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DEVELOPMENT OF A SEMI-SWATH CRAFT FOR MALAYSIAN WATERS

(PEMBANGUNAN KAPAL SEMI-SWATH UNTUK PERAIRAN MALAYSIA)

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ABSTRACT

Small Waterplane Area Twin Hull (SWATH) and Catamaran vessels are known to have more stable platform as compared to mono-hulls. A further advantage of SWATH as compared to Catamaran is its smaller waterplane area that provides better seakeeping qualities. However, the significant drawback of the SWATH vessel is when encountering head-sea at high forward speed. Due to its low stiffness, it has a tendency for large pitch motions. Consequently, this may lead to excessive trim or even deck wetness. This phenomenon will not only degrade the comfortability but also results in structural damage with greater safety risks. In this research a modified SWATH design is proposed. The proposed design concept represents a combination of Catamaran and SWATH vessel hull features that will lead to reduce in bow-diving but still maintains good seakeeping capabilities. This is then called the **Semi-SWATH vessel.** In addition, the full-design of this vessel has been equipped by fixed fore fins and controllable aft fins attached on each lower hull. In the development of controllable aft fins, the PID controller system was applied to obtain an optimal vessel's ride performance at speeds of 15 (medium) and 20 (high) knots.

In this research work, the seakeeping performance of Semi-SWATH vessel was evaluated using time-domain simulation approach. The effect of fin stabilizer on the bare hull performance is considered. The validity of numerical evaluation was then compared with model experiments carried out in the Towing Tank at Marine Technology Laboratory, UTM. It is shown that the Semi-SWATH vessel with controllable fin stabilizer can have significantly reduction by about 42.57% of heave motion and 48.80% of pitch motion.

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ABSTRAK

Adalah diketahui bahawa Small Waterplane Area Twin Hull (SWATH) dan Catamaran mempunyai pelantar yang lebih stabil dibandingkan dengan mono-hull. Kelebihan SWATH adalah kawasan ‘waterplane’ yang lebih kecil untuk menghasilkan kualiti pergerakan kapal yang lebih baik berbanding Catamaran. Walaubagaimanapun, kekurangan SWATH adalah apabila menempuh laut pada kelajuan yang tinggi. Berdasarkan pada tahap kekerasan kapal ini yang rendah, kapal ini cenderung megalami pergerakan picth yang besar. Ini akan mengakibatkan trim yang berlebihan atau kebasahan dek. Fenomena ini bukan sahaja mengurangkan keselesaan tetapi juga menyebabkan kerosakan stuktur dengan risiko keselamatan yang lebih tingi. Penyelidikan ini mencadangkan rekabentuk SWATH yang diubahsuai. Konsep rekabentuk yang dicadangkan menunjukkan kombinasi ciri-ciri badan kapal Catamaran dan SWATH yang akan mengurangkan ‘bow-diving’ tetapi masih mengekalkan kebolehan pergerakan yang baik. Ini dikenali sebagai Semi-SWATH. Tambahan pula rekabentuk lengkap bagi kapal ini dilengkapi dengan sirip tetap dibahagian depan dan sirip bolehkawal dibahagian belakang pada setiap badan kapal yang lebih rendah. Dalam membangunkan sirip bolehkawal dibahagian belakang kapal, sistem pengawal PID telah digunakan untuk mendapatkan keadaan perjalanan kapal yang optimum pada kelajuan 15 (sederhana) dan 20 (tinggi) knots.

Dalam kerje penyelidikan ini, pergerakan Semi-SWATH telah dinilai menggunakan kaedah simulasi domain-masa. Kesan pemantap sirip pada badan kapal yang kosong dipertimbangkan. Kesahan nialaian menggunanakan persamaan matematik dibandingkan dengan eksperimen model yang telah dilakukan dalam Tangki Tunda di Makmal Teknologi Marin, UTM. Ini menunjukkan Semi-SWATH dengan pemantap sirip bolehkawal mempunyai pengurangan sebanyak 42.57% bagi pergerakan heave dan 48.80% bagi pergerakan pitch.

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NOMENCLATURE

Vessel/ Environment Parameters

GM_L	:	Longitudinal metacentric height
GM_T	:	Transverse metacentric height
T_d	:	Deep draught
T_s	:	Shallow draught
Δ_d	:	Displacement at deep draught
Δ_s	:	Displacement at shallow draught
LCG	:	Longitudinal center of gravity
SWATH	:	Small Waterplane Area of Twin Hull
LOA	:	Length overall of ship
Cb	:	Block coefficient
C_m	:	Midship area coefficient
KG	:	Vertical height of centre of gravity from the Keel

PID Controller

PID	:	Proportional-Integral-Derivative
K_p,	:	Proportional gain
K_i,	:	Integral gain
K_d	:	Derivative gain
Y_{ref}	:	A desired response
T_c	:	Critical period of waveform oscillation
K_c	:	Critical gain
T_i	:	Integral time constant

T_d	:	Derivative time constant
POS	:	Percent overshoot
d	:	Amplitude of the relay output
a	:	The amplitude of the waveform oscillation
e	:	The error deviations
δ_c	:	Control variable
Plant	:	A system to be controlled
Controller	:	Provides the excitation for the plant; Designed to control the overall system behaviour
SISO	:	Single-input single-output
MIMO	:	multiple-input multiple-output

DC Motor

V_a	:	Motor Voltage [V]
L_a	:	Motor Inductance [H]
i_a	:	Motor Current [A]
R_a	:	Motor Resistance [Ω]
K_a	:	Back emf constant [mV/(rad/sec)]
ω	:	Motor shaft angular velocity [rad/sec]
θ	:	Angular displacement [rad]
T_m	:	Motor Torque [Nm]
K_m	:	Torque Constant [Nm/A]
T_L	:	Load Torque [Nm]
Jm	:	Motor Inertia [Nm.sec^2]

Co-ordinate Systems

$O^e x^e y^e z^e$	:	The earth fixed co-ordinate system
$O^* x^* y^* z^*$	:	The fixed ship system being located at the centre of gravity of the ship

Fin Stabilizer

ρ	:	Fluid density
A	:	Projected fin area

$C_{L\alpha}$	:	Lift coefficient of the fin
B	:	Body
W	:	Wing
K and k	:	Fin-hull interaction factors for a fixed fin and for an activating fin
s	:	Span
c	:	Chord
a	:	Hull radius
AR	:	Aspect ratio
X_{FP}	:	Distance from the ship forward perpendicular to the fin axis
R_n	:	Reynolds number
D	:	Distance between leading edge of fins
ω	:	Oscillation frequency
E_{FS}	:	Effect of free surface
$C_{Z\alpha}$	:	Lift coefficient of the fin attached to the hull
D	:	Maximum diameter of the hull
C_D	:	Drag coefficient
KC	:	Keulegan Carpenter
T	:	Encounter period
M_{ij}^{f}	:	Mass of fin
A_{ij}^{f}	:	Added mass of fin
t	:	The maximum thickness of the fin

Equations of Motion

m	:	Mass of body
I_x, I_y, I_z	:	Principal mass moments of inertia about the x, y and z axes respectively
u, v, w	:	Linear velocities along the respective x, y and z axes
p, q, r	:	Angular velocities along the respective x, y and z axes
F_x, F_y, F_z	:	Force acting in x, y and z direction respectively

Forces and Moments

p	:	Pressure acting on the wetted surface
ρ	:	Density of water
g	:	Gravitational acceleration
∇	:	Under water volume of vessel
ω	:	Frequency of excitation
ω_e	:	Frequency of encounter
n_j	:	Outward unit normal vector in the j^{th} mode of motion
ϕ	:	Time dependent velocity potential
ϕ_I	:	Incident wave potential
ϕ_D	:	Diffracted wave potential
ϕ_{Rj}	:	Generated wave potential due to motions of the body in the j^{th} direction
ϕ^R and ϕ^I	:	Velocity potentials
∇	:	Vector differential operator
q	:	Source point
$\frac{\partial}{\partial n_q}$	:	Normal derivative with respect to the source point q
$\vec{n}$	:	Outward unit normal vector
G	:	Two dimensional Green's function
p	:	Field point
ν	:	Real variable
PV	:	Denotes Principal Value of an integral
$\bar{q}$	:	Complex conjugate of q
u_a	:	Motion amplitude
ξ_a	:	Wave amplitude
$\vec{U}$	:	Forward speed
S_b	:	Wetted body surface
S_f	:	Free surface

Hydrodynamic Coefficients

m_j : Mass or mass moment of inertia of body in the j^{th} direction (j = 1,2,…,6)

a_{jj} : Hydrodynamic reaction in phase with acceleration (added mass) in the j^{th} direction (j = 1,2,…,6)

b_{jj} : Hydrodynamic reaction in phase with velocity (damping) in the j^{th} direction (j = 1,2,…,6)

c_{jj} : Hydrostatic stiffness of body in the j^{th} direction (j = 1,2,…,6)

Experiment

DAAS : Data Acquisition and Analyzing System

SCS : Signal Conditioning System

RAO : Response Amplitude Operators

ω_n : Natural frequency of ship

λ : Wavelength

Ls : Length of ship

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CHAPTER 1

INTRODUCTION

1.1 Background

The applications of twin-hull vessels particularly SWATH vessel and conventional Catamaran have widely designed regarding for purpose of providing better seakeeping quality than mono-hull vessels inherently.

Holloway and Davis (2003) and Kennell (1992) stated that inherent to the advantages of SWATH vessels, as compared to the conventional Catamaran is its smaller waterplane area that provided smaller wave excitation forces, lower amplitude motion associated with its lower accelerations responses and better seakeeping performances. Dubrovskiy and Lyakhoviyskiy (2001), Fang (1988) and Kennell (1992) mentioned that the SWATH vessels have larger natural period as twice as long the natural periods of roll, pitch, and heave of a mono-hull of comparable size.

Based on Dubrovskiy and Lyakhoviyskiy (2001) and Ozawa (1987) have presented the advantages of conventional Catamaran features compared to the SWATH vessels have shallower draft and lower cost of construction. Their larger waterplane areas as compared to the SWATH vessel has increased the stiffness as result as improve vessel's longitudinal stability.

Conversely, the particular drawbacks of SWATH vessel and conventional Catamaran geometrically cannot be neglected. It is shown that the SWATH vessel with its small waterplane area is tender in large pitch motion due to low stiffness resulted as increase in speed. Djatmiko (2004), and Dubrovskiy and Lyakhoviyskiy (2001), and Kennell (1992) stated that the low value of this parameter is linked to its insufficient values of longitudinal metacentric height (GM_L). Consequently, this may lead to pitch instabilities, which caused slamming, deck-wetness, excessive trim or even bow diving and degrade the passenger comfortability.

Having considered some extensive reviews of several obtainable advantages both SWATH and conventional Catamaran hull forms, an alternative hull form design is proposed to overcome and minimize their drawbacks. The proposed design concept represents a combination of conventional Catamaran and SWATH hull features. In addition, this new modified hull form configuration conceptually was emphasized on the variable draught operations i.e. shallow draught and deep draught. Then, this vessel is called "**Semi-SWATH vessel**."

Holloway (1998 and 2003) investigated that as the hybrid design hull form; the Semi-SWATH configurations generally offered two ways that make the most of Semi-SWATH vessel's benefits. First, its primary premise is to maintain a good seakeeping quality. Second, it is intended to prevent the bow diving phenomena at high-speed. It means the maturity of Semi-SWATH vessel is going to provide an improvement of conventional Catamaran and SWATH vessel drawbacks considerably.

Furthermore, the placement both of fixed bow fins and controllable stern fins on each lower hull of Semi-SWATH vessel will provide additional pitch restoring moment to improve not only the longitudinal stability but also reduce the vertical motion responses. Consequently, the serious inconveniences will degrade the vessel performance during sailing especially at high-speed head sea waves can be alleviated. Haywood, Duncan, Klaka, and Bennett (1995) stated that the seakeeping of the Semi-SWATH vessel is going to be better evidently.

The simulation program of Semi-SWATH vessel incorporated with fixed fore and controllable aft fins were developed to evaluate the seakeeping performance during operation at both medium speed (15 knots) and high-speed (20 knots). The mathematical model comprising of heave and pitch motions, which incorporated with the fins stabilizers on the simulation was presented in a simple block diagram using Matlab-SIMULINK. In this simulation, a conventional PID controller was developed and applied on the controllable aft fins. Segundo, et al (2000) developed simulation program using PID controller to alleviate vertical accelerations due to waves. The results of simulation had been validated by experiments in the towing tank confirm that by means of flaps and a T-foil, moved under control, vertical accelerations can be smoothed, with a significant improvement of passengers comfort. In addition, Caldeira, et al (1984), Ware, et al (1980a), (1980b), 1981, and 1987, and Chinn, et al (1994) applied conventional optimal PID controller design to improve the vertical motion response of marine vehicles.

In this PID controller method, some parameter of tuning controller will involve some chosen controller gain parameters of PID (K_p, K_i, *and* K_d are the proportional, integral, and derivative gains, respectively). Those parameters are obtained using method of Aström and Hagglund. Then, they will be considered to satisfy certain control specifications by minimizing the error after achieving steady state. This controller mode is applied by controlling the aft fin's angle of attack properly, the sailing style of Semi-SWATH vessel must be adjusted to be in even keel condition. The theoretical prediction results will be validated with the model experiments carried out in the Towing Tank of Marine Technology Laboratory, Universiti Teknologi Malaysia.

1.2 Research Objective

1. To evaluate the seakeeping performance of Semi-SWATH vessel before and after installation both of fixed fore and controllable aft fins in regular head sea using time domain simulation and validated by model test in Towing Tank.

2. To apply a ride control system on the controllable aft fins, the conventional PID controller will be used to achieve a better quality the Semi-SWATH seakeeping performance.

1.3 Scopes of Research

1. The mathematical dynamics equations model covers Semi-SWATH vessels with fins in two degrees of freedoms i.e. heave and pitch motions operating in regular head sea.
2. The numerical method simulation is based on Time-Domain using Matlab-SIMULINK.
3. In the simulation, the regular waves generated using MATLAB for any wavelength of interest as well as experiment done (range of regular wave lengths: $0.5 \leq \lambda/L \leq 2.5$ and steepness of the incident wave: $H/\lambda = 1/25$)
4. The hydrodynamic coefficients of Semi-SWATH vessel motions will be obtained using numerical program, which was developed by Adi Maimun and Voon Buang Ain (2001).
5. The proper fin stabilizers were selected using NACA-0015 section due to high lift curve slope and low drag.
6. Coefficient of Lift (C_L) previously will be obtained using CFD software (Shipflow 2.8).
7. A conventional PID controller will be applied on the Semi-SWATH vessel to improve the stability and performance of plant system with adequate reliability.
8. A parameter tuning of PID controller is obtained using method of Aström and Hagglund i.e. K_p, K_i, *and* K_d. Then, they will be applied to satisfy certain control specifications by minimizing the error after achieving steady state.
9. The simulation program result will be validated of by the Semi-SWATH model test carried out in Towing Tank of Marine Technology Laboratory, Universiti Teknologi Malaysia.

1.4 Research Outline

An achievement of the excellent seakeeping qualities for ship design requires extensive consideration as guidelines to reflect the safety, effectiveness, and comfort of vessel in waves. The present research follows a systematic procedure to modify concept design of twin-hull vessel by minimizing their drawbacks. This study starts from the review of SWATH and conventional Catamaran hull forms. The final design of the new modified hull form will deal to enhance the vessel's stiffness associated with improving seakeeping qualities at high-speed in head seas waves condition. Then this vessel is called Semi-SWATH vessel.

The flexibility of the Semi-SWATH vessel can be operated in two variable draughts i.e. shallow draught and deep draught with still maintain seakeeping quality. In these variations of operational draughts, the Semi-SWATH vessel will be operated in two speed services i.e. medium speed (15 knots) and high-speed (20 knots). Furthermore, the effects of vertical motions on the Semi-SWATH vessel (heave and pitch motions) when encountering head sea at those service speed will be investigated considerably.

For this reason, an advanced prediction analysis both numerically and experimentally to achieve a desired goal will be done. In stage of the Time-Domain Simulation approach theoretically will be used to predict and analyze the seakeeping performance in head sea waves, which was developed using Matlab-SIMULINK. Then, the mathematical model comprising of heave and pitch coupled motions before and after attached fixed bow and active stern fin stabilizers are investigated. Then, the conventional PID controller is applied on the active stern fin stabilizer by tuning its angle of attack to enhance the improvement of ride quality ideally to be even keel riding condition. Then, the real-time simulation results will be validated by experimental model test carried out in Towing Tank at Department of Marine Technology, Universiti Teknologi Malaysia.

Finally, the seakeeping evaluation of Semi-SWATH vessel is identified based on the motion response, which presented by Response Amplitude Operators (RAOs).

The outline of thesis organization is shown in Figure 1.1.

