of mesh size and time step (Kim, 2011). In this study, a minimum of 40 cells per wavelength is used.

The aim of simulation is to obtain the data of pitch and heave motion. The result will be shown as a curve of \( \frac{\theta_a L}{\zeta} \) and \( \frac{z_a}{\zeta} \) versus \( \frac{\lambda}{L} \). ( \( \theta_a \) is the amplitude of pitch, \( L \) is length of model, \( z_a \) is the amplitude of heave, \( \zeta \) is wave height.)
Chapter 4
CFD and Model Development

4.1 Description of Model

The model being used in simulation in this thesis has been generated with Rhinoceros 3D. The boat is set as a 6 degree of freedom (6-DOF) body. Since this thesis only focus on the motion in head sea condition, Z-axis motion and Y-axis rotation is activated (other motion and rotation are locked) and half boat is considered. The original point of the is located at stern, on the cross line of water plane and center plane. The weight is set to 167580 kg as half boat of full scaled model. Center of mass is set at (16.2, 0, -0.65) m. Mass moment of inertia is \( (1, 1.512 \times 10^7, 1) \) kg\(\cdot\)m\(^2\). Hydrofoil NACA 6409 is applied to HAST. The parameter of hydrofoil and HAST is listed in Table 2.
Table 2: Parameters of HAST & Hydrofoil

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall Length (m)</td>
<td>37.4</td>
</tr>
<tr>
<td>Beam (m)</td>
<td>7.6</td>
</tr>
<tr>
<td>Draft (m)</td>
<td>3.05</td>
</tr>
<tr>
<td>Displacement (t)</td>
<td>335</td>
</tr>
<tr>
<td>Cruise speed (knot)</td>
<td>44</td>
</tr>
<tr>
<td>Hydrofoil Type</td>
<td>NACA 6409</td>
</tr>
<tr>
<td>Chord Width (m)</td>
<td>1.5</td>
</tr>
<tr>
<td>Attack Angle (fl)</td>
<td>0</td>
</tr>
</tbody>
</table>

1. Domain and Boundary Conditions

To simulate the fluid field around the hull, a computational domain was first established. Due to the symmetry of the geometry and simulation time, only half of the hull was modeled. The domain size has a significant impact on the accuracy and computational cost. In current study, considering both boundary influence and computational time, an appropriate size of domain was adopted. The dimensions of the computational domain around the hull are given in Table 3. The boundary conditions have been specified. For the inlet, the velocity (both for water and air) is specified by the velocity of hull. At the outlet boundary, hydrostatic pressure is applied. Symmetry condition is used at the central plane of the hull. The hull body is considered as a rigid body and
non-slip condition is imposed on the hull surface. Free slip condition is used for the other boundaries.

**Table 3: Limits of Domain**

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>X (longitudinal)</td>
<td>-80 m</td>
<td>80 m</td>
</tr>
<tr>
<td>Y (beam)</td>
<td>0 m</td>
<td>50 m</td>
</tr>
<tr>
<td>Z (height)</td>
<td>-50 m</td>
<td>50 m</td>
</tr>
</tbody>
</table>

2. Mesh Generation

The domain is divided into two regions. Due to the complication of geometrical characteristics of the hull, a kind of trimmed cell mesh is used in meshing structure and hulls. The resolution of the mesh around the hull has evident influence on the computational accuracy. In this thesis, the boundary layer mesh was refined with prism elements. A mesh-dependent analysis was conducted for the non-dimensional wall distance of the first cell center, \( y^+ \), and the surface mesh size of the HAST to ensure the selected mesh produces the accurate results.

The domain volume is divided into small cells to generate the mesh. The largest cells on the hull are approximately \( \Delta (X, Y, Z) = 0.36 \text{ m} \) in size. In areas with large curvature and small features, cells as small as \( \Delta (X, Y, Z) = 0.003 \text{ m} \) were used to ensure the flow features have a good resolution. Extra cells were added to the hydrofoils to ensure a good resolution in the boundary layer. The first cell near the wall was set to have a size of about 0.00064 m, such that non-dimensional distance \( (y^+) \) to the wall was approximately 50 m. Cells around the air-water interface were refined to have a size of 0.006 m in
Z-direction. Figure 6 shows the mesh resolutions. The cells size of virtual towing tank and refined parts are suggested by STAR CCM+ tutorial. The cells size in the tutorial is represented by percentage of model length. For example the largest cells are set to 1% of model length and the smallest cells are set to 0.008% of model length (STAR CCM+ Tutorial Guide)

![Figure 6: Mesh Scene](image)

3. Specific Mesh Setting

The specific setting of refined mesh is shown in this part. The simulation is based on continuity equation, so that the domains which have more flow features require higher intensity of cells to improve the resolution.