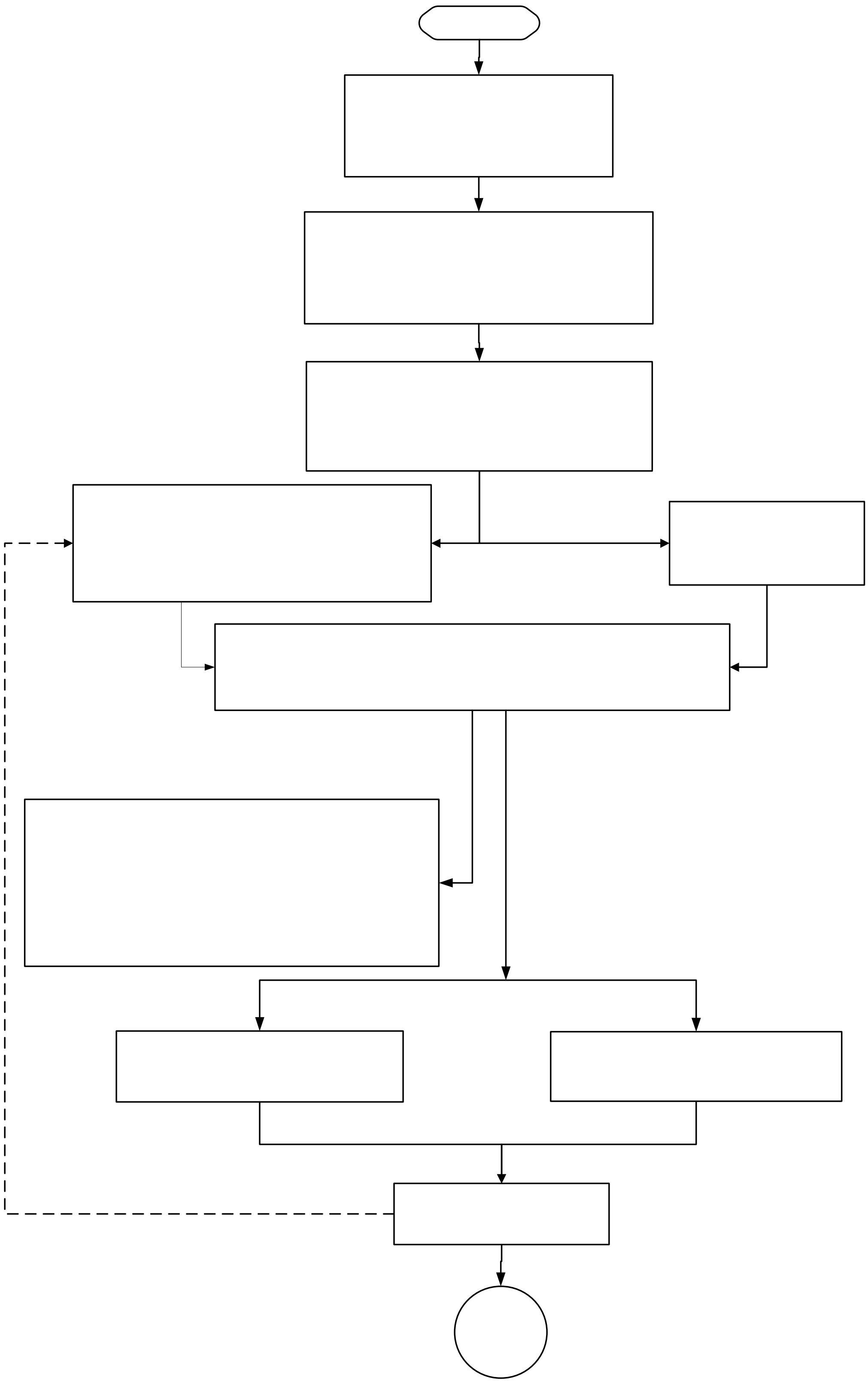


Figure 1.1 Outline of the thesis organization

CHAPTER 2

LITERATURE REVIEW

2.1. General

The aim of this chapter is to give an overview and the assessment method to evaluate seakeeping performance of the Semi-SWATH vessel. The extensive reviews of the pertinent literature have been done to obtain a useful information and methodology for this work. This thesis is organized in six sections, as follows; **the first section** is to provide a better understanding of the basic design of Semi-SWATH vessel modified between SWATH vessel and Catamaran. **The second section** treats on the critical review of the twin-hull vessel due to effect of vertical motion i. e. pitch and heave motion. These motions primarily have a significant effect to dynamic stability criteria especially at high forward speed. The additional feature of cross-structure submergence with a bow-diving of Semi-SWATH vessel add to the severity which degrade comfortability and structural damage with greater safety risks. **The third section** is to evaluate the effect of pitch stabilization on the vessel motions during operation. **The fourth section** utilizes the development of a rationale simulation control systems of fin stabilizers. The conventional PID controller will be established. **The fifth section** is the critical review of the existing seakeeping criteria of Semi-SWATH vessel as a type of twin-hull vessel.

2.2. Historical Design of Semi-SWATH vessel

Early, the development of twin-hull high-speed vessels naturally focused on developed a reputation for poor seakeeping performance when encountering head sea at high forward speed. Beaumont and Robinson (1991), Brown, Clarke, Dow, Jones and Smith (1991) and Roberts, and Watson and Davis (1997) stated that this bad reputation was shown by their tendency for larger pitch motions or even bow diving and more severe dynamic structural loads than for mono-hull vessels. Consequently, they will threat the vessel comfortability and safety.

Inherently, several solutions have been attempted to improve their performances particularly to minimize their drawbacks and risks. One of the solutions have been proposed is create or modify a new hull form design. In this thesis, the author had endeavored to develop an alternative design hull forms or modified design of twin-hull vessel hull forms to meet those requirements. This new design hull form will not only maintain the quality of good seakeeping performance in seaway but also directly is able to minimize the large pitch motion by increasing the vessel’s stiffness longitudinally. Thus, some extensive reviews of twin-hull vessels especially for Catamaran and SWATH vessel have been studied considerably. Where, the coupled mother vessel between a SWATH vessel and Catamaran vessel result a new genetic vessel design of a ***Semi-SWATH vessel***.

2.2.1 Catamaran

Dubrovskiy and Lyakhoviyskiy (2001) explained the local term “katto maram”, meaning “coupled tress”, become the commonly accepted word Catamaran. The principle of the Catamaran vessel geometrically is the connecting structure between the two hulls was used for navigation and became known as "the bridge" or cross deck. The modern feature of Catamaran vessel can be seen at figure 2.1.

In the recent decades, the research and development of Catamaran vessels had widely spread in world; Japan, USA, UK, Australia, Norway, Russia etc. Gartwig (1974) stated that the first Russian high-speed Catamaran Express was built on the Black Sea before World War II. By that time, the Catamaran vessels were designed to fulfill comfortability and seaworthiness. Furthermore, Michael (1961), Mandel (1962), and Meier (1968) discussed that the application of Catamaran concept was extended for special purposes, namely, oceanographic research, oil-drilling platforms, and ferries.

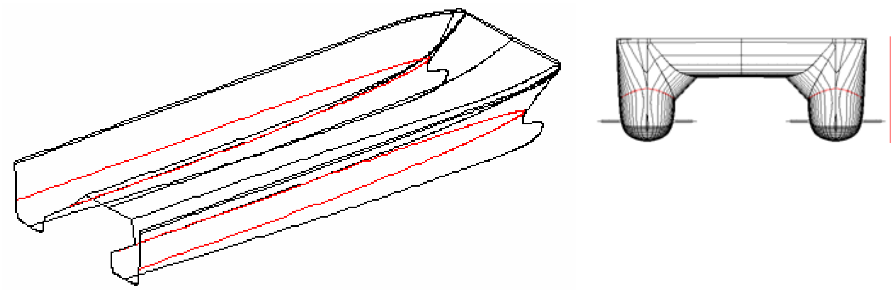


Figure 2.1; Catamaran vessel profile and section

2.2.2.1 The advantages of Catamaran

Kenevissi (2003) addressed that a Catamaran vessel is based on design feasibility and its operation. Hadler, et al (1974), Ozawa (1987) and Kamlesh (2000) investiagted design feasibility of Catamarans offer many practical advantages such as; large deck areas around 20 and 40 percent greater than a corresponding mono-hull and have better stability characteristics in favor will enhance a seakeeping in moderate sea states.

In the view of economic aspects, the feature of Catamaran vessels have offered some benefits as coefficient pared both SWATH vessel and mono-hull vessels. Dubrovskiy, and Lyakhoviyskiy (2001) have compared Catamaran vessel to SWATH vessel, it has proven that the Catamaran vessel has lower cost of construction with shallower draft operation. Dubrovskiy and Lyakhoviyskiy (2001), Hadler et al (1974), and Min et al (1987) compared with other high-speed craft; they possess good transport efficiency at moderately high-speeds.

Other advantages as compared to SWATH vessel, Catamaran vessels have better either longitudinal stability or transverse stability as compared to SWATH vessels and mono-hull vessels. Dubrovskiy and Lyakhoviyskiy (2001) state that the better longitudinal stability (GM_L) is offered by larger waterplane area as consequence as increased the vessel stiffness. In addition, due larger waterplane areas at the bow results more buoyancy forces, which can reduce the pitch motion as compared to the SWATH vessel. Thus, the tendency of Catamaran to the bow diving can be minimized. Dubrovskiy and Lyakhoviyskiy (2001) has investigated that the transverse metacentric height of Catamaran is 8-10 times greater than comparable mono-hull.

2.2.1.2 The drawback of Catamaran

Inherent to feature of Catamaran due to larger beam has negative effect to the performance in seaway. Comparing the vertical deck edge acceleration, wave excitation finds considerably larger amplitudes for the Catamaran than the mono-hull vessel. This is probably caused by the fact that the two fore-bodies of the Catamaran in bow sea encounter the wave crest with a certain difference in phase, which is unfavorable. Khristoffer (2002) investigated that Catamarans has revealed to the cross-structure slamming problems in seaway. Consequently, Hadler, et al (1974) had shown that Catamaran has a limited operation because of the cross-deck structure will have local indentations or even rupture.

Other drawback of the Catamaran geometrically in waves is higher wave resistance components compared to mono-hull. Molland et al. (1994) and (1996) gave evidence and found approximately 10% greater form factor than mono-hull due to viscous interaction effect. This phenomenon was caused by owing to high wetted surface area and hence skin fiction drag. Accordingly, Catamaran will require more high power for ocean going and its construction cost is slightly higher compared to mono-hull vessel.

2.2.3 SWATH vessel

The initials S.W.A.T.H. stand for Small Waterplane Area of Twin Hull. The SWATH vessel is a relatively recent development in ship design. Kennell (1992), the superior seakeeping quality is usually the primary motivation for considering SWATH vessel. Although patents employing this concept show up by Nelson (1905), Blair (1929), Faust (1932), Creed (1946), Leopold's (1969), and Lang (1971).

The general configurations of a SWATH vessel geometrically is that two fully and two partially submerged hulls connected by columns and slender beams or thin vertical members are called struts. They were located on each side of vessel in a tandem configuration, and a cross structure Liu and Wong (1986) and Goodyear et al (1989). These configurations offer the streamlined vertical columns (struts) can pierce the water surface and support a cross-structure. The feature of SWATH ship can be seen at figure 2.2.

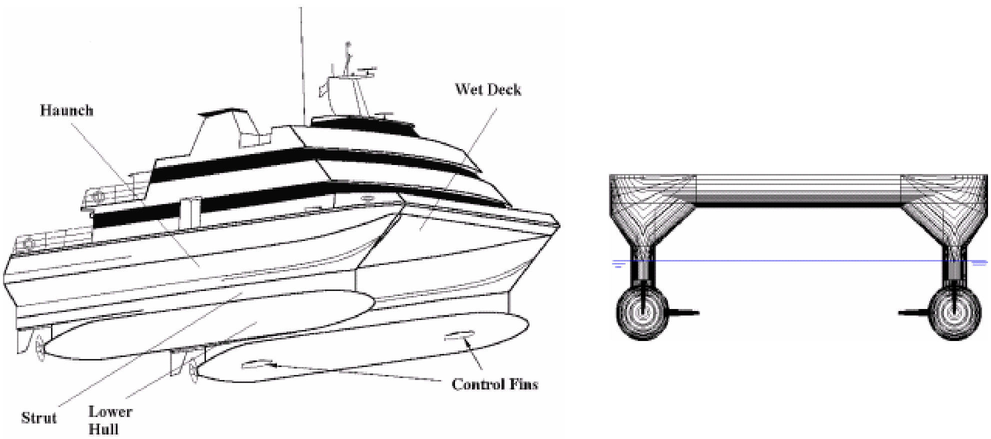


Figure 2.2. Conventional SWATH vessel profile and section

2.2.3.1 The advantages of SWATH vessel

Seren (1983) and Kennell (1992) presented that the major attribute of SWATH vessel is considered to offer good ride quality exceeding compared with a monohull vessel of equivalent displacement under rough condition. Kennell (1992), McCreight (1987), and Fang (1988) stated that the better seakeeping performance of the SWATH vessel concept is associated with its small waterplane concept. Those advantages were naturally provided by a stable platform in seaway and their features indicate significantly minimizing a drag as well as keeping the resistance down, low wave excitation forces, greatly reduced deck motions exhibited while at rest or underway, lower accelerations for given amplitude of motion, eliminate the seasickness, and give the vessel mobility comparable to mono-hulls.

Inherent to their operational draughts, the SWATH vessel has deeper draught than Catamaran vessel with comparable displacement. Seren (1983) presented an advantage to this feature considered be able to offer medium speeds, a wide stable platform with good stability, and good seakeeping ability exceeding that of much larger conventional mono-hull vessels under rough sea conditions.

Ozawa (1987) summarized advantages of the SWATH ships inherently compared with the equivalent monohull vessel can be taken as follow:

(i) less motions and acceleration by waves, longer resonant frequency characteristic of larger monohull vessels; especially as the rolling resonant frequency is very long, motions and acceleration by waves are less than for conventional hulls.

(ii) less speed loss in rough seas than conventional vessels because of minimal pitching motions.

(iii) directional stability and maneuverability at both low and high speeds are good, owing to the widely separated struts and the effective differential thrust system.

(iv) good intact and damage stability due to large reserve buoyancy of the strut's flare and the deck and also moderate ballasting system.

The above cited activities and advantages of the SWATH vessel design concept support a belief in its unique capabilities and practicability over a broad range of missions.

2.2.3.2 The drawback of SWATH vessel

The seakeeping of SWATH vessels are gained at some sacrifice. Holloway (2003) explained that the increased motion amplitudes of SWATH vessel due to an increased wave force associated with the longer wavelengths is possible to occur resonant motions. Furthermore, the lower natural frequency means that resonance occurs in longer waves, also contributing to longer motions.

Djatmiko (2004), Dubrovskiy, and Lyakhoviyskiy (2001), McCreight (1987) and Clark, et al (1990) explained in the view of safety aspect, the significant drawback accompanied with low waterplane area. This feature was caused the SWATH vessel has more sensitivity of draught to changes in weight during design and operation. It means, the low waterplane area brings about reduction in the moment to change trim, which also means low hydrostatic restoring moment, hence makes the vessel vulnerable towards pitch instabilities due to dropped or low restoring pitch moment resulted as increase in speed.

Another problem of the SWATH hull form in view of the commercial viability has caused the higher cost of construction. McGregor (1992) and Ozawa (1987) presented thatt, the SWATH vessel is still prohibitive due to require very high power to cruise at moderately high-speeds. This problem was caused by the greater wetted surface that disproportionately deeper draught as compared to mono-hull or Catamaran, Kennell (1992).

Ozawa (1987), summarized disadvantages of the SWATH ships inherently compared with the equivalent monohull vessel can be taken as follow:

1) an increase in propulsion power due to its greater wetted surface area which causes an increase of frictional drag.
2) sensitivity to trim and heel changes by weight ship on deck due to its small tones per centimeter properties, and comparatively small metacentric height it requires more severe KG allowances in the design stage than conventional vessels, ballast compensation systems or necessary for a SWATH ship.
3) a greater draught causing docking and restricted draught problems, especially in a large SWATH ship.
4) in the case of vessel built of steel, a smaller ratio of payload to structural weight compared with the equivalent conventional displacement type monohull vessels.
5) relatively larger turning diameter in relation to length in a high speed SWATH ship.

2.3 The Concept of the Semi-SWATH vessel

Based on the reviewing both SWATH vessel and Catamaran vessels are driving the search for new design concept, which had better satisfy design requirements. Those were shown by, SWATH and conventional Catamaran vessels, which provide several obtainable advantages.

Shack (1995) began from the review of conceptual design of fast passenger vessels that involves several unsolved problems regarding resistance, propulsion, seakeeping and passenger comfort. The objective of this study has been to establish a broad view of the concepts that could possibly be used to produce a new generation of high-speed passenger of the future. The study includes mono-hulls, Catamarans, SES (Surface Effect Ship), SWATH vessel (Small Waterplane Area Twin Hull). Then, he proposed a hybrids hull form with very promising results and would therefore serve as a good platform. Then this vessel is called the Semi-SWATH vessel.

Atlar (1997) had been studied comprising with Catamaran and SWATH vessel designs. He recommended a new concept of vessel design is to minimize the vessel responds to any disturbances, producing a harsh and so-called “stiff” ride especially at high-speed as compared to SWATH vessel. As may be known, the vessel’s stiffness has significant effect of creating high absolute vertical accelerations, a well-known cause of motion sickness. Joseph, et al (1984) and Atlar (1997), they proposed a mutation of generation hull form that conceptually is still taking the small waterplane area of twin-hull (SWATH) on hull form designs. Then, this vessel is called Semi-SWATH vessel.

Holloway (2003) presented that the semi-SWATH idea is an obvious opportunity to exploit the positive aspects of both SWATH and conventional Catamaran hull forms. Gaul, et al (1984, 1987, 1994, and 1988), Lang, et al (1979) and Holloway (2003) investigated that conceptually this design also was emphasized on the variable draught operations to minimize the single draught of SWATH vessel and to trade off performance and seakeeping. The primary objective of Semi-

SWATH vessel is to give the best possible ride in a seaway, especially when the vessel is adrift, holding station, or underway at low speeds.

Holloway (2003) developed the fundamental concept design relating to semi-SWATH geometrically is emphasized on the waterline beam reduction for the whole length of the boat or only for part of the length (for example only for the forward half). There is considerable scope for variations, and as yet there is no “normal” semi-SWATH design. In the view of his concept design, the distinction between a SWATH and a conventional hull is that the waterplane area in the former should be smaller than the maximum submerged plan area that is the waterline beam for at least some sections is smaller than maximum beam at the same sections. On other hand, Kristoffer Grande (2002) had considered the same background design to find a better seakeeping performance of twin-hull vessels in seaway. He has proposed a design of the Semi-SWATH vessel hull conceptually by reducing the waterline line width in the bow, and produces a very fine entry.

2.3.1 Advantages of Semi-SWATH vessel

Holloway, (1998) and (2003) investigated that generally the design concept of Semi-SWATH vessel configuration offered two ways that make the most of Semi-SWATH vessel’s benefits. First, its primary premise is to maintain a good seakeeping quality. Second, it is intended to prevent the deck-diving phenomena at high-speed. It means the maturity of Semi-SWATH vessel is going to provide an improvement of conventional Catamaran and SWATH vessel drawbacks considerably.

In the view of seakeeping quality, Gaul (1988), the Semi-SWATH vessel or Semi-Submerged ship has promised to provide a better operability for overboarding in high sea states. Gaul (1988), Coburn (1995) and Holloway (2003) presented a better seakeeping quality was achieved due to variable range of her draught mode operations, which is able to trade off the performance and seakeeping quality accordingly. These were shown, in the view of SWATH mode (deep draught condition with ballasting system) is evident greatly to the reduced waterplane area and deck motions exhibited while at rest or underway. This draught mode gives the steadiest platform for slow and medium speed operations incident rough sea conditions in deep water. Holloway (2003) gave a reason that the Semi-SWATH vessel provides for lower the natural frequencies of heave and pitch response as well as less wave-exciting forces. In addition, Davis and Holloway (2003) stated that the SWATH mode will give benefit in order to reduce the magnitude of motions particularly in the forward parts of the vessel. McGregor (1992), Lang (1988) stated that the greatly reduced pitch motions will improve seakeeping quality while at rest or underway. Gaul (1988) studied even in the deeper draught, the seakeeping quality of Semi-SWATH vessel can be superior to an equivalent SWATH vessel.

In the view of preventing a deck-diving phenomenon at high-speed was offered in her operational mode (conventional Catamaran mode). Holloway and Davis (2003) explored in this mode, the Semi-SWATH vessel will exist to be operated in shallow draught conditions such as in estuary area and sheltered water with unique deballasting system appropriately. The condition of shallow draught is set up for a transit draught where it is fully loaded with the ballast tanks empty. As results, this feature is especially useful during transit to permit higher speeds. The configuration by enlarging waterplane area of vessel has substantially increased the hydrostatic stiffness relative to the ship mass as consequence as satisfy the seakeeping design objectives especially at high-speed that SWATH vessel cannot do. As a result, the vessel’s tendency to an excessive trim or even bow diving when encountering head seas at high-speed can be reduced. Gaul (1979) presented other advantages of Semi-SWATH vessel is due to capitalized on wide footprint, generous deck space, and trim flexibility to give low motion profile.

2.3.2 Motion Response of Semi-SWATH vessel

Holloway (2003), Lang (1979) and Gaul (1988) presented that the dominant advantage that the Semi-SWATH vessel or Semi-Submerged vessel offers is drastic reduction in ship motion. Gaul (1988), the improved seakeeping is provided in both the transit mode and, more importantly, during on-station research operations. This motion response reduction allows the Semi-SWATH vessel to operate in much higher seas than can be tolerated with an equivalent mono-hull. The radical decreases in waterplane area above the lower hull reduce buoyant forces caused by wave action and ameliorate motion response. It should be noted that this is a key difference between a Semi-SWATH vessel and a conventional Catamaran vessel.

Holloway (2003) studied on two model tests that the Semi-SWATH vessel or Semi-Submerged ships, even at transit draught, will have lower motion response that a comparable conventional hull. Gaul (1988) explained that in beam seas, the roll of mono-hull can be more than five times higher than that of Semi-SWATH vessel. In pitch, the mono-hull response typically is two to three time higher. This feature alone prompts the consideration of Semi-SWATH vessel in place of mono-hulls for passenger vessels.

Gaul (1988) stated that the response of the vessel to a seaway is highly dependent on wave periods or, more precisely, on encounter frequency, which is a function of wave frequency, vessel speed, and relative heading. When underway in head seas, the apparent encounter frequency is longer than wave frequency, so vessel motions are very small. On the contrary, when the apparent frequency is shorter than wave frequency, the Semi-SWATH vessel motions are very long. As results will cause exceed deck motion as threshold for seasickness discomfort.

Schack (1995) studied on the demihull series of hull forms. He developed ranging from the pronounced Semi-SWATH hull form to a conventional high-speed catamaran with a typical U-shape. Due to there being very little interference and interaction between the demihulls both regarding resistance and seakeeping, it was decided that the calculation and model test should be carried out with the demihull only, this was furthermore validated via the model test program. On the basis of the small systematic study conducted via numerical calculations and model tests, it can be concluded that the Semi-SWATH hull form in the most probable sea states is superior to the conventional catamaran hull forms.

Holloway (1998) has summarized the response related to model test of Semi-SWATH vessel in Towing-Tank, as follow;

- as speed is increased motions also increase, primarily due to the increased forcing resulting from encountering resonance in longer waves, but only to the point where the resonant frequency is encountered in wavelengths significantly longer than the boat length, in which case the motions are asymptotic to their maximum value. This means that at high-speeds poor seakeeping is inevitable for all hull types (excluding the effect of appendages).

- for the same reason increasing SWATHness (waterplane area reduction) also increases motions. However, the lower natural frequency of SWATHs at the same time reduces accelerations, and this to some extent cancels the effects of the increased motions. The increased motion effect dominates at low speeds where the rate of change of forcing with frequency at the natural frequency is high, but at high speeds where it is asymptotic (and therefore changes only slowly with frequency) the lower acceleration effect dominates.

2.4. Prediction of Ship Motion

Currently, the most popular methods for computing the seakeeping performance of vessel are based on strip theory. Various forms of strip theory have been is use and development since the early 1950s (Beck, 1989). The essence of the method is the approximation of the three-dimensional fluid flow problem over a hull by a series of two-dimensional strips as illustrated in figure 2.1 (Faltinsen, 1990). Strip theory is based on the assumption that the oscillatory motions of the vessel are linear and harmonic and occur at the frequency of the incident wave. In this method, all motions and force coefficients are computed as function of frequency and so are generally said to be computed in the frequency domain, (Oglivie, 1964).

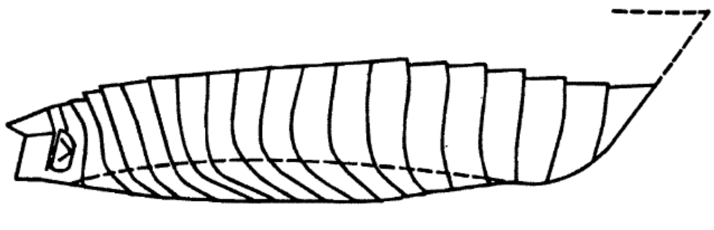


Figure 2.3; Illustration of strip theory for ships, Faltinsen (1990)

Many of the early methods were limited to zero speed, head seas or motion in the vertical plane only. During 1969 and 1970, several papers were published by different groups working independently which introduced more general forms of the theory including Söding (1969), Tasai and Takaki (1969) and Borodai and Nesvetayev (1969). The method described by Salvesen et al. (1970) has been the most widely accepted (Beck, 1989). All these new strip theories have identical forward-speed terms satisfying the Timman and Newman symmetry relationships, and, interestingly enough, the equations of motion for heave and pitch in head waves derived in the present work have the same speed terms. This method also includes prediction of sway, roll, and yaw motions as well as wave induced loads a ship at constant speed at an arbitrary heading in regular waves.

There are three main stages to computing the ship's response using strip theory. First, the ship is divided into a number of transverse sections or strips, typically numbering twenty to forty. Second, the two-dimensional hydrodynamic coefficients (added mass, damping, wave-excitation and restoring force) are computed for each section. Third, these values are then integrated along the length of the vessel to obtain the global coefficients for the couple vessel motions. Finally, the equations of motion are solved algebraically.

Two methods are commonly used to calculate the hydrodynamic forces on the strips: are conformal mapping and Close-Fit methods. In the first method, the section coefficients are calculated by relating the actual section shape, via a conformal mapping, to that of a union semi-circle for which the solution is known. Various methods have been proposed to perform this mapping such as those of Lewis (1929) and Ursell (1949). The most commonly used of these is based on the Lewis forms which use two parameters based on the sectional beam-to-draft ratio and the sectional area coefficient to define the mapping.

Close-fit methods are those, which attempt to solve the potential flow problem directly on the actual sectional geometry using boundary integral techniques. The method Frank (1967) is perhaps the most commonly used of these methods. The Frank Close-fit method represents the section shape as a series of straight-line segments. A series of fluid sources of constant but unknown strength are distributed along each segment. Applying the boundary conditions permit solution of the source strengths and hence the velocity potential on each segment. The pressure associated with the velocity potential is then integrated over the surface of the section to yield the added mass and damping coefficients.

While the formulation of strip theory assumes a long, slender (Faltinsen, 1990), its use for alternate hull forms was anticipated by Salvesen et. al. (1970) over thirty years ago. The method has subsequently been applied to more complex hull forms including, Catamaran, SWATH vessel and others types of multi-hull vessels. The method has been proven to provide reliable estimates of motions and hull loads for a surprisingly wide range of hull forms and sea conditions as evidenced by the wide range of engineering software products in use today. A theoretical method for predicting the ship motions in waves was applied to the Semi-SWATH vessel for similar to that applied to SWATH vessel (Lee, 1976). He applied a conventional strip theory of Salvesen, Tuck and Fatilsen to twin-hull ship in order to develop potential flow coefficient. Where, he has developed the SWATH vessel motion prediction in waves using strip theory with involving with three main extra considerations included in the hydrodynamic coefficients.

1. the hydrodynamic interactions between two-hull.
2. effect of viscous damping. It is of the same order of magnitude of the wave making damping. Consequently neglecting it results in twin-hull system being under damped.
3. effect of stabilizing fins. A Semi-SWATH vessel can become unstable in pitch above a certain speed. The longitudinal unsymmetrical pressure distribution on the two lower hull produce a destabilizing moment causing the vessel to “bow down.” The moment is called the “munk-moment” and is proportional to the square of the speed. This associates needs for compensating stabilizing fins. The fins also help damping through hydrodynamic lift. Therefore, a Semi-SWATH vessel is not viable without fins and their effects must be included in any calculations of motion.

2.5 Motion Characteristics of High-Speed Vessels

The Semi-SWATH vessel is known to offer a high quality in low and moderate sea states compared to conventional Catamaran. However, high-speed Semi-SWATH vessel in heavy sea states there are problems with discomfort owing to high frequency vertical accelerations especially due to large heave and pitch motions. Furthermore, the lower natural frequency means that resonance occurs in longer waves, also contributing to larger motions, Segundo, et. al (1999 and 2000). Therefore, a reduction in heaving and pitching will in general improve comfort for passengers.

In addition, designers and builders are starting to see the benefits of better seakeeping not only in terms of passenger comfort but also in terms of dynamic structural safety problems. Christoffer (2002) had shown that slamming and deck wetness events are likely to be both more severe and more frequent than for mono-hull vessels at high forward speeds. Consequently it is not only degrade the vessel performances but also damage the structure with greater risk. This phenomenon was occurred on a high-speed vessel, which receives a not inconsiderable amount of lift forward from the spray rails. Tank tests have shown that, once a certain speed is exceeded, these may cease to deflect the bow wave or spray and become engulfed. When this happens, the bow may drop, to the accompaniment of large sheets of green water thrown into the air in the region of the fore body.

Therefore, the evaluations of motion characteristic of high-speed twin-hull vessel particularly due to large vertical motion (heave and pitch motions) a will be taken respectively in a present considerable research that has already undertaken in the area.

2.6 The Effect of Heave and Pitch Motion Responses

Goodrich (1968), during operation, the heave, and pitch motions would routinely influence to the magnitude of vessel. The effects were concerned to the parametric their large excitation motions as considered as critical situation suffered by the vessel. As results, they will generate wave impacts on the bottom of the cross structure of a Semi-SWATH vessel, as well as the usual problems of deck wetness on the forecastle or of slamming on the forefoot. The motion of ships in the sea was violent, pitch ± 8 degrees being commonplace. Sariöz and Narli (2005), for a passenger vessels, vertical motions (heave and pitch motions) are of main concern due to effect of accelerations on the comfort and well-being of the passengers. Ozawa (1987), hence, their large vertical motions affected to high accelerations, which deal to affect some negative aspects on the vessel during operation, such as slamming.

Djatmiko (2004) stated that as the twin-hull vessels, the small waterplane area at the bow of SWATH vessel at high-speed operation will cause large pitch motions as consequence as more sensitive to trim and heel changes by weight ship on deck due to its small tones per centimeter properties, and comparatively small metacentric height. Further, the low waterplane area due to heave and pitch motions bring about reduction in the moment to change trim, which also means low hydrostatic restoring moment, hence makes the vessel vulnerable towards pitch instabilities.

Kenevissi (2003) explained that the large tendencies heave and pitch motion of the SWATH vessel in high forward speed produces a harsh stiff ride or low restoring moment. Where, this stiffness has the effect of creating absolute vertical motion acceleration. The low value of restoring moment is linked to its values of longitudinal metacentric height (GM_L). In addition, the insufficient of GM_L may experience a static trim and at the same endure a much sinkage compared to its draught. Djatmiko (2004) was shown, on top of that, sinkage and static trim naturally yields an increase in total resistance due to involuntarily additional wetted surface area. Katayama, (2002b), Djatmiko (2004), Dubrovskiy and Lyakhoviyskiy (2001) presented that those phenomena occurred when hydrodynamic forces are dominant compared with hydrostatic forces like buoyant forces.

Further effect of pitch and heave motions to the Semi-SWATH vessel particularly at her operational modes has different characteristics subjected to the vessel's motion responses. In the conventional Catamaran mode, its larger waterplane area has effect to reduce the dynamic instability of vessel's motions above a certain critical speed. Oppositely, on the SWATH mode, its small waterplane area will affect on the minimizing of wave-exciting forces. As a result, the Semi-SWATH vessel has relatively less resistance and longer natural periods of heave or pitch motion compared to Catamaran. McCreight (1987), these longer natural periods of motion enable the vessel to avoid resonant motions resulting from wave-excitation forces in normally encountered seas.

Inherent to the geometry of the Semi-SWATH hull form is maintaining the waterline line width as SWATH hull form characteristic in the bow produces pitch motion. Excessive pitching on the relative vertical motion in the bow has strongly influenced by sea conditions, in terms of ship motions and relevant sea loads, causing uncomfortable conditions for passengers and hazardous situation for structural integrity. Fan and Xia (2002), Esteban (2000), ISO (1995) and (1997), Dubrovskiy and Lyakhoviyskiy (2001), and Davis and Holloway (2003) were shown in the worst condition as a softer bow end responses make the vessel more prone to bow diving in head seas and to a loss of longitudinal stability.

2.7 Pitch Motion Stabilizations

A high-speed vessel is riding in seaway with longer wavelength (one to two times as long as the length of vessels) will take some unacceptable motion responses. This results in severe pitching motion of high-speed vessels.

Abkowitz (1959), Bhattacarya (1978) and Conolly and Goodrich (1968) discussed that the severe pitching motions of ships, particularly in head seas, are responsible for loss of speed for a given power, for slamming with its associated risk to the structure, motion sickness and possible injury to humans and damage to equipment from the high accelerations that can be produced. Dubrovskiy and Lyakhoviyskiy (2001) and Kennell (1992) discussed as well as twin-hull vessels, inherently the SWATH vessels have limitation factors due to unacceptable pitch motions particularly in high-speed operations. The unacceptable pitch motions have often characterized by vertical struts of Semi-SWATH vessel designs (SWATHness mode) during underway due to inherent small pitch damping. This pitch instability was occurred due to its insufficient values of longitudinal metacentric height (GM_L) resulted as increase in speed.

In addition, Fairlie-Clark (1990), Hadler, et al. (1974), Pinto, et al. (2001) investigated a hydrodynamic force on the lower hull of SWATH vessel configuration is susceptible to trim changes. Liu and Wong (1986), Papanikolaou (1991) presented a result that the destabilizing bow down pitch moment (munk-moment) phenomenon is generated as consequence as a pitch-instability can therefore occur at higher speeds, while the normal static restoring force due to the waterplane is dependent only on the pitch angle and ship speed. Holloway and Davis (2003) had studied that in this situation the seakeeping quality of ship is poor because of resonance occurring in combination with significant wave excitations.

By comparison, Semi-SWATH vessels have lower wavemaking damping in all modes of motions as compared to mono-hull vessels. Since, the motions of Semi-SWATH vessel produce little wave excitation action. For this reason, the motions of Semi-SWATH hulls are lightly damped and can produce large responses if excited at their natural periods. Kennell (1992) and Sariöz and Narli (2005) stated that negative aspects of these large motions can be avoided by selecting natural periods different from the most probable wave encounter periods or providing additional damping forces to limit resonant responses. Therefore, the pitch instability must be corrected in order to allow operation of the vessel above a certain critical speed.

In general, many efforts have been attempted to overcome such a problem, and the most potential method attained is by introducing the anti-pitching fins. Bhattacarya (1978) and Kennell (1992) stated that the one device that very useful and currently used to minimize unacceptable pitching motions or pitch instability resulting from high forward speed is the anti-pitching fin. Since, this device can increase the hydrostatic stiffness of the vessel, which have much less effect at higher and lower frequencies. Holloway (1998) explained that this of course refers to the bare hull, and damping can be significantly increased with the use of fixed fins, and even more by using controllable fins. Besso et al. (1973) and Kennell (1992) discussed the idea of reduction on ship motions by anti-pitching fin can be attached at the bow and stern of the ship. The results shown, the horizontal fins near the bow and stern dampen the pitch, heave resonant responses, and diminish the resulting slamming.

The reduction of the pitching motion of ships by means of pitch stabilization (the anti-pitching fins) at the bow and stern is analyzed both qualitatively and quantitatively, through theoretical and model experiments. Abkowitz (1959) examined the extra resistance of anti-pitching fins at the bow in still water and the effect of anti-pitching fins on ship motions in slamming conditions. In other hand, Wu and McGregor (1985) carried out an experimental investigation for the effect of motion stabilizing on the heave and pitch motions of a SWATH vessel. In addition, Ohmatsu et al. (1983) investigated the lift characteristics of fins attached to a semi-submerged ship in calm water and wave conditions. Their results showed that the fins are most effective near the resonance frequency range and inefficient at frequencies higher than 0.55 Hz in model scale.

Kennell (1992), Warlee and Luth (2000) stated that the anti-pitching fins may not only be used to lift the hull partially due to excessive trims or full out of the water but also to provide forces dampen the motions of the craft when operating in waves. By means of reducing the adverse motions and accelerations of the ship, as a result, the anti-pitching fin significantly enhances the seaworthiness at high-speed condition. Also, the fins mounted generally on the sides of Catamaran, SWATH vessel and Semi-SWATH vessel hulls have been used to increase hydrodynamic damping.