The bows and sterns for both main body and side hull are refined. These refined bodies are built as cones instead of blocks to reduce the cells number and speed up the simulation.
The cells size of volume mesh in the highlighted cones shown in Figure 7 is 0.05 m (The cell size is described by side length because trimmed cells are applied in mesh generation.). Because the geometry surface is covered in these volume control domains, prism layer need to be generated. The prism layers number is set to 5 and total thickness is 0.003 m. Target surface size, surface curvature and other setting are followed the default setting.

The hydrofoil refinements are shown in Figure 8. There are 5 prism layers and total thickness is 0.003 m. The calculation of flows around foils need better accuracy so it requires smaller cells but large number of cells prolong the simulation time. Considering time factor the cell size was set to 0.03 m.
Figure 8: Hydrofoil Refinements

Figure 9: Free Surface Refinements

Figure 9 shows the free surface refinements. Two domains are generated for free surface. The cell size for these two blocks are anisotropy. In X and Y direction, the size is set to 0.7 m. In Z direction, the thin layer is set to 0.18 m and the thick one is set to 0.3 m. The reason of two blocks being built for free surface is too much difference of cell size between domains may produce error or even shut down the simulation. Figure 10 is the refinement of Kelvin wake. The horizontal size is 0.5 m and Z direction size is 0.3 m. The angle between the highlight area side and X-axis s 19.26° which is the same as Kelvin Ship Wake Angle.
The waterline refinement which is shown in Figure 11 is in order to increase the cells number between main hull and side hull. Using the general wave refinement produced some vortex and led to error in this area. The horizontal cell size is 0.7 m and vertical size is 0.008 m.

The whole domain is moving and rotating with ship when the ship exhibits heave and pitch motion if using the normal mesh setting. According to several tests, the results are not reasonable when trim angle reaches 1.5° or higher. To solve this problem and ensure the accuracy of simulation, overset mesh is
applied. Figure 12 and Table 4 shows the overset parameters.

**Figure 12: Overset Tank**

**Table 4: Overset Domain**

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>X</td>
<td>-8 m</td>
<td>39 m</td>
</tr>
<tr>
<td>Y</td>
<td>0 m</td>
<td>8 m</td>
</tr>
<tr>
<td>Z</td>
<td>-7 m</td>
<td>5 m</td>
</tr>
</tbody>
</table>

The virtual towing tank is divided into two parts, background tank and overset tank.

**Figure 13: Overset Tank - Inlet & Outlet Refinement**
The inlet and outlet parts are refined as shown in Figure 13. The inlet and outlet refinement block are anisotropy. The cell length is 1 m in horizontal direction and 0.16 m in vertical direction. The inlet and outlet blocks are connection of background tank and overset tank. The overset tank is rotating by Y-axis so that more intensity of cells is required in Z-direction.

### 4.2 Simulation Setting Up

Several physical models are activated to simulate the forces acting on the hull. The simulation models the behavior of two fluids (air and water) within the same continuum, and uses the Volume of Fluid model to do so.

As there are two fluids in different phases, the Eulerian Multiphase model is activated, and the effect of gravity acting on both is included using the Gravity model. The effect of turbulence on the fluid is modeled using the default K-Epsilon turbulence model.

A default Physics Continuum was created when the mesh was imported. All necessary models are defined using this continuum. For the physics continuum, the setting is shown in Table 5.
1. Setting the Material Properties

Material properties for the Eulerian phases is set in this part.

The materials corresponding to each of the mixture phases under the Eulerian Phases node are defined. The model of water is selected as liquid and air is selected as gas. Constant density is applied for the models.

In both instances, the default material appropriate to each phase (water for the liquid and air for the gas) is applied. The material selections is shown in the Figure 14.
2. Selecting the 6-DOF Motion Model

The 6-DOF motion solver is activated by creating the relevant DFBI motion object and assigning it to the fluid region. A model including DFBI Rotation and Translation is applied.

The motion is set to DFBI Rotation and Translation is set in Region node. Figure 15 shows the 6-DOF model properties.
3. Setting the Body Initial Coordinate System

The 6-DOF Solver requires the initial orientation of the body relative to the laboratory coordinate system. A local coordinate system in which the X-axis is aligned with the forward direction of the body and the Z-axis in the vertical direction is defined.

A new Cartesian Coordinate System is created as the local coordinate system. The directions of axis are set as Figure 16.
4. Creating the 6-DOF Body

The surface of the floating body and its properties is set in this part. To define the surface and its properties, first create a 6-DOF body to which the boundary representing the boat hull is assigned.

Figure 16: Coordinate System Setting
The body mass is set to 167580 kg (Half of the total weight). Release time is set to 1.0 s. The motion option is selected as free motion.

In this simulation, the body is only allowed to rotate about the Y-axis to simulate motion in the head waves. The mass moment of inertia about the Y-axis is required, which is set to \([1.0, 1.512 \times 10^7, 1.0]\) kg\(\cdot\)m\(^2\). Center of mass is located at \([16.3, 0.0, -0.65]\) m. Center of mass is also activated in Moment of Inertia node.

![Figure 17: Ship Body Property](image17)

![Figure 18: Mass Moment of Inertia](image18)

In this simulation focus is on heaving and pitch motion, Z motion and Y rotation is activated in free motion node. Other motions are locked.

5. Setting Initial Conditions

The initial conditions are set based on field functions which are associated with the defined 5th order VOF wave.
The initial condition setting is shown in Table 6.

**Table 6: Initial Condition Setting**

<table>
<thead>
<tr>
<th>Node</th>
<th>Property</th>
<th>Setting</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volume Fraction</td>
<td>Method</td>
<td>Composite</td>
</tr>
<tr>
<td>Composite: Water</td>
<td>Method</td>
<td>Field Function</td>
</tr>
<tr>
<td></td>
<td>Scalar Function</td>
<td>Volume Fraction of Heavy Fluid of Head Wave</td>
</tr>
<tr>
<td>Composite: Air</td>
<td>Method</td>
<td>Field Function</td>
</tr>
<tr>
<td></td>
<td>Scalar Function</td>
<td>Volume Fraction of Light Fluid of Head Wave</td>
</tr>
<tr>
<td>Velocity</td>
<td>Method</td>
<td>Field Function</td>
</tr>
<tr>
<td></td>
<td>Vector Function</td>
<td>Velocity of Head Wave</td>
</tr>
<tr>
<td>Pressure</td>
<td>Method</td>
<td>Field Function</td>
</tr>
<tr>
<td></td>
<td>Scalar Function</td>
<td>Hydrostatic Pressure of Head Wave</td>
</tr>
</tbody>
</table>

The volume fraction consists air component and water component. Field functions are applied to represent water and air. The scalar function is selected as volume fraction of heavy/light fluid of head wave to associate with the defined wave. The velocity and pressure functions are also set to associate with defined wave.
6. Boundary condition setting.

There are five kinds of boundaries in this simulation.

1) Hull of the boat is set to no-slip wall boundary.

2) The side face of towing tank and the Z-X plane at Y = 0 are set as symmetry boundary.

3) The face of towing tank which is named as “Outlet” is set as pressure outlet.

4) The other faces of the towing tank are set as velocity inlet.

5) Overset boundaries on faces named as “Overset”.

The defined VOF wave is a head wave which enters the simulation through the inlet boundary. The field functions associated with the defined wave are used to define the oncoming wave at the inlet boundaries. Table 7 shows the inlet boundaries’ setting.
As the same as inlet boundaries, the pressure outlet boundary is also associated with the defined wave. Table 8 represents the specific setting of pressure outlet boundary.
Table 8: Outlet Boundaries Setting

<table>
<thead>
<tr>
<th>Node</th>
<th>Property</th>
<th>Setting</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td><strong>Physics Values</strong></td>
<td></td>
</tr>
<tr>
<td>Pressure</td>
<td>Method</td>
<td>Field Function</td>
</tr>
<tr>
<td></td>
<td>Scalar Function</td>
<td>Hydrostatic Pressure of Head Wave</td>
</tr>
<tr>
<td>Volume Fraction</td>
<td>Method</td>
<td>Composite</td>
</tr>
<tr>
<td>Composite: Water</td>
<td>Method</td>
<td>Field Function</td>
</tr>
<tr>
<td></td>
<td>Scalar Function</td>
<td>Volume Fraction of Heavy Fluid of Head Wave</td>
</tr>
<tr>
<td>Composite: Air</td>
<td>Method</td>
<td>Field Function</td>
</tr>
<tr>
<td></td>
<td>Scalar Function</td>
<td>Volume Fraction of Light Fluid of Head Wave</td>
</tr>
</tbody>
</table>