Ikeda et al. (1995) conducted model experiments to evaluate the effect of anti-pitching fins on ship motions. The calculation method is developed to estimate the ship motions for a ship with anti-pitching fins. In addition, a new evaluation method of passenger comfort is proposed to improve the accuracy of estimating the ratio of seasick persons during a long voyage. This method takes into account the exposure duration, amplitude of vertical acceleration and frequency of acceleration. Both the measurement of the ship motions and a survey of passenger comfort were conducted to investigate the relationship between ship motions and seasickness.

Shigehiro & Kuroda (2000) evaluated the effect of anti-pitching fins on the ship motions by a new method of evaluating passenger's comfort (seasickness) based on questionnaires on a training ship. The evaluation method took into account exposure duration as well as amplitude and frequency of oscillation. Model tests with and without anti-pitching fins at the bow and/or stern were performed and compared to strip theory calculations. Added resistance was measured during the model tests.

Thus, the considerable research of pitch stabilization to provide additional damping forces and additional pitch restoring moment integrated with installed a ride controls systems either active or passive at either stern or bow were being undertaken in the area. For example Haywood, Duncan, Klaka, and Bennett, (1995), also Caldeira-Saraiva and Clarke (1988) introduced the effects of the anti-pitching fin on a SWATH vessel.

2.7.1 Fixed Bow Fin

Conolly and Goodrich (1968) had studied the problem of pitch reduction fixed bow fins. His knowledge of activated roll damping fins led him to believe that a reasonable amount of pitch damping could be achieved using fins, even though the problem was different because of the higher inherent damping of the hull in the pitching mode. In 1958, NRDC approached the admiralty with a proposal to fit anti-pitching fins to a warship and to test the ship full scale. These fixed bow fins have been fitted to the SS Rhyndam and USS Compass Island. The result of sea trial showed that the fixed bow fin attenuated the pitching motions by between 20-30 percent in waves having a length of the same order that of ships. In longer waves of twice the ship length, however, the effect disappeared, whilst in longer waves the fins actually increased the motions. The heaving results show similar trends although the attenuation of motions in short waves was somewhat more marked, reaching almost 40 percent. In addition, Ikeda et al. (1995) discussed the effect of anti-pitching fins at the bow from the viewpoint of passenger comfort. As pitching motion was reduced by 10 percent, the ratio of vomiting was reduced by 20percent.

Vugts (1967) also presented a method to calculate the effect of anti-pitching fin fixed bow fins on the motions of a ship model. The results showed a pitch reduction of about one-third while the heaving was not significantly changed. In addition, Hadler et al. (1974) had conducted a full-scale trial on Hayes near the end of first winter of operating in the North Atlantic (1972), when it became apparent that there were problems of cross structure slamming. The purpose of these trials was to assess the seakeeping characteristics of Hayes before undertaking the study to improve her seakindliness. The second set of trials was conducted again in the North Atlantic (1973), after the hydrofoil had been installed at bow of ship. The result shows the relative bow motion was reduced about 30 percent, resulting in a corresponding reduction in frequency and magnitude of cross structure slamming. As consequence, the general seakindliness of ship significantly was improved.

Kallio and Ricci (1976) and Wu (1984) had investigated the function of horizontal fixed fins to reduce vertical motion responses for SWATH vessels. Kennell (1992), some SWATH vessels have been produced with fixed fins such as two hydrographic surveys ships KOTOZAKI and OTHORI have been equipped by passive fins. Furthermore, Fang and Lin (1995) had evaluated the motion responses of a SWATH vessel with fixed fins in waves. In addition, Dubrovskiy (2000) investigated the passive bow foil of the Catamaran that can decrease pitch amplitudes and vertical accelerations by 30 percent for a speed 30 knots. The effect of fixed fin in longitudinal waves was developed generally using both Frequency Domain and Time-Domain Simulation technique method. They found the fin stabilizers were indeed useful to improve the performance of SWATH vessel.

2.7.2 Controllable Aft Fin

Semi-SWATH vessel usually needs horizontal stabilizers for providing steady progress in still water. Conolly and Goodrich (1968) conducted an experiment of the model test and concluded that the spectacular reductions of pitching and heaving could not be achieved with fixed fin. because of that, a controllable or activated fins at stern would be required to produce large reductions of motions. It was nevertheless decided to undertake a full-scale evaluation of fixed bow fins before proceeding to the much more difficult problems which would be involved with a controllable fin installations. Controllable fin stabilization systems had been developed and utilized since the 1890's. The most significant advances came between World War I and World War II when Denny, Brown Brothers, and the Admiralty Research Lab in the UK began a joint effort to develop an active fin system. In 1959, the Royal Navy equipped all new vessels with these devices as a matter of course.

Liu and Wong (1987), Beenaand and Subramanian (2003), Fairlie-Clark et al (1990), and Pinto et al. (2001) presented that the utilized active anti-pitching fins have already improved the seakeeping characteristics of Semi-SWATH vessel by providing additional active damping forces. The active anti-pitching fins can control wave-induced ship motion, forestall the calm water pitch instability, and provide improved longitudinal stability resulting from high forward speed.

In many instances, the seakeeping performance of a SWATH ship can be greatly the use of activated fins. A method has been developed, which provides the naval architect with the necessary tools for designing active fin stabilizers for SWATH ships. Kennell (1992) investigated that the most of the SWATH vessels with speed greater than 15 knots have been built with active motion control systems. Some SWATH vessels have been produced with fins that actively control motions are KAIMALINO, SEAGULL, CREED and NAVATEK I. Active fins have been fully movable (incidence control) of flapped. These fins have been supported by both hulls or cantilevered off each hull. In addition, Ikeda et al. (1995), has evaluated the effect of active stern anti-pitching fins from the viewpoint of passenger comfort improvement. The results have been shown by the reduction of slamming forces at the bow of ship.

Kenevissi (2003) had shown the two vessels selected are SWATH6A, which is a well known conceptual benchmark vessel, and a high-speed Semi-SWATH vessel. The former has a pair of forward and aft fin stabilizers, whereas the latter has a pair of control surfaces only at the aft. His investigations of controllable stern fins have been recognized as being very effective in maintaining the stability and significant reduction of heaving and pitching motions of Semi-SWATH vessel at high-speed in head seas condition.

Fang and Lin (1998) presented a time simulation technique to evaluate the motion response of a SWATH vessel with either fixed or controllable fin in regular waves. He concluded that the pitch motion is indeed reduced effectively by using a suitable fin control procedure. The heave motion can also be reduced but only suitable fin control with respect to the flow in head waves and usually higher speed is helpful to fin control in head seas.

2.7.3 Design of Fin

Selection of anti-pitching fin characteristics is related to the performance of the hull form of vessels and the choice of passive (fixed) or active fins systems. Design objectives are to avoid calm water pitch instability, provide adequate damping in near resonance wave conditions, and improve motions. Manual control of the fins can be used to adjust trim while underway more quickly than is possible by transferring fuel or seawater ballast. These objectives are addressed in a design through selection of the total fin area, forward/ aft area ratio, location of the fins, and fin geometry (section shape, aspect ratio). McCreight (1987) presented the selection of fin locations and geometric characteristics for specific applications can be accomplished by respectively exercising a seakeeping program for a variety of fin configurations.

Stahl and McCreight (1984) selected the total fin area is based on maximizing motion reduction while providing adequate pitch instability. He had provided adequate formulation to obtain the effectiveness of fin areas in reduction motions. The selection of total fin areas and the bow/ stern area ratio was the specific function inherent to their characteristics during underway. Dubrovskiy and Lyakhoviyskiy (2001) explained the longitudinal stability of SWATH ship in forward motion, which can be enhanced by the stern stabilizers, whereas bow fins can lead to the opposite result. As was shown however, the stern fins alone can result in a decrease in heaving stability. Therefore, installation of both stern and bow stabilizers is recommended with ratio of their areas from 3 to 1 (i.e. stern fins are greater). Kennell (1992) presented that total fin area has been 10-30 percent of the waterplane area of the ship, with values 15-20 percent typical. Holloway, D.S. (2003) and Dubrovskiy and Lyakhoviyskiy (2001) showed that an increase in the bow fin area results in deteriorating its efficiency due to the ship at higher forward speeds, the bow fin area can be reduced up to 5-6% of the waterplane area and the fin can even be split in to smaller cantilever bow fins. In addition, Conollyand Goodrich (1968) adopted a final design bow fin area about 4-6% of the load waterline area. In other hand, Guliev (1972) and Neng, et al (1999) found that bow fin with an area of approximately 10% of the waterplane area is the best solution for easing Catamaran pitching. Besso, et al (1973) discussed the idea of reduction on ship motions by the anti-pitching fins at the bow and stern. The area of fins exceeds 10% of the water plane area of a ship at design draft. The results were validated with model experiments of a container ship.

Then, to determine the fin locations both vertically and horizontal must be considered as well based on design to increase the value of GM_L. Kennell (1992) found that the addition of fins near the bow will augment the destabilizing effects of the munk-moment. Conolly and Goodrich (1968) explained that one of the problems was the shallow draught, it being anticipated that the bow might emerge in waves and that slamming could be a problem. It was therefore essential that the fins should be as near to the keel as possible, yet should not present too flat a surface on the underside. Also they suggested that the fins had to be as far forward as possible to give maximum stabilizing effect.

Lee (2000) presented that the performance of foils or fin stabilizers supported craft is usually verified by using computational tools in an initial design stage and by means of model testing as a final check. It is shown, the effective position submergence depths of fin stabilizer is than three times the chord length. Dubrovskiy and Lyakhoviyskiy (2001) stated that lengthwise location of fin has a little effect on the force’s derivative but the moment’s derivative is proportional to the arm of the fin.

Guliev (1972) suggested that the foil should be placed as far forward as possible (for greater arm relative to the midship) and as close as possible to baseline in order to reduce the slamming probability for the fin. Installation of an additional stern foil can be helpful but its effect in head seas is rather low because of known migration of the instant axis of longitudinal oscillations towards stern.

2.8 Ride-Control System

In this context, the effective design and use of motion control systems have become an integral part of the design and operation of the vessels. Although, the use of motion control systems is nothing new, this new-generation hull form (Semi-SWATH vessel) also require a ride control system to take advantage of recent developments in computer technology.

Haywood, Duncan, Klaka, and Bennett (1995) have full considerable research in ride control system. Ware et al (1980), Ware and Scott (1980) and Ware et al (1981) presented an optimal control theory has been applied to improve the seakeeping characteristics of SWATH vessel by means of active fin stabilizers. Kennell (1992) presented significant reductions in the motions of SWATH vessels in harsh sea, environments have been demonstrated through the use of active motions control systems. This was recognized that the Ride-Control System most effective at speeds above 10 knots. To a certain degree, the lifting characteristics of fins have been reasonably well predicted by semi-empirical formulae shown, for instance, by Whicker and Fehlner (1958) and Pitt et al. (1959). The low wave exciting forces and hydrostatic restoring forces associated with SWATH hulls result in enhanced effectiveness for these fin systems.

The design of Ride-Control System for actively controlling the motions of a vessel in head seas is generally approached by developing a vertical plane i. e. pitch-heave controller. This controller is generally feedback systems that sense the motion of the vessel and drive the fins produce forces to counteract the motions. More sophisticated systems sense additional motions related data such as wave characteristics or relative motion at the bow to control others aspects of ship motions.

In Semi-SWATH vessel, the vertical clearance is a critical parameter characterizing a margin of seakeeping in designing and tuning the ship control systems. Considerable reserves for improving the seaworthiness of Semi-SWATH vessel in resonant regimes can be revealed by programming the sophisticated motions control systems to the appropriate sea state parameters.

Zarnick (1998) and Caldeira and Clarke (1988) gave sophisticated methods have been developed to model the interactions of the components of the motions control systems, determine the gain and phase relations for the control algorithms needed to produce the desired motions results, and predict the benefits of active control to design. Control system data needed for these methods can be estimated from available data for similar systems early in a design and subsequently revised as characteristics of equipment selected for the design become available.

Kang et al (1993) and Kang and Gong (1995) investigated an attitude control system that was designed for a high-speed Catamaran with controls fins in waves by the experimental methods. Optimal control theory was applied for the vertical plane of motion. Results showed that the controller was very effective on the pitch motion but no so much on the heave motion. These investigations were based on the frequency domain technique to improve the heave and pitch motions.

Among various new-generation control techniques, an optimal PID controller technique as an alternative solution to the motion control problems of Semi-SWATH vessel. In recent study, many researchers explored applied this technique for the yaw, roll, heave, pitch motion control either mono-hull or twin-hull vessels operating in regular seas. Tsuchiya (1990) applied the concept of PID control to minimize the error between reference input and controlled variable by using current and future information of reference input or disturbance.

2.8.1 Application of PID Controller on The Ship Motion Improvement

Giron-Sierra et al (2002) stated that the vertical damping motions of fast ship, which have a negative on comfort and safety, can be attenuated using moving controlled actuators. These actuators must move to counteract the vertical motions induced by waves using conventional PD controller. The results show the clearly satisfactory reduction of vertical motions with improved the comfort of passengers. In addition, Segundo, et al (2000) developed simulation program using PID controller to alleviate vertical accelerations due to waves. The results of simulation had been validated by experiments in the towing tank confirm that the MSI can be dramatically reduced about 59.3 %. Rueda, et al (2002) developed various classical control structures by means of genetic algorithms with the aim of reducing the vertical acceleration of a high-speed craft in order to decrease motion sickness incidence (MSI). Using Genetic Algorithm technique incorporated with Standard and Series PD, Parallel PD get better results by greatest reduction in the vertical acceleration of the craft is due to the reduction in the pitch acceleration. Esteban, et al (2000), in this research considers a fast ferry with transom flaps and a T-foil. A control method based on the prediction of perturbations has been developed and applied to a fast ferry. The results of the new control methods are compared with an optimal PD, and in every case, the new control performs better. The results for 20, 30, and 40 knots shows good improvement over the PID reference. By means of flaps and a T-foil, moved under control, vertical accelerations can be smoothed, with a significant improvement of passengers comfort.

Haywood, et al (1995) developed a Ride Control System (RCS) for fast ferries Catamaran 40m length. The simulation was able to provide an excellent test bed for the controller algorithm in both its operation and the robustness of the software. In this simulation, the adaptive PID controller had applied. The simulation studies showed that the best control was obtained by using the vertical velocity as the controlling variable rather than acceleration. The velocities were obtained by applying an integrating filter to the accelerations, calculated at the accelerometer positions.

Neng, et al (1999) investigated the usefulness of active stabilizing fins to reduce ship pitching under varying forward speeds, and in varying wavelengths using PID controller. Pitch angles, pitch rates, and ship forward speeds were measured and feedback to a controller to calculate the deflection angle of the stabilizing fin. The result shows how the lift capability of the fin limits the ability of the feedback control to change the pitch behavior of the ship. It allows control design and mechanical design to be integrated, and can provide basic information about the anti-pitching performance at an early stage of the ship's design process.

The above review PID control techniques indicates that basic control techniques have greater potential to enhance the performance, flexibility, robustness, and reliability of the control systems. Therefore, their applications on marine vehicles have been gradually increasing. In addition, the controllers design to ensure satisfactory performance controller to the ship performance in real-time is preferably based on Time-Domain Simulation technique.

2.9 Time Domain Simulation

The aim of the Time Domain Simulation approach is to relate the performance of ship motion responses. Previously, this concept is hard to be followed because it involves many complicated computation procedures. However, with the advancement of computer technology, this study becomes easier and it has attracted many researchers to follow this concept such as Paulling et al. (1975), Hamamoto and Akiyoshi (1988), de Kat and Paulling (1989) and Umeda et al. (2000). Under this concept, the equations of motions, which are made up by three translation components: surge, sway and heave, and three rotational components: roll, pitch, and yaw are solved simultaneously by utilising numerical integration procedure.

Although this approach is able to provide a faster result, but care should be taken when applying this approach. The following points must be properly considered while using this approach.

- Mathematical modelling

Time Domain Simulation approach depends on the mathematical model and the assumptions incorporated in the analysis. Up till now, many mathematical models have been developed around the world. These mathematical models are generally divided into seakeeping approach and manoeuvring approach. As yet, there is no real proof to show that which type of mathematical model is the most suitable one; each mathematical model has its own advantages.

- Non-linearity

In large amplitude and capsizing situation, the equation of ship motion is strongly dominant by many non-linearity terms. Therefore, the use of linear theory (frequency domain) is not suitable to be applied in this case. Consequently, the complicated non-linear equations of motion require the use of Time Domain Simulation approach as a tool to solve this problem.

- Hydrodynamic coefficient

The accurate prediction of the hydrodynamic coefficient such as added mass and damping is a complicated process. It is because of both the added mass and damping strongly depending on the frequency and underwater geometry. The well-known strip theory proposed by Salvesen et al. (1970) is the method frequently used to solve this problem. But, it should be noted that this theory is developed for ship motion in small amplitude and viscous effect is neglected.

- Reliability

Since it is very difficult to cover all the effects in the equations of motions, this approach provides only some trends and solutions of ship safety in certain conditions. Generally, performing experiment is the only proven method to evaluate the reliability of the simulation result.

- Practical application

One of the limitations of the results obtained from the theoretical approach is very hard to be understood by those without the specialized knowledge in this filed. As a result, ship designers usually do not appreciate the physical meaning of the results. In order to avoid this problem, the results should be presented in a form, which is simple and easily understood by the user.

2.10 Seakeeping Assessment

A method of seakeeping assessment proposed originally by the Dubrovskiy in 1978, offer the possibility of taking into account all limitations of speed in waves and other characteristics, including motion amplitudes, acceleration at any point, slamming, deck wetness and bending moments for every type of ship. General seakeeping characteristics are presented as one number for needed sea conditions.

In accordance with common methods of calculations, the assessment method to evaluate seakeeping behaviors due to vertical motion (heave and pitch) is usually conducted by the motion response amplitudes values (RAOs). The RAOs define the amplitude of the response due to a unit wave excitation. Then, Sariöz and Narli, (2005), RAOs are computed for each critical mode of motion, i.e. angular motion or translation of any point on the vessel. The displays RAOs of some experimental results involving pitching, heaving motion, and acceleration, where the horizontal axis denotes the wavelength divided by the ship's length.