7. Other parameters setting

The simulation is using the finite element method so that the time step, maximum inner iteration number need to be configured. The time step is set to 0.01 s and the maximum number of inner iterations is 5. The maximum physical time is inactivated because the time for each working condition achieving to steady status are different.

These setting are aim to not only guarantee the accuracy of results but also shorten the simulation time.
Chapter 5
Results Analysis and Conclusion

5.1 Descriptions of working conditions

6 series of simulations are conducted in this thesis. There are 3 series of simulations carried out for HAST which are shown in Table 9.

<table>
<thead>
<tr>
<th>Series</th>
<th>Wave Height (m)</th>
<th>Wave Length (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.875</td>
<td>61.64</td>
</tr>
<tr>
<td>2</td>
<td>1.875</td>
<td>37.4</td>
</tr>
<tr>
<td>3</td>
<td>Calm Water</td>
<td></td>
</tr>
</tbody>
</table>

3 series of simulations are done for a traditional trimaran hull form which has same displacement (335 t) and similar length (40 m) for comparative analysis. The trimaran model is scaled version of USS Independence (LCS-2), Thompson (2010). Table 10 shows the working condition series of trimaran.
Table 10: Working Conditions of Trimaran

<table>
<thead>
<tr>
<th>Series</th>
<th>Wave Height (m)</th>
<th>Wave Length (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.875</td>
<td>61.64</td>
</tr>
<tr>
<td>2</td>
<td>1.875</td>
<td>37.4</td>
</tr>
<tr>
<td>3</td>
<td>Calm Water</td>
<td></td>
</tr>
</tbody>
</table>

Each of the simulation series has been done with different Froude volume number from 0.2 to 3.1 (Froude number from 0.1 to 1.3). To describe the working conditions, simulation series 2 of HAST are discussed in this part.

Working condition 1

The setting of working condition 1 is shown in Table 11. The length of HAST is 37.4m. The displacement is 335 t because the simulation is aimed to test the seakeeping performance when HAST is floating on water surface in head sea.

Table 11: Working Condition 1

<table>
<thead>
<tr>
<th>Fn/Fn&lt;sub&gt;e&lt;/sub&gt;</th>
<th>0.1/0.24</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wave Height</td>
<td>1.875</td>
</tr>
<tr>
<td>Wave Length</td>
<td>37.4</td>
</tr>
<tr>
<td>Wave Period</td>
<td>1.29</td>
</tr>
<tr>
<td>Head Sea</td>
<td></td>
</tr>
</tbody>
</table>

The sea states are shown as Table 12.
<table>
<thead>
<tr>
<th>WMO Sea State Code</th>
<th>Wave height</th>
<th>Characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0 m</td>
<td>Calm (glassy)</td>
</tr>
<tr>
<td>1</td>
<td>0 m to 0.1 m</td>
<td>Calm (rippled)</td>
</tr>
<tr>
<td>2</td>
<td>0.1 m to 0.5 m</td>
<td>Smooth (wavelets)</td>
</tr>
<tr>
<td>3</td>
<td>0.5 m to 1.25 m</td>
<td>Slight</td>
</tr>
<tr>
<td>4</td>
<td>1.25 m to 2.5 m</td>
<td>Moderate</td>
</tr>
<tr>
<td>5</td>
<td>2.5 m to 4 m</td>
<td>Rough</td>
</tr>
<tr>
<td>6</td>
<td>4 m to 6 m</td>
<td>Very rough</td>
</tr>
<tr>
<td>7</td>
<td>6 m to 9 m</td>
<td>High</td>
</tr>
<tr>
<td>8</td>
<td>9 m to 14 m</td>
<td>Very high</td>
</tr>
<tr>
<td>9</td>
<td>Over 14 m</td>
<td>Phenomenal</td>
</tr>
</tbody>
</table>

Sea state 4 is usually chosen as a typical sea state in seakeeping performance test. The average wave height in sea state 4, 1.875 m, is chosen as the typical wave height for the simulations. The working conditions with a 37.4 m length wave and a 61.64 m length wave are investigated. The 37.4 m length wave is chosen because it is equal to model length. The period is 4.893 s which is reasonable because the typical wind wave period is 0.2 - 9 s (Exploring Florida). The 61.64 m length wave is chosen as a lower frequency wave. The wave frequency is 1 rad/s and wave period is 6.283 s which is also between 0.2 s and 9 s.
In Table 13, the amplitude of oscillation and mean value of HAST’s motions are shown. These data are collected from 10 s to 37 s. The release time of simulation is 5 s which means heave and pitch motions are restricted before 5 s. The simulation were carried out in STAR CCM+ using finite element method. Sudden changes in motion, resistance and some other parameters will reduce the accuracy or even, interrupt the simulation. Releasing all of motions and current flows at 5 s gives the simulation enough buffer time which can avoid the sudden changes being generated. After 10 s, the results are stable.

**Table 13: Heave & Pitch in Head Sea (37.4 m, $F_{n_{z}} = 0.24$)**

<table>
<thead>
<tr>
<th></th>
<th>Amplitude</th>
<th>Mean Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heave (m)</td>
<td>0.213</td>
<td>0.147</td>
</tr>
<tr>
<td>Pitch (deg)</td>
<td>1.555</td>
<td>-0.559</td>
</tr>
</tbody>
</table>

The motions result changes periodically with the period of 3.32 s. The encounter frequency is 1.62 rad/s which is calculated by the equation 21:

$$\omega_e = \omega_w - \frac{\omega_w^2 V}{g} \cos \mu$$  \hspace{1cm} (21)

$\mu$ is the wave approaching angle. $\mu$ has been chosen as 180° here because in working condition 1 to 5 is focus on head sea condition. It is to be noted that the positive value of heave means lift and negative value means sinkage. For pitch motion, the positive value means trim by bow and negative value means trim by stern.
Figure 19 is the VOA scenario in head sea testing. The percentage of water and air in the domain is shown. Blue is 100% of water and red is 100% of air. This figure shows the free water surface around the ship hull on the symmetry plane.

**Working condition 2**

Setting of working condition 2 is listed in Table 14.

<table>
<thead>
<tr>
<th></th>
<th>Working Condition 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\frac{F_{n}}{F_{n,\text{\textsuperscript{\textregistered}}}}$</td>
<td>0.4/0.96</td>
</tr>
<tr>
<td>Wave Height</td>
<td>1.875</td>
</tr>
<tr>
<td>Wave Length</td>
<td>37.4</td>
</tr>
<tr>
<td>Wave Period</td>
<td>1.29</td>
</tr>
<tr>
<td>Head Sea</td>
<td></td>
</tr>
</tbody>
</table>

The time of simulation in working condition 2 has been shorted because the
results get stable at 5 s. The data is collected from 5 s to 30 s. Motions and water flows are unlocked after 5 s so that the data is available. The mean value and oscillation amplitude are shown in Table 15.

**Table 15: Heave & Pitch in Head Sea (37.4 m, \( \text{Fn}_\varphi = 0.96 \))**

<table>
<thead>
<tr>
<th></th>
<th>Amplitude</th>
<th>Mean Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heave (m)</td>
<td>0.058</td>
<td>-0.135</td>
</tr>
<tr>
<td>Pitch (deg)</td>
<td>0.807</td>
<td>-0.252</td>
</tr>
</tbody>
</table>

The motions results change periodically with the period of 2.37 s. The encounter frequency is 0.439 rad/s.

The velocity is .746 m which is 4 times higher than it in working condition 1. The oscillation amplitude of heave is reduced by 72%. For pitch, the oscillation amplitude is reduced by 50%.

![Wave Scene in Head Sea (37.4 m, \( \text{Fn}_\varphi = 0.96 \))](image)

**Figure 20: Wave Scene in Head Sea (37.4 m, \( \text{Fn}_\varphi = 0.96 \))**
Figure 20 illustrates the wave pattern. In this figure the ship is partially submersed where the heave value is negative. It can be seen that the trim angle in Figure 21 is less than it is Figure 19.