In addition, the important investigation of vertical motion effects on the vessels is to apply the seakeeping criteria that emphasized particularly on passenger's comfort. Such criteria would limit the motions of heave, pitch and roll, deck wetness and deck slamming events, and accelerations in the vertical plane. Sariöz and Narli (2005), Kenevissi, et al (2003), and Giron-Sierra, et al, (2002) proved that large pitch motions and accelerations can degrade the operational capabilities of the ship and affect on the comfort and well-being of the passengers. McCreight (1987) and Dubrovskiy (2000) presented specific criteria of twin-hull vessels that may be set as in Table 2.1.

Roll (deg)[a]	8.0
Pitch (deg)[a]	3.0
Vertical acceleration (deg)[a]	0.4
Slams/ hr	20
Wetness/ hr	30
Single amplitudes of vertical acceleration at a point, which placed at 15% length from the bow perpendiculars	0.25 g

[a] Significant Amplitudes

Table 2.1. Seakeeping Criteria in Transit

CHAPTER 3

APPROACH

3.1 General

This chapter puts emphasis on the evaluation of modified design hull form of vessel to minimize the magnitude of vertical motions (pitch and heave motions) and maintain the seakeeping quality. The genetic mutation of hull form design was taken originally from SWATH and conventional Catamaran hull forms (has been done by previous student). The flexible advantage of vessel is able to be operated in variable range of draught operation i.e. shallow draught and deep draught at regular head sea with different wavelengths. The design is readily adaptable to other applications such as passenger vessel, geophysical research, and explorations. The over riding design goal is good seakeeping, both dead-in-the-water and underway.

Furthermore, the detail investigations Semi-SWATH vessel performance especially due to vertical motion at high-speed in head sea waves are presented. Since the Semi-SWATH still has low stiffness (pitch restoring force) than the mono-hull vessels, it result the large pitch motion, which is susceptible to trim changes in waves. Consequently, it may cause deck wetness and makes the vessel unstable or discomfort. While the attached pitch stabilization design (anti-pitching fins) with the variation angle of attacks relative to the hull on the vessel can increase the vessel's stiffness by providing additional damping is a main consideration.

In order to obtain a good prediction of Semi-SWATH vessel performance in step of time, the Time Domain Simulation is used to investigate the motion response of Semi-SWATH vessels traveling in head seas. In addition, the development of mathematical modelling comprising of coupled heave and pitch motion associated with pitch stabilization is enough to identify the vessel behaviors and ensure particularly the pitch stability in real-time simulation. However, an application of ride control motion using conventional PID controller on the controllable stern fins has to take into main consideration to the dynamics of Semi-SWATH motions in which the vessel operates. The main advantage of the ride control system development is able to adapt to every changing environment due to principal disturbances acting on the Semi-SWATH (forces and moments generated by the waves).

Finally, the validation of theoretical simulation will be confirmed by experimental model test in regular head sea waves at Towing Tank of Marine Technology Laboratory, Universiti Teknologi Malaysia.

3.2 Framework of Study

The framework of study is divided into seven phases. Explanation of each phase is summarized as follows.

- Problem Identification

The first phase of the research is to encourage on the modified twin-hull vessel design, which lead to reduction of magnitude vertical motion when traveling in head seaway at high-forward speed. As mentioned in the previous chapter, this is the most critical operation condition for twin-hull particularly Semi-SWATH vessel, where the vessels is usually excited to large pitch motion.

- Time Domain Simulation approach

The second phase of the research is to discuss the Time Domain Simulation approach as a tool to describe the seakeeping performance of the Semi-SWATH vessel in head seas at high-speed. This involves the discussion of the equations of motion and also the assumptions taken into account in the mathematical model.

- Validation of the Simulation Program

The third phase is to validate the simulation program results, which conducted by the experimental Semi-SWATH model of 2.31m length carried out in Towing Tank at Marine Technology Laboratory, Universiti Teknologi Malaysia. In this phase is to present the validation between simulation and experiment prediction results in a comprehensive way.

- Parametric Study

The fourth phase is to conduct parametric study. The importance of this part is to investigate the effect of attached fin stabilizer on fore and aft body to the vertical motion especially pitch motion in waves. The main parameter inherent to this investigation is determined the controllable stern fin's angle accurately to minimize the pitch angle of vessel. Then, parametric study is carried out on the Semi-SWATH vessel for two draught conditions with different speeds both simulations and experiments. For the detail of this parametric study can be seen in Table 3.1 and 3.2. It defined the comparison between simulations and experiments correspondingly.

- Investigation of the controller scheme of the fin stabilizer on the vessel performances

The fifth phase of the research is to investigate control scheme of fin stabilizer as the fixed fins and controllable fins. In this phase, divided in two parts as follows;

a) Fin with constant inclined angle relative the main hull body

This case means that the stabilizer fin is adjusted manually to have an inclined angle relative to the ship body. The total lift on the ship is then due to the resultant of angle of attack of incident flow to the body and the inclined angle of fin. This technique can be applied to adjust the trim condition of Semi-SWATH vessel advancing in waves with initial trim by bow or stern.

b) Time-Varying Controllable Fin

The purpose of this technique is to reduce pitch motion of the Semi-SWATH vessel advancing in waves. The principle is to adjust the fin lift force to balance the force due to ship motion and hence reduce the pitch motion effectively. By controlling the aft fins properly, the sailing style of Semi-SWATH vessel may be adjusted to be in even keel condition.

From the point's view of the automatic control, it seems not so easy to control a suitable angle of attack to reduce the pitch motion because the pitch motion is non-linear, time-varying, and coupled with other factors, e.g. heave displacement, heave velocity, heave acceleration, and pitch displacement, pitch velocity, pitch acceleration, etc. If the limitation of speeds, sea state, angle of attack, and control type are included, it will be come even more complicated and cannot be achieved by an easy and simple control system. Therefore, the systematic procedure of the control rule, which can be used to reduce the pitch motion, is investigated.

Generally the controlling technique of controllable fin was applied to find an easy and direct way reducing the pitch motion effectively by controlling the fin stabilizer so as to produce the lift, which opposes the sense of the pitch mode. This control technique can be achieved by the controllable rule for aft fins i.e. change the angle of aft fin relative to the main hull body, as follow;

<Aft Fin control>:

$$\left(\alpha_B\right)_{aft} = \begin{cases} \operatorname{sgn}\left(\dot{\eta}_5\right)\cdot\left[\left(\delta_B\right)_{aft}\right] & \text{if } \dot{\eta}_5 \neq 0 \\ 0 & \text{if } \dot{\eta}_5 = 0 \end{cases}$$

- The Assessment of the PID Control Tuning

The sixth phase of the research is to further investigate the application of the PID controller on the adjustable stern anti-pitching fins on the Semi-SWATH vessel. In this phase, a controller is only as effective as its tuning**:** the adjustment of the PID settings relative to the process parameters to optimize load and set-point response. Where, controller tuning involves the selection of the best values of kc, Ti and T_D.

Furthermore, in order to obtain initial controller gain parameters of PID (K_p, K_i, *and* K_d are the proportional, integral, and derivative gains, respectively), the Astrom-Hagglund method based on a relay feedback controller is carried out to attain the critical period of waveform oscillation (T_c) and critical gain (K_c).

- Evaluation of the Seakeeping Performance

The seventh phase is to present the evaluation of seakeeping performance in all ship conditions considerably. An assessment of seakeeping characteristic inherently produces large volume of data. Therefore, it is often necessary to select certain parameter of the vessel for the purposes of design evaluation. The governing criteria may vary greatly with differing vessel type and application. Moreover, they must be appropriately selected based on the intended utilization of the design.

During simulations and experiment, the ship model without and attached fin stabilizers incorporated with the PID controller put as a focus mainly on attenuation of the pitch and heave position motion responses. In this thesis, the identified seakeeping performance will be evaluated based on Response Amplitude Operators (RAOs). For that reason, the two parameters were selected to investigate seakeeping performances that were measured, as follows;

1. Significant heave amplitude
2. Significant pitch amplitude

3.3 Choosing a Systematic Procedure

The main reason to choose a systematic procedure is to ensure that an objective design of the Semi-SWATH vessel development can be achieved properly. These choosing systematic procedures will guideline for the vessel that will provide a better seakeeping quality.

The adopted systematic procedure to implement the frame work of study is divided into three main parts. There are the selection of parameters, parametric study, and evaluation of the motion criteria in term of RAOs. The descriptions of the main aspect in the simulation procedure are shown in the following sections.

3.3.1 Selection of Parameters

The parameters involve in the study are categorized into three groups, namely: environmental parameters, vessel geometry, loading condition and operation parameters.

- Environmental Parameters

The environmental parameters represent the external forces acting on the vessel. Generally, wave, current, and wind are the main components usually considered in the seakeeping aspects analysis. However, only the wave forces are interested in the present study. This is because of the wave force being the main component due to vessel suffered some uncomfortable ride and structural damage.

The waves parameters such as wave height, wave length and wave speed are selected in the group of environmental parameters. The motions of the model in regular wave were measured for any interest of wavelength and wave height correspondingly. The commonly used wave height to wavelength ratio on experiments was 1/50 (as the recommendation of the ITTC and DTNSRDC). For a slender Semi-SWATH hull, as the present one, ship responses are assumed to be linear for wave steepness of 1/50 or less. In addition, the range of wavelengths covered in the experiments was restricted to 0.5 to 2.5 times the ship’s length. It expected the peak value of Response Amplitude Operators point at certain frequency as closed as the natural ship frequency ω_n both of heaving and pitching will be achieved.

- Vessel Loading Condition

Both in the simulation and experiment, the vessel loading condition is set up for two draught operation modes i.e. shallow draught and deep draught. The distinguishing of Semi-SWATH vessel draught is sufficient allowance for ballast transfer that made to enable the ship to vary its draught under load conditions. Technically, the number of sea water weight is used to ballast and deballast of Semi-SWATH vessel considerably.

3.3.2 Parametric Study

The importance of parametric study is to give a better understanding of the influence of parameter on the seakeeping performance. This helps in the improvement design in view of vessel comfortability. In this study, pitch angle is taken as a reference to be controlled to avoid undesired pitch angle. In the worst case, the severe pitch angle causes all way to be lost and a severe bow down becoming inundated forward can be alleviated. Therefore, it is very important to determine the minimizing of pitch angle inherent to the environmental parameters and also the ship conditions. It can be done by control and check the occurred maximum point pitch angle during operations. However, the results can be used as guideline for the vessel during operations.

The detail parametric investigation that will be taken to reduce the undesired pitch angle was examined by fixed and controllable fins angle. The all parameter works will be studied in two sections or conditions either simulation or experiment as follow;

a) First section; bare hull vessel condition then simultaneously attached fixed fore fins and adjustable aft fin (manually). Furthermore, bare hull with fixed fore fin and controllable aft fin using PID controller will be applied. Both the simulation programs and experiments will be done to study an effective and reliable angle that lead to reduction heave and pitch angles. The detail of this parametric study was summarized in Table 3.1.

b) Second section; bare hull vessel after attached fixed fore fins and controllable aft fins (automatically). In this case, only the simulation program incorporated with the conventional PID controller will be studied properly. The main goal of the second section is to achieve the pitch angle as minimum as possible with consider the some aspects design such less resistance and passenger's comfortability. The detail of second section of parametric study can be seen in Table 3.1.

On the second section of parametric work will involve some chosen controller gain parameters of PID (*K_p, K_i, and K_d* are the proportional, integral, and derivative gains, respectively). The detail of this parametric study was summarized in Table 3.2.

Table 3.1 The two sections of parameter works

Section Case No	*S	**E	Vessel Condition	Fixed Fin Angle (Degree)		T (M)	Vs (Knots)
				Fore	Aft		
1	√ √	√ -	1. Bare hull then bare with fixed fore and aft fin 2. Bare hull with fixed fore fin and controllable aft fin using PID controller	15	5, 10,15 20,and 25	2.0	15
2	√ √	√ -	1. Bare hull then bare with fixed fore and aft fin 2. Bare hull with fixed fore fin and controllable aft fin using PID controller	15	5, 10,15 20,and 25	1.4	20

*S for simulation
**E for Experiment

Table 3.2 The parameter of PID tuning on the second section of parameter works

Case No	PID Controller Gain		Vessel Condition	Fin Angle (Degree)		T (M)	Vs (Knots)
	Inner Controller	Outer Controller		Fixed Fore (^{0})	Saturation of Aft fin angle (^{0})		
1 2	K_p and K_d K_p and K_d	K_p, K_d & K_i K_p, K_d & K_i	Bare hull with controllable aft fins	15 15	0 to 25 0 to 25	2.0 1.4	15 20

3.3.3 Evaluation of Motion Response

After implementation the systematic procedures with gained the first and second parametric studies, the results will be evaluated based on the Response Amplitude Operators (RAOs) that plotted the each points again the ratio of $\frac{\lambda}{L_s}$. The simple formulation of the RAOs is derived by dividing the motion amplitude by the wave amplitude;

$$\frac{u_a}{\xi_a} = \frac{\text{motion amplitude}}{\text{wave ampitude}}$$

3.4 Concluding Remarks

The procedure to develop pitch improvement of Semi-SWATH vessel operating in head seas is presented. Emphasis of the systematic procedure is to enable achieve the desired pitch angle as minimum as possible that provide some advantages not only in view of comfortability but also in safety. This procedure consists of seven important phases. There are: problem identification, Time Domain Simulation approach, validation of the simulation program, parametric study, the assessment of the PID Control Tuning, and evaluation of the Seakeeping Performance. Both the simulation and experiments results may be the basis for the proposed comfort and safe design guidelines operation mainly for Semi-SWATH vessel.

CHAPTER 4

MATHEMATICAL MODEL

4.1 General

This chapter presents the formulation for the hydrodynamic forces and moments acting on a twin-hull vessel equipped with control surfaces, and its motion response in step of time-domain. This formulation was developed to predict the ship motions using strip theory. *Korvin-Kroukovsky and Jacobs* (*1957*) presented the original strip theory especially suitable for numerical computations of heave and pitch motions in head waves. A numerical study using a computer program based on this formulation is also included to investigate the accuracy of the program.

The main objective of the chapter is to establish a backbone of a numerical tool for the mathematical modeling of a ship and control surfaces in step of time-domain with a reasonable accuracy. This model will provide a basis for a more sophisticated time-domain model of the motion responses in the presence of active control surfaces to be developed in chapter 5.

Therefore, in the following section (4.2) a re-statement of the formulation for the hydrodynamic forces and moment (i.e. added mass, damping, and wave-excitation) on a bare-hull is presented in the context of the two-dimensional linear strip theory. The formulation of the forces and moment due to control surfaces is presented in section 4.3 based on available semi-empirical methods. The formulation of the linear motion responses for the coupled heave and pitch mode is given in section 2.4.

4.2 Formulation of Hydrodynamic Forces and Moments Based on Strip Theory

In order to investigate the seakeeping characteristic and motions of a ship moving in a seaway, a linear strip theory has been recognized as the most practical numerical tool. The theory used is based on the assumptions of potential flow, slender ship, and small amplitude of motions. The potential flow implies a homogeneous, incompressible, and inviscid fluid, defined as the ideal fluid, with n surface tension. The potential flow field is considered to model a particular flow generated by the existence and forward motion of a ship traveling in sea waves.

The above assumptions are considered to find a gradient of scalar function Φ to denote the velocity potential of the generated flow. Lapalce's equation may be set up to describe the potential flow and solved by means of the initial boundary conditions described in the following sections.

4.2.1 Co-ordinate system

The ship is assumed rigid and undergoing six degrees-of-freedom oscillations when traveling at a constant mean forward speed $\overset{\text{u}}{U}$ with arbitrary heading in regular waves. In the mathematical formulation, three different co-ordinate systems are necessary for expressing the forces and moments acting on the vessel, and for solving equations of motion. It is assumed that the oscillatory motions are linear and harmonic. Let (x, y, z) be a right-handed coordinate system fixed with respect to the mean position of the ship with a vertically upward through the center of gravity of the ship, x in the direction of forward motion (see figure 2.2.1). These are as follows;

1. the earth fixed system is defined by $O^e x^e y^e z^e$, where the z^e axis directs vertically upwards and $O^e x^e y^e$ plane lies in the still waterplane. The earth

fixed co-ordinate system is used to express the free surface boundary condition.

2. the ship system described by $O^*x^*y^*z^*$ is fixed in the ship with the origin O^* being located at the centre of gravity of the vessel. x* is directed forward (towards the bow), z* is directed upward at right hand angles to O*x*y* and y* is directed to starboard at right angles to O*x*z*. The system is used to
3. derive the boundary condition on the ship's wetted surface.
4. the vessel may have forward speed, Vs, and a third system; translate in the Oxz plane with the vessel at its mean speed. In this system, x points in the mean forward direction of the vessel, that is along the x* axis and z points vertically upwards. The steady moving co-ordinate system is an inertial reference frame in which the motions are periodic.

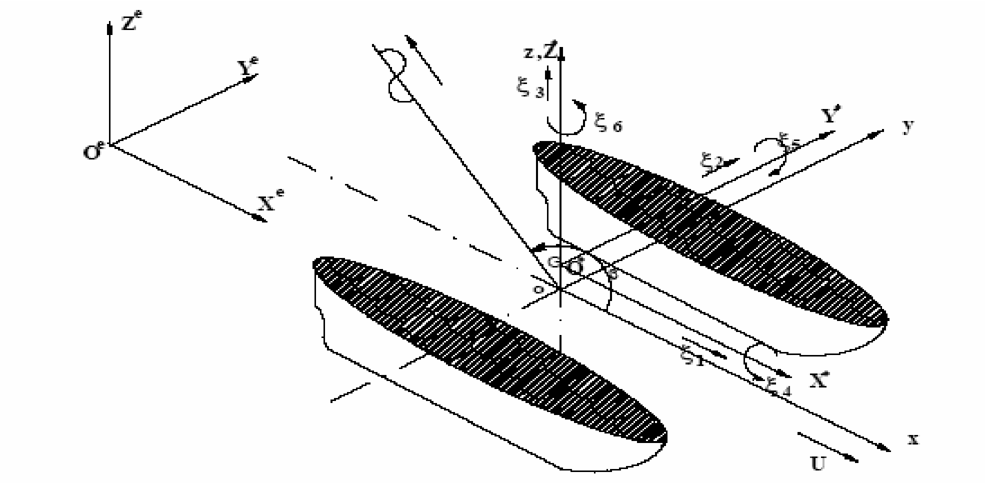


Figure 4.1. Definition of vessel's coefficient-ordinate system

4.2.2 Boundary Conditions and Linearization

The flow field disturbed by the existence and forward motion of the ship in waves is assumed to be ideal. Therefore, the velocity potential of the flow field, ϕ (x,y,z,t), must satisfy Laplace's equation in the fluid domain. Moreover, the velocity potential has to satisfy the equation of continuity in all points of the flow field except at singular points. The boundaries enclosing the fluid domain consist of the wetted body surface, the free surface, the seabed and a control surface at the far-field. It is assumed that the sea is infinite in all horizontal directions and in depth. The velocity potential must satisfy the conditions imposed by these boundaries.