![Figure 21: VOA Scene in Head Sea (37.4 m, Fn_\nu = 0.96)](image)

**Working condition 3**

**Table 16: Working Condition 3**

<table>
<thead>
<tr>
<th></th>
<th>0.7/1.67</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Fn/Fn_\nu</strong></td>
<td>0.7/1.67</td>
</tr>
<tr>
<td><strong>Wave Height</strong></td>
<td>1.875</td>
</tr>
<tr>
<td><strong>Wave Length</strong></td>
<td>37.4</td>
</tr>
<tr>
<td><strong>Wave Period</strong></td>
<td>1.29</td>
</tr>
<tr>
<td><strong>Head Sea</strong></td>
<td></td>
</tr>
</tbody>
</table>

Table 16 is the setting of working condition 3. The period of oscillation is 1.85 s which is due to the encounter frequency increasing.
The mean value and oscillation amplitude of motions are shown as Table 17.

**Table 17: Heave & Pitch in Head Sea (37.4 m, \( F_{n\sigma} = 1.67 \))**

<table>
<thead>
<tr>
<th></th>
<th>Amplitude</th>
<th>Mean Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heave (m)</td>
<td>0.087</td>
<td>1.055</td>
</tr>
<tr>
<td>Pitch (deg)</td>
<td>0.985</td>
<td>-5.116</td>
</tr>
</tbody>
</table>

The heave value is increased because the speed has increased and more lifting force are produced by hydrofoil so that the hull is lifted upward. The oscillation amplitude are increased by about 11% comparing with working condition 2.

The VOA scene is shown in Figure 22. The trim angle is obviously increased and the waves generated by boat is also greater. The resistance in this circumstance must have increased.

**Figure 22: VOA Scene in Head Sea (37.4 m, \( F_{n\sigma} = 1.67 \))**
Working condition 4

Table 18: Working Condition 4

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$\text{Fn/Fn}_{v}$</td>
<td>1.0/2.39</td>
</tr>
<tr>
<td>Wave Height</td>
<td>1.875</td>
</tr>
<tr>
<td>Wave Length</td>
<td>37.4</td>
</tr>
<tr>
<td>Wave Period</td>
<td>1.29</td>
</tr>
<tr>
<td>Head Sea</td>
<td></td>
</tr>
</tbody>
</table>

In working condition 4 and 5, the waterline point is set at $z = -1.2$ m. The main hull of HAST appears to go out of water. Simulating the process of lifting the hull out of water always cause errors. These errors can even lead to shut down the simulation. To solve this problem and shorten the duration of simulation, the water surface is set to be lower to assume the craft is hold at the position which part of hulls is out of water. This change has no effect to pitch angle result but the heave data need to be corrected. The heave monitor measures the sinkage with the original point at the initial waterline point. So that the heave results need to add another 1.2 m. The results shown here are after correction.

The response period of motion is 1.53 s. The encounter period is 3.550 rad/s which is calculated by the equation 21 before.

Comparing with previous working condition 3, the amplitude of heave motion is reduced by 51 %. The amplitude of pitch angle is reduced too. The reductions are due to the hull lifting out of water and the effect of wave is less
pronounced. The results are shown in Table 19.

<table>
<thead>
<tr>
<th>Heave &amp; Pitch in Head Sea (37.4 m, $F_{n,v} = 2.39$)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Amplitude</strong></td>
</tr>
<tr>
<td>Heave (m)</td>
</tr>
<tr>
<td>Pitch (deg)</td>
</tr>
</tbody>
</table>

Figure 23 is the VOA scenario in working condition 4. The hull is mostly supported by hydrofoil but still in partial contact with water surface.

**Working condition 5**

Table 20 is the parameters of working condition 5.
Table 20: Working Condition 5

<table>
<thead>
<tr>
<th></th>
<th>1.3/3.11</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \text{Fn}/\text{Fn}_\nu )</td>
<td></td>
</tr>
<tr>
<td>Wave Height</td>
<td>1.875</td>
</tr>
<tr>
<td>Wave Length</td>
<td>37.4</td>
</tr>
<tr>
<td>Wave Period</td>
<td>1.29</td>
</tr>
</tbody>
</table>

Head Sea

The oscillation period in working condition 5 is 1.08 s. The encounter frequency is 5.479 rad/s. The motions are shown in Table 21. The heave amplitude is slightly changed but pitch amplitude is higher.

Table 21: Heave & Pitch in Head Sea (37.4 m, \( \text{Fn}_\nu = 3.11 \))

<table>
<thead>
<tr>
<th></th>
<th>Amplitude</th>
<th>Mean Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heave (m)</td>
<td>0.046</td>
<td>2.969</td>
</tr>
<tr>
<td>Pitch (deg)</td>
<td>1.181</td>
<td>-2.807</td>
</tr>
</tbody>
</table>
The VOA scenario has been shown in Figure 24. The hull is totally out of water and the wave made by boat is extremely reduced.

5.2 Result Analysis

The comparison of the mean value of rising height among calm water, head sea with 37.4 m wave length and head sea with 61.64 m wave length is shown in Figure 25. Dimensionless quantity, \( \frac{z}{\zeta} \), is applied to represent the heave motion. \( z \) is hull’s heave height and \( \zeta \) is wave amplitude.
The hull appears to rise beyond Froude volume number 0.9. The sinkage happens at Froude volume number around 0.9. It has been found that it is common for hydrofoil assisted crafts have trim by bow when speed is low (Moomlan, 2005). Trimming by bow will lead the boat having sinkage. But this effect is minor when the speed is very low and gets larger with speed increasing. For HAST, the effects of trimming by bow is neutralized by the lifting force generated by hydrofoil and the hulls navigation attitude gets back to normal.

HAST starts rise out of water at around $F_{n_{V}} = 1.7$. This is the transition process from floating to planning. Comparing Figures 22 to 24 we can find out the sailing attitude changing. The process is almost same in calm water with 61.64 m wave length. When the Froude volume number reach to 3.1, the heave value get stable around 2.95 m because the HAST is planing on water surface.
and supported by hydrofoils. The designed draft of HAST is 3.05 m. The distance from the bottom of hydrofoil to the bottom of hull is 0.55 m. This implies that when rises up to 2.5 m means HAST the main hull is totally lifted out of water.

The heave performance comparison among calm water and two different wave lengths head sea conditions shows the same trend of heave value versus Fn$\nu$. The hull’s heaving response in head sea condition is 15% higher than it in calm water when Fn$\nu$ higher than 2.4.

The comparison of the mean value of trim angle is shown in Figure 26. The trim angle is also represent by dimensionless quantity, $\theta \cdot L / \kappa$. $\theta$ is the trim angle which is collected form the simulation process. $L$ is wave length.

The trend of trim angle shown in Figure 26 also proves the feature of hydrofoil supported craft that trimming by bow always occur in low Froude numbers. HAST starts to rise up and hold at the attitude of trimming by stern beyond F$n\nu = 0.96$. Craft start to hold at lower trimming angle when F$n\nu$ is greater than 1.67. The amplitude of heave and pitch also reach to a peak value here. This phenomenon will be further discussed with heave and pitch amplitude.

The tend of trim angle changing by speed in calm water and head sea condition are also similar. For head sea condition, the trim angle is 47% higher than it in calm water condition.
Figure 26: Pitch Response in Calm Water and in Head Sea of HAST

(Wave height = 1.875 m, Wave length = 37.4 m/61.64 m).

The comparison of heave and pitch motions oscillation amplitude are shown in Figure 27 and Figure 28 by curves of dimensionless quantity against Foude number.
Figure 27: Heave Oscillation Amplitude in Calm Water and in Head Sea of HAST (Wave height = 1.875 m, Wave length = 37.4 m/61.64 m)

Figure 28: Pitch Oscillation in Calm Water and in Head Sea of HAST (Wave height = 1.875 m, Wave length = 37.4 m/61.64 m)
These figures shows the amplitude of oscillation in waves is higher than it in calm water due to the wave effects. At \( F_{n_v} \) around 0.9, the amplitude of heave and pitch is extremely high because of sailing attitude is unstable. The low-speed trimming caused the downward force against with lift force lead to this result.

The amplitude of heave and pitch reach to a peak around \( F_{n_v} = 1.67 \). The trim angle of HAST at \( F_{n_v} = 1.67 \) is around 4.53 degree, trimming by stern. For the hydrofoil applied in this simulation, the lift force reaches to a peak at the angle reach to around 4 degree and reduces quickly with the trim angle increasing or reducing. This is also due to the craft is in transition phase which is breaking from displacement phase to planing phase. This results in the lift force of HAST changes quickly which leads to the high oscillation amplitude (Moolman, 2005).