- **Free surface condition**

On the free surface, the kinematic and dynamic boundary conditions imply that a fluid particle at the surface always remains at the surface and the pressure on the free surface is atmospheric and satisfies Bernoulli's equation. Based on the assumption of ideal fluid and neglecting the surface tension at the free surface, the dynamic and kinematic boundary conditions can be combined on the free surface and written assumption (Newman, 1978)

$$\left(\frac{\partial}{\partial t}\text{-}V\frac{\partial}{\partial x}\right)^2\Phi+2\nabla\Phi\left(\frac{\partial}{\partial t}\text{-}V\frac{\partial}{\partial x}\right)\nabla\Phi+\frac{1}{2}\nabla\Phi\nabla\left(\nabla\Phi\nabla\Phi\right)+g\Phi_z=0 \qquad (4.1)$$

at $z=\zeta$ the free surface

- **Body surface condition**

The kinematic body boundary condition indicates that the fluid velocity component normal to the instant submerged body surface is equal to the velocity component of the surface normal to itself. If $\vec{V}$ denotes the velocity of the flow and $\vec{V}_s$ denotes the local velocity of a point on the wetted body surface, therefore:

$$\vec{V}\cdot\vec{n}=\vec{V}_S\cdot\vec{n} \qquad \text{on wetted body surface, } S_b \qquad (4.2)$$

where $\vec{n}$ is the unit vector.

- **Sea bed condition**

The sea bed boundary condition implies that the velocity component of fluid particle normal to the seabed is zero since the seabed is assumed to be stationary. Thus

$$\frac{\partial\phi}{\partial n}=\frac{\partial\phi}{\partial z}=0 \qquad \text{on the sea bed } z=\text{-}\infty \qquad (4.3)$$

- **Radiation condition**

In order to ensure having a unique, a radiation condition at infinity is necessary to be considered. The far-field radiation condition signifies that the fluid motion generated by the steady oscillatory body travels away from the body and vanishes at infinity in all directions.

Since the boundary condition are highly non-linear, a perturbation expansion scheme should be adopted. Furthermore by assuming that the vessel is slender, the derivatives generated around the vessel will be small up to moderately high speeds. Therefore, the derivatives can be expanded into a series form given by:

$$D \cong \varepsilon^0 D^{(0)} + \varepsilon^1 D^{(1)} + \varepsilon^2 D^{(2)} + \ldots \tag{4.4}$$

where D indicates all derivatives around the vessel,

$\varepsilon = B/L$ is slenderness ratio,

B and L are the beam and the length of vessel respectively,

0,1 and 2 indicate the zeroth, first and second order respectively.

For the problem of a traveling vessel in waves, based on linearization, the total velocity potential can be separated into three components given as:

$$\Phi(x,y,z,t) = -V_x + \bar{\phi}(x,y,z) + \tilde{\phi}(x,y,z,t) \tag{4.5}$$

The first term on the right hand side of equation (4.5) is due to the forward speed. The second term is called the steady perturbation potential due to the forward motion of the vessel. The last term is the unsteady velocity potential due to the incident and diffracted waves and the induced motions of the vessel in 6 degrees of freedom.

Based on the above assumptions, the first order unsteady potential in equation (4.3.5) can be expressed as:

$$\bar{\phi} = \left[\phi_0 + \phi_7 + \sum_1^6 \eta_j \phi_j \right] e^{-i\omega t} \tag{4.6}$$

η_j indicates the complex amplitude of the motion responses.

Where the first term is the first order incident wave potential assumption given by:

$$\phi_0 = -\frac{ig\zeta_a}{\omega_0} e^{k_0 z} e^{ik_0 (x \cos\beta + y \sin\beta)} \tag{4.7}$$

The second term is the diffraction potential, and the last term is the radiation potential due to the j^{th} mode of motion ($j=1$ denotes surge motion, $j=2$ for sway, $j=3$ for heave, $j=4$ for roll, $j=5$ for pitch, $j=6$ for yaw). A vessel advancing with constant speed V at an arbitrary angle β (180^0 for head waves) experiences waves of encounter frequency, ω, defined by

$$\omega = \omega_0 - Vk_0 \cos\beta \tag{4.8}$$

where $k_0 = \omega_0^2/g$ is the wave number, ω_0 and g denoting the wave frequency and gravity acceleration respectively.

In order to calculate the pressure and forces generated on the vessel, a linearization as well as some simplifications of the boundary conditions is required to solve effectively the problem. Assuming the perturbation of the steady flow due to the presence of the body is negligible, the unsteady motion problem can then be treated independent of the steady motion problem in calm water. Further, this assumption leads to the fact that the radiation potential on the body surface can be written as Newman (1978):

$$\begin{aligned} &\frac{\partial\phi}{\partial n} = -\omega n_j + Vm_j \\ &\vec{n} = (n_1, n_2, n_3) \\ &\vec{r} \times \vec{n} = (n_4, n_5, n_6) \\ &m_j = 0 \ \text{ for } j = 1,2,3,4 \\ &m_5 = n_3 \text{ and } m_6 = -n_2 \end{aligned} \tag{4.9}$$

with $\vec{n}$ the outward unit normal vector and $\vec{r}$ the position vector with respect to the origin of the reference system.

Considering that the radiation and diffraction problems are independent of each other, the following relationships holds on the body surface

$$\frac{\partial \phi_7}{\partial n} = -\frac{\partial \phi_0}{\partial n} \quad \text{on the fixed body surface} \tag{4.10}$$

Where ϕ_7 represents the diffraction potential while ϕ_0 is the incident wave potential. Neglecting terms of second order or more on the free surface boundary condition (4.1), and further assuming that the forward motion is small and the frequency of oscillation is high, the equation is reduced to the simple form:

$$-\omega^2 \phi + g\frac{\partial \phi}{\partial z} = 0 \qquad \text{on } z = 0 \tag{4.11}$$

However, in the low frequency range, the forward speed effect on the free surface makes the theory somewhat arguable. If the encounter frequency ω is replaced by wave frequency ω_0, this equation becomes the zero speed free surface boundary condition. For the forward speed problem, equation (4.11) can be used together with equation (4.9) as an approximate method.

4.2.3 Application of Strip Theory

According to the strip theory rules, the vessel is split into many strips along its length and the hydrodynamic interaction between the strips in the longitudinal direction is neglected. It is assumed that the vessel is sinusoidally oscillating in small amplitude incident waves with small response amplitudes. Based on the described assumptions and the strip theory, the three-dimensional Laplace's equation and boundary conditions can be reduced to two-dimensional problems. Applying Green's second identity and using the Frank's Close-Fit method, Frank (1967), one can describe a two-dimensional velocity potential of the fluid. Figure 4.2 shows the general fluid domain for a twin-hull vessel.

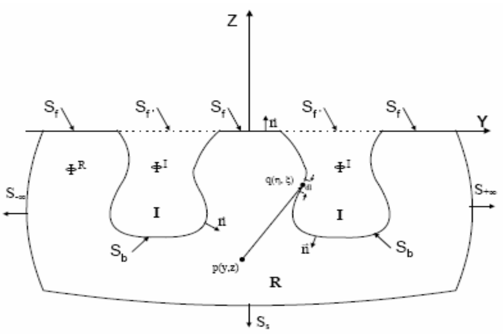


Figure 4.2. Description of defined boundaries fluid for twin-hull vessels

The fluid domain is divided into two distinct regions:

(a) **I**, which is inside the twin-hull contour enclosed by the body contour (S_b), and the free surface (S_f) at the body sections;

(b) **R**, which is exterior to the body, enclosed by the free surface (S_f), the body contours (S_b), the far-field control surfaces $S_{\pm\infty}$, and the seabed S_s at the bottom.

ϕ^R and ϕ^I are the velocity potentials in the R and I regions respectively.

Green's second identity for the described system can be written as Atlar, (1985)

$$\int_{S_b}\left(\phi_j\frac{\partial G}{\partial n_q}-G\frac{\partial\phi_j}{\partial n_q}\right)dl=\iint_{R+I}\nabla\cdot\left(\phi_j\nabla G-G\nabla\phi_j\right)ds=\iint_{R+I}\nabla\cdot\left(\phi_j\nabla^2 G-G\nabla^2\phi_j\right)ds \quad (4.12)$$

$\nabla=\frac{\partial}{\partial x}\vec{i}+\frac{\partial}{\partial y}\vec{j}+\frac{\partial}{\partial z}\vec{j}$	Vector differential operator
$q=\eta+i\xi$	Source point
$\frac{\partial}{\partial n_q}$	Normal derivative with respect to the source point q
$\vec{n}$	Outward unit normal vector
G	Two dimensional Green's function

Wehausen and Laitone (1960) defined the two-dimwnsional Green's function of a pulsating source, which satisfies the linearised boundary conditions, as:

$$G(p,q)=\operatorname{Re}\left\{\log\left(\frac{p-q}{p-\bar{q}}\right)+2PV\int_0^\infty\frac{e^{-i\nu(p-\bar{q})}}{K-\nu}\right\}-2\pi i\operatorname{Re}\left\{e^{-iK(p-\bar{q})}\right\} \quad (4.13)$$

Where

$p=y+iz$	Field point
ν	Real variable
PV	Denotes Principal Value of an integral
$\bar{q}$	Complex conjugate of q

As ϕ_j and G must satisfy Laplace's equation in the fluid domain except at singular point, equation (4.12) can be reduced to:

$$\int_{S_b} \left(\phi_j \frac{\partial G}{\partial n_q} - \frac{G \partial \phi_j}{\partial n_q} \right) dl = 0 \tag{2.14}$$

This equation has three different characteristics, which depend on the relative position of the field point p to the source point q, described as follows:

(a) The field point p is inside the **I** region

(b) The field point, p, lies is inside the fluid domain **R**

(c) The field point, p, lies on the boundary contour $\mathbf{S_b}$

Let the source point, q, approach the field point, when p=q, the Green's function G becomes singular and makes equation (4-13) invalid. This problem can be avoided by introducing a small circle of radius ε with contour S_ε and with its origin located in R domain. Then $S_b + S_\varepsilon$ is a closed contour surrounding the fluid domain but exterior to S_ε. Defining

$$\sigma(q) = \frac{\partial \phi_j^R(q)}{\partial n_q} - \frac{\partial \phi_j^I(q)}{\partial n_q} \tag{4.15}$$

as the source strength, the velocity potential can be written in the following form.

$$2\pi \phi_j(p) = \int_{S_b} \sigma(q)\, G(p,q) dl \tag{4.16}$$

Thus, the velocity potential is an integral of a source distribution of strength σ(q) over the body contour and the Green's function defined in equation (4-13).

In order to determine the source strength σ(q), the linearized body conditions (4.-9) and (4-10) can be applied to equation (4-16). Therefore

$$\pi\sigma(p) + \int_{S_b} \sigma(q) \frac{\partial G(p,q)}{\partial n_q}\ dl = 2\pi \frac{\partial_j(p)}{\partial n_p}$$

Where, the first term on the left hand side avoids the singularity problem.

The next step after calculating the source strengths is the velocity potential, which can be determined by equation (4-16). Then, the fluid pressure, P, can be obtained by substituting the velocity potential into unsteady bernoulli's equation,

$$P= -p\left[gz + \tfrac{1}{2}(\nabla\phi)^2 + \left(\frac{\partial}{\partial t} - V\frac{\partial}{\partial x}\right)\phi \right] \qquad (4.18)$$

and the fluid force, F, acting on the wetted surface, S_b, as follows:

$$\vec{F} = \int_{S_b} P\vec{n}ds \qquad (4.19)$$

Where, the normal vector, $\vec{n}$, directs outwards from the body surface.

4.4.4 Hydrodynamic Forces and Moments

In linear theory, the total velocity potential is the superposition of incident velocity potential ϕ_I diffraction velocity potential ϕ_D and radiation velocity potential ϕ_R therefore in order to calculate the hydrodynamic forces and moments, the appropriate velocity potential should be substituted into the linearised Bernoulli's equation and then integrated over the mean wetted surface of the vessel.

If the pressure, given by equation (4.18), is expanded in a Taylor series about thr undisturbed position of the hull and considering only terms to first order in steady potential $\bar{\phi}$ and unsteady potential $\tilde{\phi}$ the linearised time-dependent pressure on the hull Salvesen et al. (1970) can be written as:

$$P = -\rho\left(i\omega - V\frac{\partial}{\partial x}\right)\tilde{\phi}e^{i\omega t} - \rho g\left(\eta_3 + \eta_4 y + \eta_5 x\right)e^{i\omega t} \qquad (4.20)$$

The last term gives the ordinary buoyancy restoring forces and moment. Integration of the pressure (ignoring the buoyancy term) over the hull surface yields the hydrodynamic forces and moment amplitudes:

$$H_j = -\rho\iint_{S_b} n_j\left(i\omega - V\frac{\partial}{\partial x}\right)\tilde{\phi}ds \qquad j=1, 2, \ldots, 6 \qquad (4.21)$$

By considering

$$\tilde{\phi} = \phi_I + \phi_D + \sum_{j=1}^{6} \eta_j \phi_j \tag{4.22}$$

the force and moment can be divided into two parts as

$$H_j = F_j + G_j \tag{4.23}$$

Where F_j is the exciting force and moment:

$$F_j = -\rho \iint_{S_b} n_j \left(i\omega - V\frac{\partial}{\partial x} \right) \left(\phi_I + \phi_D \right) ds e^{-i\omega t} \tag{4.24}$$

and G_j is the force and moment due to six degrees of body motion:

$$G_j = -\rho \iint_{S_b} n_j \left(i\omega - V\frac{\partial}{\partial x} \right) \sum_{k=1}^{6} \eta_k \phi_k ds = \sum_{k=1}^{6} T_{jk} \eta_k$$

Here T_{jk} denotes the hydrodynamic forces and moment in the j^{th} direction per unit oscillatory displacement in the k^{th} mode:

$$T_{jk} = -\rho \iint_{S_b} n_j \left(i\omega - V\frac{\partial}{\partial x} \right) \sum_{k=1}^{6} \phi_k ds \tag{4.26}$$

After separating T_{jk} into real and imaginary parts as:

$$T_{jk} = \omega^2 A_{jk} - i\omega B_{jk} \tag{4.27}$$

the added mass and damping can be defined in the form

$$A_{jk} = \frac{-\rho}{\omega^2} \mathrm{Re} \iint_{S_b} n_j \left(i\omega - V\frac{\partial}{\partial x} \right) \sum_{k=1}^{6} \phi_k ds \tag{4.28}$$

$$B_{jk} = \frac{-\rho}{\omega} \mathrm{Im} \iint_{S_b} n_j \left(i\omega - V\frac{\partial}{\partial x} \right) \sum_{k=1}^{6} \phi_k ds \tag{4.29}$$

4.3 Modelling of Fin Effect

The forces and moments generated by the presence of fin stabilizers can be represented in term of the motion equation coefficients and external forces terms as,

$$F_i^f = \sum_j \left[A_{ij}^f + M_{ij}^f \right] \ddot{\eta}_j + B_{ij}^f \dot{\eta} + C_{ij}^f \eta - \left[F_i^{fe} + F_i^k \right] \tag{4.30}$$

where superscript is associated with fins, e the wave-excitation, and k the control forces due to the activation of the fins.

The forces and moments on n pairs of fins, can be assumed to have three component, which are associated with the lift, L_{in} drag, D_{in} and virtual inertia effects, I_{in} ,of the surrounding flow. Therefore,

$$F_i^f = \sum_{n=1}^{N} \left[L_{in} + D_{in} + I_{in} \right] \tag{4.31}$$

The lift component generated by the quasi-static pitch of the hull and deflection of the fins can be written as

$$L_{in} = \frac{\rho}{2} V^2 A \left\{ C_{L\alpha}\alpha + C_{L\delta}\delta \right\} \tag{4.32}$$

where ρ is the fluid density, V is the fluid velocity, A is the projected fin area, $C_{L\alpha}$ is the lift coefficient of the fin due to hull pitch angel α , the fin being considered fixed to the hull, and $C_{L\delta}$ is the lift coefficient due to control deflection δ with respect to the hull. The values of the lift coefficients based on the slender body theory for a single fin attached to a circular hull, Caldeira-Saraiva and Clarke (1988), are given by

$$C_{L\alpha} = \left[K_{W(B)} + K_{W(B)} \right] \left[C_{L\alpha} \right]_W \tag{4.33}$$

$$C_{L\delta} = \left[k_{W(B)} + k_{W(B)} \right] \left[C_{L\alpha} \right]_W \tag{4.34}$$

Where, the subscripts B and W represent Body (or hull) and Wing (or fin) respectively. The fin-hull interaction factors, K for a fixed fin and k for an activating fin, are devined by the following relationships Kuerti et al (1952),

$$K_{W(B)} + K_{W(B)} = \frac{1}{2}\left[1 + \frac{a}{s} \right]^2 + 2\frac{a}{s} \tag{4.35}$$

And

$$k_{W(B)} + k_{W(B)} = \frac{2}{\pi(\lambda-1)^2} \left\{ \frac{1}{2} \left[\lambda^{3/2} - \lambda^{-3/2} + 7\lambda^{-1/2} - 7\lambda^{1/2} \right] + \right. \tag{4.36}$$

$$\frac{(\lambda+1)^4}{4\lambda^2}\left[\frac{\pi}{2}+\sin^{-1}\left[\frac{\lambda-1}{\lambda+1}\right]\right]-2\pi\Bigg\}$$

Where, $\lambda=\frac{s}{a}$ and s is the span of the fin including the hull radius, a, (see Figure 4-3 and Figure 4-4), Denny (1988).

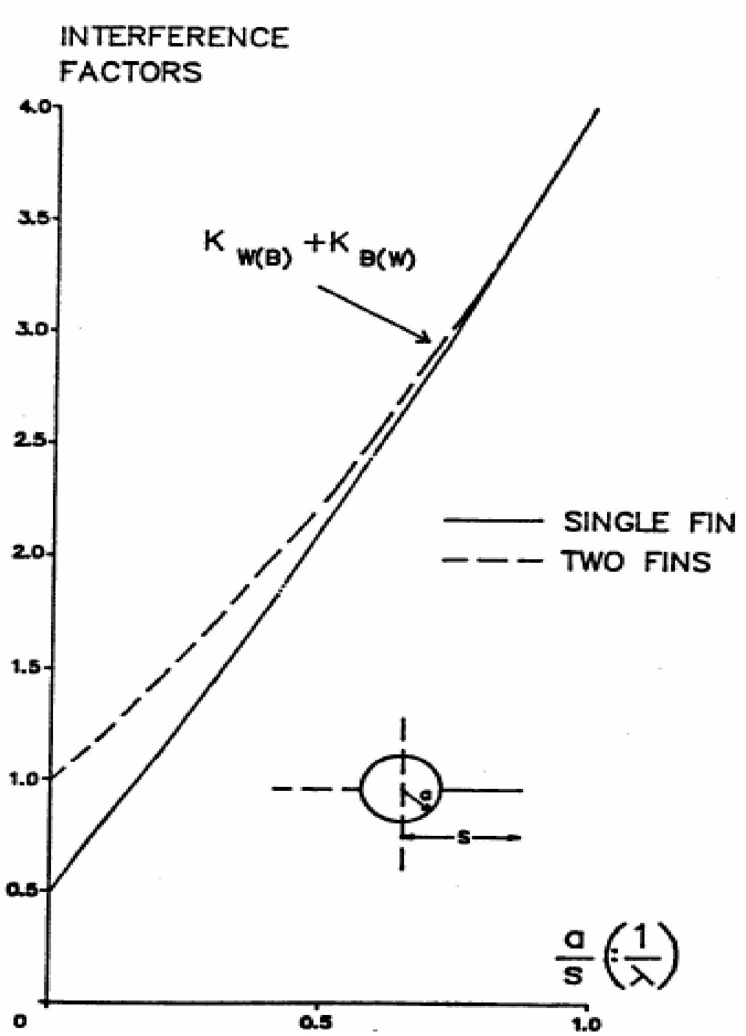


Figure 4.3. Comparison of Wing-Body interaction Factors for Fixed Fin (Caldeira-Saraiva and Clarke, 1988)

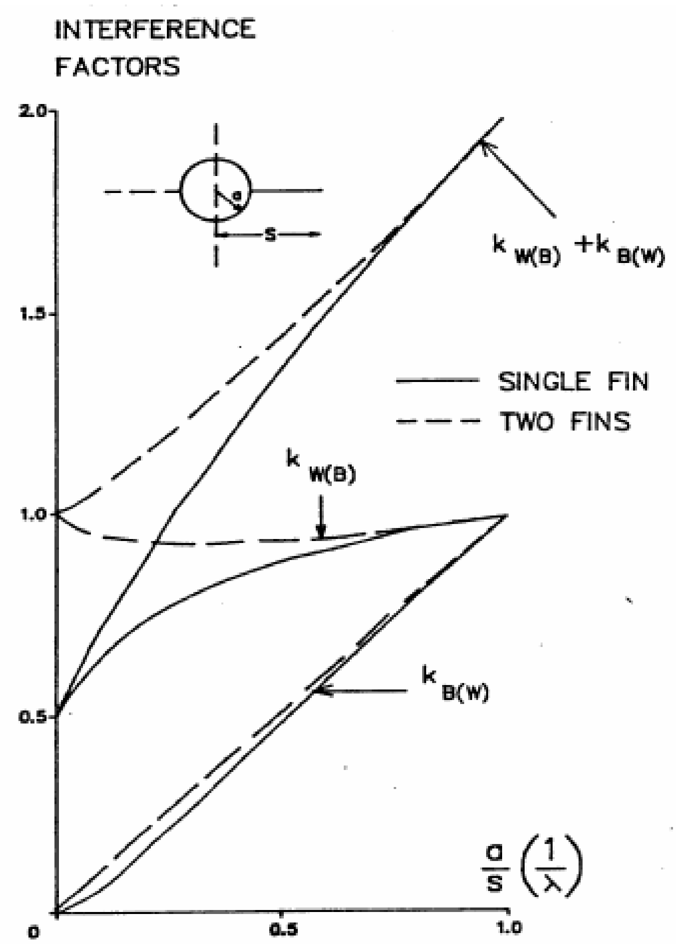


Figure 4.4. Comparison of Wing-Body interaction Factors for Movable Fins (Caldeira-Saraiva and Clarke, 1988)

The $\left[C_{L\alpha}\right]_W$ is the lift coefficient for the fin with a zero sweepback angle in free stream, given by Whickers and Fehlner (1985) as

$$\left[C_{L\alpha}\right]_W = \frac{1.8\pi(AR)}{1.8+\sqrt{\left[(AR)^2+4\right]}} \quad \text{(per radian)} \qquad (4.37)$$

Where, AR is the aspect ratio of the fin

Besides the effect of fin-hull interaction, there will be losses in the free stream lift coefficient due to the hull boundary layer, which can be represented by the following relationship given by Lloyd (1976),

$$E_{BL} = \frac{\text{Lift developed in boundary later}}{\text{lift in free stream}} = 1.0\text{-}0.21\frac{\delta}{S} \qquad (2.38)$$

The boundary layer thickness δ on the hull, Figure 2.5, may be obtained by

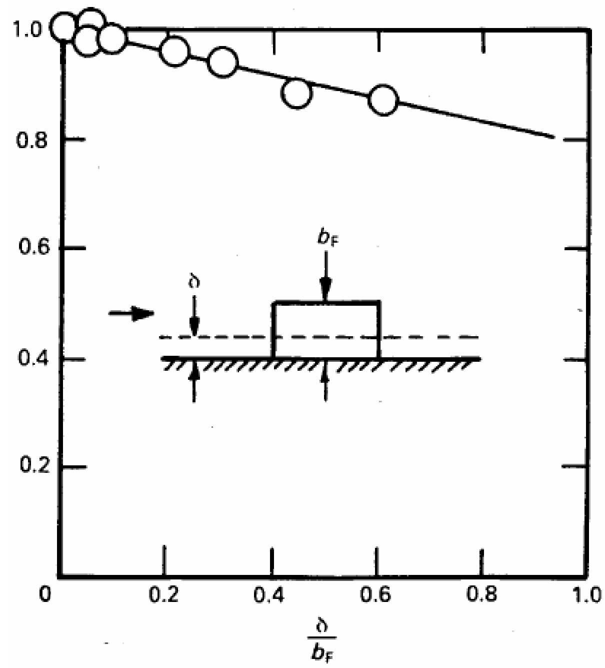


Figure 4.5. Effect of Boundary Layer on Fin Lift, Lloyd (1989)

$$\delta = 0.377 X_{FP} (R_n)^{-0.2} \tag{4.39}$$

where X_{FP} is the distance from the ship forward perpendicular to the fin axis and R_n is the Reynolds number based on X_{FP} and the flow velocity .

Fin stabilizers, like all lifting surfaces, operate by developing a pressure difference between their upper and lower surfaces. The fluid tends to circulate round the tip of the fin from the high pressure to the low-pressure surface and a vortex is formed. Figure 4.6 shows the vortex generated by a fin at a fixed angle of attack α. This vortex is shed from close to the tip of the fin and trails away along the side of the hull giving a swirling motion to the water close to the hull. This causes a "downwash" in the region between the vortex and the hull surface and an "upwash" in the region out board of the vortex.

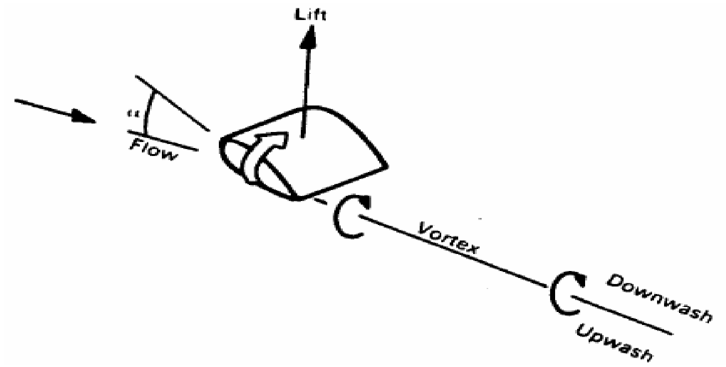


Figure 4.6. Trailing Vortex Generated by a Lifting Surface, Lloyd, (1989)

An oscillating fin produces a vortex of continually varying strength and direction, which is convected away along the side of the hull. In fact the vortex is a record of the lift developed by the fin. Figure 4.7, illustrates the flow behind an oscillating fin and it can be seen that there are alternate regions of downwash and upwash in the wake of the fin, depending on the lift developed in the immediate past.

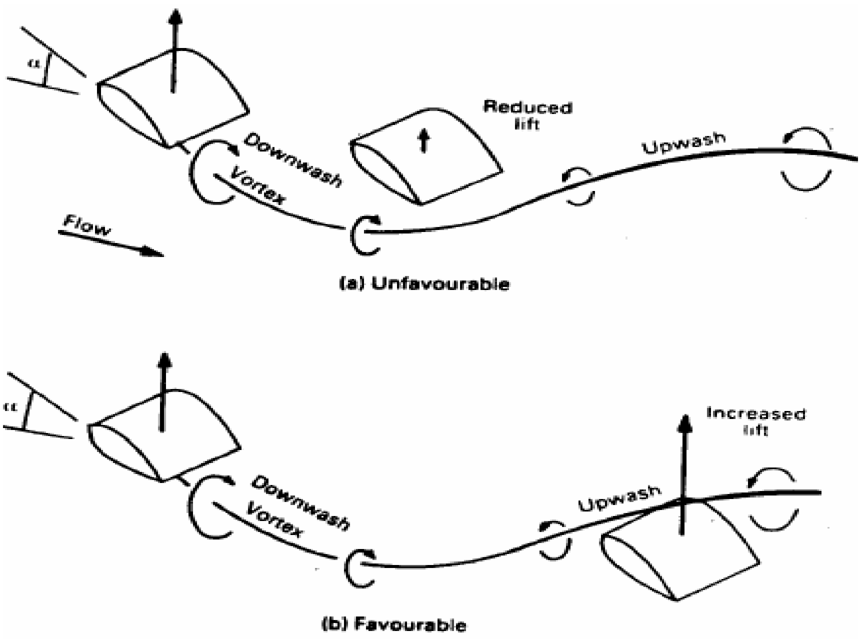


Figure 4.7. Fin-Fin Interference for Oscillating Fins, Lloyd, (1989)

When a fin is the wake of another fin there will be a change in the lift coefficient dependent upon a downwash (and hence a reduction) or upwash (and hence an increase) effect. Lloyd (1976) reported this effect and Cox and Lloyd

(1977) gave useful experimental data for the interaction effect for two identical fins, in combination with the above mentioned boundary layer effect. Figure 4-8 shows the fin-fin interference factor.

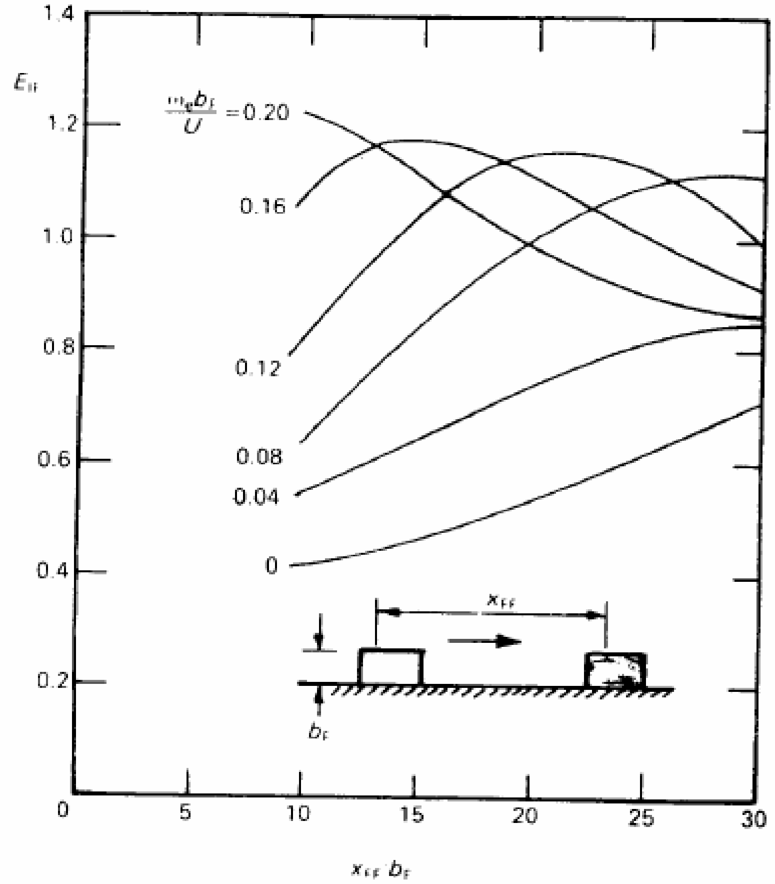


Figure 4.8. Fin-Fin Interference Factors, Lloyd, (1989)

Later, this data was represented in a different non-dimensional form by McCreight and Stahl (1983) and used to obtain the lift coefficients for SWATH ship stabilizers (see Table 4.1).

Table 4.1. Ratio of Lift on Aft Fin to Lift on Forward Fin for Variations Fin Separation and Oscillation Frequency, McCreight et al (1983)

$\omega s / V \rightarrow$ $x / s \downarrow$	0.00	0.04	0.08	0.12	0.16	0.20
10	0.412	0.544	0.643	0.824	1.074	1.221
15	0.462	0.638	0.846	1.046	1.080	1.109
20	0.529	0.732	1.000	1.151	1.110	0.971
25	0.641	0.816	1.099	1.132	1.011	0.897
30	0.706	0.853	1.118	1.006	0.912	0.853

D Distance between leading edge of fins
S Span of fin
V Forward speed
ω Oscillation frequency

The effect of free surface on the lift characteristics of a control surface becomes significant when the ratio of the submergence to chord is less than five and the effect is dependent upon the chord based Froude number. Figure 2-9 shows this effect, (E_{FS}) on a 2-D flat plate, based on theoretical work by Hough and Maran (1965), and is interpolated for fine interval depths of the submergence values for practical use, Atlar, Kenevissi et al. (1997).

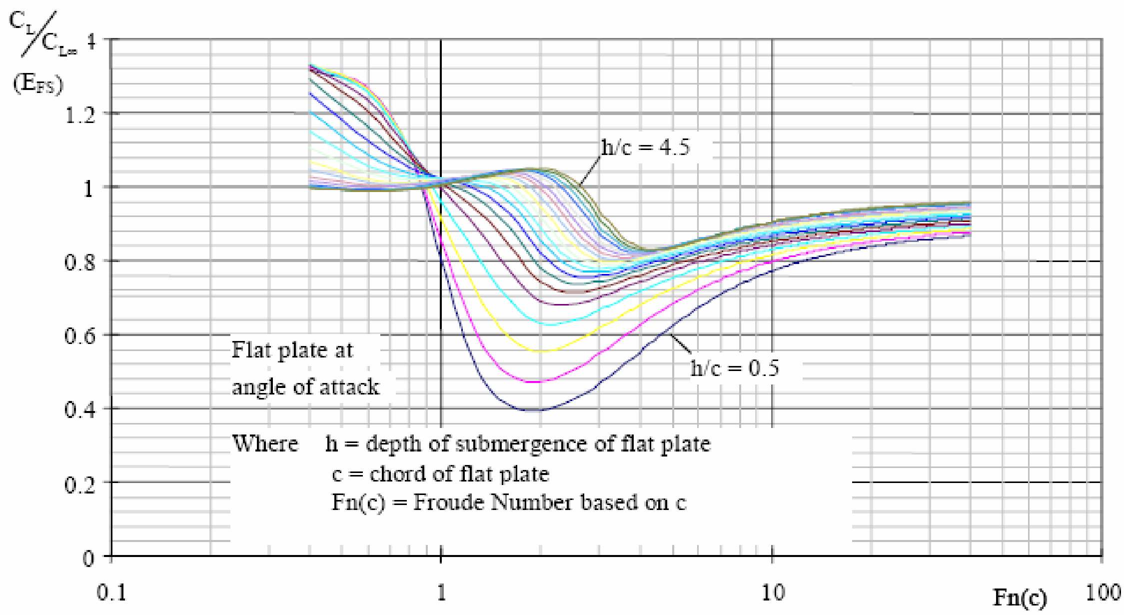


Figure 4.9. Variation of Lift with Submergence, Atlar, Kenevissi et al (1997)

Another source of interference on lift is the effect of the hull on the aft fins, which can be masked by the wake of the lower-hull resulting in a lift reduction. This effect was investigated by Dempsey (1977) to determine the contributions of control surfaces to the forces and moments on the total fin-body combinations. Based upon the data obtained from her experiments, the following semi-empirical expression was provided for a correction to the free stream lift coefficient for stern planes

$$\frac{C_{Z\alpha}}{C_{L\alpha}} = 1-\frac{0.2556}{\left[\frac{2b}{D}\right]^2}\left[\left[\frac{2b}{D}\right]^2 - 0.1612\right]^{1/2} - 0.6366\sin^{-1}\frac{0.4015}{\left[\frac{2b}{D}\right]} \qquad (4.40)$$

Where, $C_{Z\alpha}$ is lift coefficient of the fin attached to the hull; b is the distance between the stern plane tip chord and hull centerline and D is the maximum diameter of the hull (see Figure 4.10).