The same working conditions are applied to a 40 m length trimaran with same displacement for comparative purposes against HAST.
The 40 m trimaran heave is much lower than HAST and remains stable after Froude volume number reaches a value of 2.3. The heave value holds at 0.05 m in calm water and 0.06 m in waves. The rise height in waves is 20% higher than it in calm water. Figure 30 shows the pitch of the 40 m trimaran. There is no obvious hump in these curves. The trim angle in waves is higher than it in calm water by 27%.
Figure 30: Pitch Response in Calm Water and Head Sea of Trimaran
(Wave height = 1.875 m, Wave length = 37.4 m/61.64 m).

Figure 31: Heave Oscillation Amplitude in Calm Water and Head Sea of Trimaran
(Wave height = 1.875 m, Wave length = 37.4 m/61.64 m)
Figure 31 shows the heave oscillation amplitude of the trimaran. The amplitude in 61.64 m length wave, a lower frequency wave, is much higher than it in 37.4 m wave and calm water. The same thing happens to pitch amplitude. In Figure 32, it can be found that the amplitude of 61.64 m wave length is obviously higher.

Figure 32: Pitch Oscillation Amplitude in Calm Water and Head Sea of Trimaran
(Wave height = 1.875 m, Wave length = 37.4 m/61.64 m)

5.3 Conclusion

Comparing the non-dimensional quantity in Figure 27 and Figure 31, it can be found that:
1. HAST has less heave amplitude than the traditional trimaran at $F_{nv} \geq 2.4$.
2. The heave amplitude of HAST has less increase in waves than in calm water at $F_{nv} \geq 2.4$.

3. Comparing Figure 28 and Figure, HAST has larger pitch amplitude in high speed than a traditional trimaran. The pitch amplitude of HAST is more than twice by trimaran.

4. Trimming by bow happens to HAST at low-speed. Due to this reason the motions in low-speed for HAST is not stable.

5. At $F_{nv}$ around 1.67 which HAST is in transition phase, the motions amplitude is higher than it in other speed.

To conclude a conclusion, HAST has better heave performance in waves at high-speed than the trimaran. As far as pitch response and low-speed performance are concerned, HAST exhibits worse performance than a traditional trimaran.

**5.4 Future Work**

1. The high pitch amplitude of HAST in high-speed need to be solved. Changing center of gravity, adding appendages and other suitable method can be applied to HAST.

2. HAST’s motions in low-speed which is not stable. Suitable solution and optimization can be applied to HAST.
3. More working conditions with different wave length and period should be done in order to obtain the pitch and heave RAO.

4. Motions in sea state 5 or higher should be investigate to evaluate the HAST’s safety in extremely rough sea.

5. Irregular wave conditions test can be done in future work.

6. Different wave direction angle test should be done in future work, especially the 90$^\circ$ wave direction angle condition. The roll response in different wave length and period need to be carried out to obtain the roll RAO.

7. Hydrofoil assisted catamarans are widely used in commercial applications. Consequently the seakeeping performance of HAST need to be compared with a similar dimension hydrofoil assisted catamaran.

8. The sensitivity of cell size should be investigate to find out if less or more mesh density will effect the results accuracy.

9. The structure of HAST is complicated erratically the connection of hydrofoil and main hulls. The structural strength would be a tough problem in building HAST. An appropriate material should be found out in the future.

10. The process and control system of deploying hydrofoils need to be design and programmed.

11. A scaled model need to be build for some experiments. The experimental data obtained from towing tank test can be used to compare with the CFD results to verify the simulation. Also the structures can be optimized by building the scaled model.
References


Yipeng Pan, Body-variable three-body water wing combined diving boat. Patent Number CN104787275(A), 2015


The AYRS Members, (1970): “Sailing Hydrofoils”.

Exploring Florida, http://fcit.usf.edu/


ITTC Quality System Manual - Recommended Procedures and Guidelines


STAR CCM+ Tutorial Guide.


Appendix

Figure 33: Lift Force Coefficient vs. Attack Angle of Hydrofoil (NACA 6409)  
(Airfoiltools.com)

Figure 34: Drag Force Coefficient vs. Attack Angle of Hydrofoil  
(NACA 6409) (Airfoiltools.com)
Figure 35: Lift Force and Drag Force Coefficient vs. Attack Angle of Hydrofoil (NACA 6409)(Airfoiltools.com)