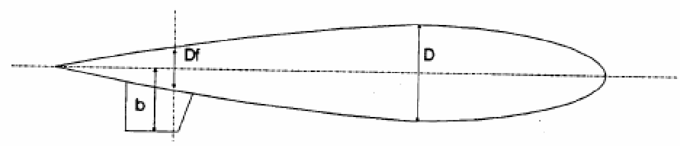


Figure 4.10. Representation of The Geometric Parameters for Hull-Fin Interference, Atlar (1991)

Bearing in mind the approximate nature of the approach, and combining these effects, the lift coefficients for a single pair of fins and a pair of aft fins in presence of forward fins (i.e. canards), can be expressed as,

For a single pair of fins:

$$C_{L\alpha} = K\left[C_{L\alpha}\right]_w E_{BL}E_{FS} \qquad \text{(Fixed)} \qquad (4.41)$$

$$C_{L\alpha} = k\left[C_{L\alpha}\right]_w E_{BL}E_{FS} \qquad \text{(Active)} \qquad (4.42)$$

For aft fins in the wake of forward fins:

$$C_{L\alpha} = K\left[C_{L\alpha}\right]_w E_{BL\text{-}I}E_{FS} \qquad \text{(Fixed)} \qquad (4.43)$$

$$C_{L\delta} = k\left[C_{L\alpha}\right]_w E_{BL\text{-}I}E_{FS} \qquad \text{(Active)} \qquad (4.44)$$

Where, $E_{BL\text{-}I}$ indicates the effect of interference as tabulated in Table 4.1.

The oscillation of the fins will generate cross-flow drag forces in the vertical modes, which can be represented as

$$D_i(t) = 0.5AC_D v_r |v_r| \tag{4.45}$$

Where, A is the projected area of the fin in vertical direction, C_D is the drag coefficient and v_r is the vertical component of the relative velocity. For the estimation of C_D it is suggested that the data for a flat plate given by Bearman and Graham (1979) can be used. A regression equation representing this data is derived, as shown in Figure 4-11, based on the Keulegan Carpenter (KC) number defined as;

$$KC = \frac{v_r T}{2s} \tag{4.46}$$

Where, T is the encounter period.

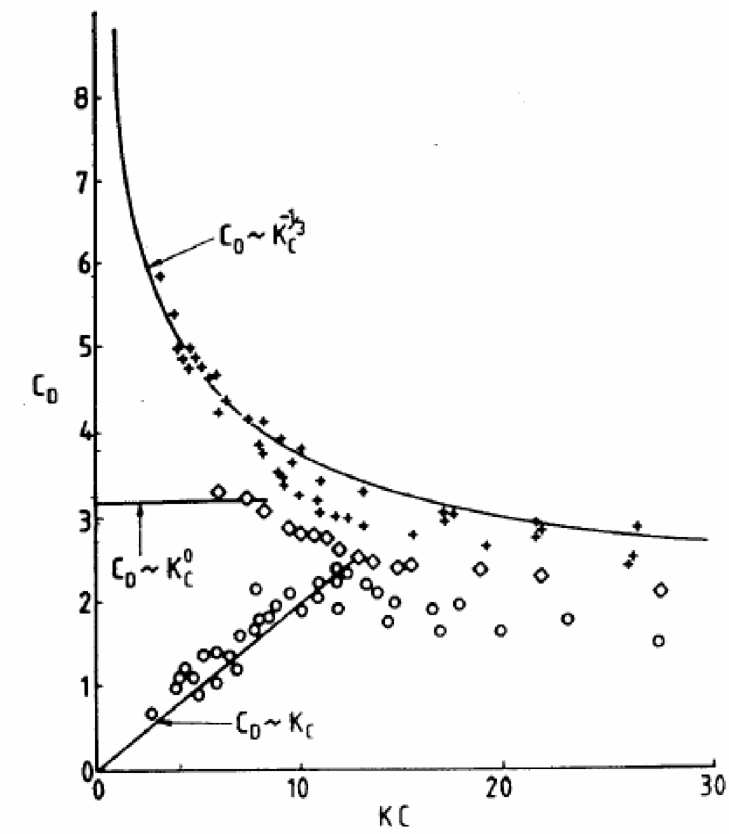


Figure 4.11. The Drag Coefficient of Flat Plate (+), Diamond (◊) and Circular (o) Cylinders at Low KC, Bearman, et al (1979)

Finally, the virtual inertia forces $I_i(t)$ associated with the control surface require the estimation of the mass M_{ij}^f for the fins and can be obtained from,

$$I_i(t) = (M_{ij}^f + A_{ij}^f)\ddot{\eta}_{Aj} \tag{4.47}$$

Where, $\ddot{\eta}_{Aj}$ is the absolute motion of the fin in the j[th] mode of motion. In cases where no specific data is available, the mass and added mass values can be approximated by a neutrally buoyant elliptical cross section, Lee and Curphey (1977).

$$M_{ij}^{f} = \rho\pi\left[\frac{t}{2}\right]\left[\frac{c}{2}\right]s \tag{4.48}$$

and

$$A_{ij}^{f} = \rho\pi\left[\frac{c}{2}\right]s \tag{4.49}$$

Where, c is the chord, s is the span of the fin and t is the maximum thickness of the fin.

4.4 Equations of Motion in Time-Domain Simulation

Under the assumptions that the oscillatory motions are linear and harmonic, the six linear coupled differential equations of motion can be written as

$$\sum_{k=1}^{6}\left[\left(M_{jk}+A_{jk}\right)\ddot{\eta}_{jk} + B_{jk}\,\dot{\eta}_{jk} + C_{jk}\eta_{jk}\right] = F_j e^{i\omega t} \qquad j = 1, 2, \ldots, 6 \tag{4.50}$$

Where, M_{jk} is the generalized mass/inertia matrix of the vessel, A_{jk}, B_{jk} and C_{jk} are the added mass, damping and restoring force matrices, and η_j, F_j are the complex amplitudes of the motion responses, wave exciting forces, respectively. ω is the wave frequency. Subscript j and k are associated with the excitation and motion modes respectively and take similar values, where J=1 denotes surge motion, J=2 for sway, for J=3 heave, J=4 for roll, J=5 for pitch and J=6 for yaw).

Based on the assumption that a twin-hull vessel has a longitudinal centre-plane of symmetry, the vertical plane modes of motions are decoupled from the lateral plane modes. By neglecting the effect of surge motion due to the high slenderness ratio of demi-hulls, the time-domain coupled motion equations for the heave and pitch modes can be written as,

$$\left(M_{33}+A_{33}\right)\ddot{\eta}_3+B_{33}\dot{\eta}_3+C_{33}\eta_3+A_{35}\ddot{\eta}_5+B_{35}\dot{\eta}_5+C_{35}\eta_5=F_3e^{i\omega t} \quad (4.51)$$

$$A_{53}\ddot{\eta}_3+B_{53}\dot{\eta}_3+C_{53}\eta_3+\left(I_{55}+A_{55}\right)\ddot{\eta}_5+B_{55}\dot{\eta}_5+C_{55}\eta_5=F_5e^{i\omega t} \quad (4.52)$$

4.4.1 Solution of The Motion Equations

To obtain the motion of the vessel in time step, the numerical investigation technique is applied to solve the equations of motion. Even the fourth order Runge-Kutta-Merson integration procedure is widely used in the Time-Domain Simulation, in this study, a fixed-step ode2 (Heun) is utilised. This procedure has been widely used in the solution of the differential equations. The advantage of this method is able to provide a very fast, realistic, and reliable control computation results. The detail of the Time-Domain Simulation incorporated with the PID controller will be studied in the Chapter V.

To applied this method, the second order term occurs in the equation of motion is transformed to the first order differential equation. The procedure to apply this method is showed as follows:

Equation (4.53) is the general equation of motion.

$$a_j\ddot{\eta}+b_j\dot{\eta}+c_j\eta+\sum_{k=1}^{6}a_{jk}\ddot{\eta}_k-\sum_{k=1}^{6}b_{jk}\dot{\eta}_k+\sum_{k=1}^{6}c_{jk}\eta_k=F_j$$

(4.53)

Then, the equation of motion is rewritten in the form that needs to be integrated.

$$\ddot{\eta}=\left[F_j-b_j\dot{\eta}-c_j\eta-\sum_{k=1}^{6}a_{jk}\ddot{\eta}_k-\sum_{k=1}^{6}b_{jk}\dot{\eta}_k-\sum_{k=1}^{6}c_{jk}\eta_k\right]/a_j$$

(4.54)

To integrate the above set of equation with the numerical integration technique, the second order term is transformed to the first order term.

Let,

$$y_1 = \eta \tag{4.55}$$

and,

$$y_2 = \dot{\eta} \tag{4.56}$$

Therefore, the first order differential equation becomes:

$$\dot{y}_1 = y_2 \tag{4.57}$$

and,

$$\dot{y}_2 = \left[F_j - b_j \dot{\eta} - c_j \eta - \sum_{k=1}^{6} a_{jk} \ddot{\eta}_k - \sum_{k=1}^{6} b_{jk} \dot{\eta}_k - \sum_{k=1}^{6} c_{jk} \eta_k \right] / a_j \tag{4.58}$$

4.5 Concluding Remarks

A time domain simulation program, which takes into account of the non-linear effect and the coupling between the motions is utilised to solve the problem of ship motion in waves. Since the study concentrates on seakeeping of Semi-SWATH vessel in head seas, accurate computations both of hydrodynamic and hydrostatic forces become important due to the vessel has low stiffness and tendency for large pitch motions. This phenomenon has a significant effect on the longitudinal ship stability especially at high forward speed, thus, pitch moment will be calculated considerably.

CHAPTER 5

IMPROVED VESSEL RIDE PERFORMANCE USING TIME-DOMAIN SIMULATION

5.1 General

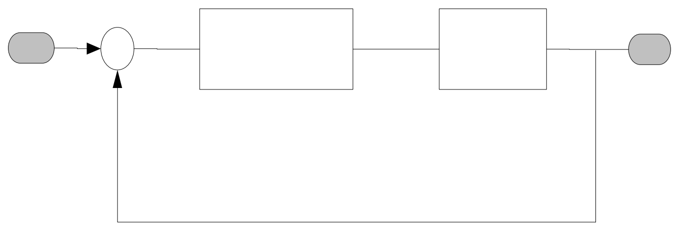
This chapter presents the formulation and numerical applications for an optimal strategy to improve the ride performance of Semi-SWATH vessel using Time-Domain Simulation. To improve the robustness of the control system, the application of conventional PID control is applied.

The PID Control is a classical control technique, its acronym of PID stands for Proportional-Integral-Derivative. Inherent to its control law, the PID Control can be designed not only to control a single-input single-output (SISO) controller but also to handle multiple-input multiple-output (MIMO) systems.

The primary advantage using The PID controller is providing a robust performance for a wide range of operating conditions. Furthermore, it is easy to implement using analogue or digital hardware and familiar to engineers. Previously, many researchers e.g. Caldeira, et al (1984), ware, et al (1980a), (1980b), 1981, and 1987, and Chinn, et al (1994) applied conventional optimal controller design to improve the vertical motion response of marine vehicles.

5.2 Simple Block of Control System using PID Controller

Figure 5.1 has shown a simple block of controller system using PID controller. This PID controller is set up to obtain a desired response, $Y_{ref}(t)$ with minimize the error (e) deviations.



Figure; 5.1, Simple Block of Control System

Plant : A system to be controlled
Controller : Provides the excitation for the plant; Designed to control the overall system behavior

5.3 The Three-Term PID Controller

The transfer function of the PID controller looks like the following:

$$K_P + \frac{K_I}{s} + K_D s = \frac{K_D s^2 + K_P s + K_I}{s} \qquad (5.1)$$

- K_P = Proportional gain
- K_I = Integral gain
- K_D = Derivative gain

A general closed loop of PID controller shown (in figure 5.1) implements the control variable u(t) as a function of the error e(t). The variable (e) represents the tracking error, the difference between the desired input value $Y_{ref}(t)$ and the actual output y(t). This error signal (e) will be sent to the PID controller, and the controller computes both the derivative and the integral of this error signal. Its means the PID controller bases its action on the sum of three values derived from the error: a proportional action, a derivative action, and an integral action. The weights of the three different actions are the parameters of a PID controller. Furthermore, the signal (u) just past the controller is now equal to the proportional gain (Kp) times the magnitude of the error plus the integral gain (Ki) times the integral of the error plus the derivative gain (Kd) times the derivative of the error.

Furthermore, this signal (u) will be sent to the plant, and the new output $Y_{ref}(t)$ will be obtained.

$$u = K_P + K_I \int edt + K_D \frac{de}{dt} \qquad (5.2)$$

This new output y(t) will be sent back to the sensor again to find the new error signal (e). The controller takes this new error signal and computes its derivative and its integral again. This process goes on and on.

5.4 The Proportional-Integral-Derivative (PID) Algorithm

As the name suggests, the PID algorithm consists of three basic modes, the Proportional mode, the Integral and the Derivative modes. When utilizing this algorithm it is necessary to decide which modes are to be used (P, I or D?) and then specify the parameters (or settings) for each mode used. Generally, three basic algorithms are used P, PI, or PID

5.4.1 A Proportional Algorithm

The mathematical representation is;

$$\frac{mv(s)}{e(s)} = k_c \text{ (Laplace domain) or } mv(t) = mv_{ss} + k_c e(t) \text{ (Time domain)} \qquad (5.3)$$

The proportional mode adjusts the output signal in direct proportion to the controller input (which is the error signal, e). The adjustable parameter to be specified is the controller gain, kc. This is not to be confused with the process gain, kp. The larger kc the more the controller output will change for a given error. Some controllers have proportional gain adjustable as percent proportional band PB, where Kc = 100/PB.

The time domain expression also indicates that the controller requires calibration around the steady-state operating point. This is indicated by the constant term mv_{ss}. This represents the 'steady-state' signal for the mv and is used to ensure that at zero error the cv is at set-point. In the Laplace domain this term disappears, because of the ‘deviation variable’ representation.

A proportional controller reduces error but does not eliminate it (unless the process has naturally integrating properties), i.e. an offset between the actual and desired value will normally exist.

5.4.2 A Proportional Integral Algorithm

The mathematical representation is;

$$\frac{mv(s)}{e(s)} = k_c\left[1+\frac{1}{T_i s}\right] \text{ Or } mv(t) = mv_{ss} + k_c\left[e(t)+\frac{1}{T_i}\int e(t)dt\right] \qquad (5.4)$$

The additional integral mode (often referred to as reset) corrects for any offset (error) that may occur between the desired value (set-point) and the process output automatically over-time. The adjustable parameter to be specified is the integral time (Ti) of the controller.

5.4.3 A Proportional Integral Derivative Algorithm

The mathematical representation is;

$$\frac{mv(s)}{e(s)} = k_c\left[1+\frac{1}{T_i s}+T_D s\right] \text{Or}$$

$$mv(t) = mv_{ss} + k_c\left[e(t)+\frac{1}{T_i}\int e(t)dt + T_D\frac{de(t}{dt})\right] \qquad (5.5)$$

Derivative action (also called rate or pre-act) *anticipates* where the process is heading by looking at the time rate of change of the controlled variable (its derivative). T_D is the ‘rate time’ and this characterizes the derivative action (with units of minutes). In theory derivative action should always improve dynamic response and it does in many loops. In others, however, the problem of noisy signals makes the use of derivative action undesirable (differentiating noisy signals can translate into excessive mv movement). Furthermore, Derivative action depends on the slope of the error, unlike P and I. If the error is constant derivative action has no effect.

5.5 Controller Tuning

A controller is only as effective as its tuning: the adjustment of the PID settings relative to the process parameters to optimize load and set-point response. Where, controller tuning involves the selection of the best values of kc, Ti and T_D (if a PID algorithm is being used). This is often a subjective procedure and is certainly process dependent. A number of methods have been proposed in the literature over the last 50 years. Tuning is required when the controller is first commissioned on a loop, and may have to be repeated if process parameters change appreciably with time, load, set point, etc. The suggestion being a PID controller can be properly tuned to improve the operational performance of process plant to match some preconceived 'ideal' response profile for the closed loop system.

5.5.1 Parameter Tuning Rules for PID Controller

As written in the equation (5.1), the transfer function of a PID controller can be modified in the following form:

$$G_{PID} = K_p + \left(1 + \frac{1}{T_i s}\right) + T_d s \qquad (5.6\ )$$

Where $T_i = \dfrac{K_p}{K_i}$ and $T_d = \dfrac{K_d}{K_p}$, T_i and T_d are the integral time constant and the derivative time constant, respectively.

The tuning objective is to determine the suitable value of three parameters (K_p, K_i, *and* K_d) to satisfy certain control specifications. Aström and Hagglund (1984), in order to obtain the initial parameters of PID controller, the Astrom-Hagglund method will be used to determine the values of critical period of waveform oscillation (T_c) and critical gain (K_c). These two values could be obtained by running the closed loop control of DC servomotor system utilizing relay feedback as a controller. The oscillation period of the output waveform is considered as the critical period attained from a proportional feedback. Based on this critical period (see fig. 5.2), the critical gain can be derived as follow:

$$K_c = \frac{4d}{\pi a} \qquad (5.7)$$

Where d is the amplitude of the relay output, and a is the amplitude of the waveform oscillation.

Based on these two values, the PID parameters (K_p, T_i , and T_d) can be specified using Ziegler-Nichols formula. Ziegler and Nichols, (1942) developed the first effective tuning methods, and these are still used today. The method is straightforward. First, set the controller to P mode only. Next, set the gain of the controller (Kc) to a small value. Make a small set-point (or load) change and observe the response of the controlled variable. If Kc is low the response should be sluggish. Increase Kc by a factor of two and make another small change in the set-point or the load. Keep increasing Kc (by a factor of two) until the response becomes oscillatory. Finally, adjust Kc until a response is obtained that produces continuous oscillations. The control law settings are then obtained from the following Table 5.1.

Table 5.1 Ziegler-Nichols Parameter Tuning

	K_p	T_i	T_d
P	0.5 K_c		
PI	0.45 K_c	0.85 T_c	
PID	0.6 K_c	0.5 T_c	0.125 T_c

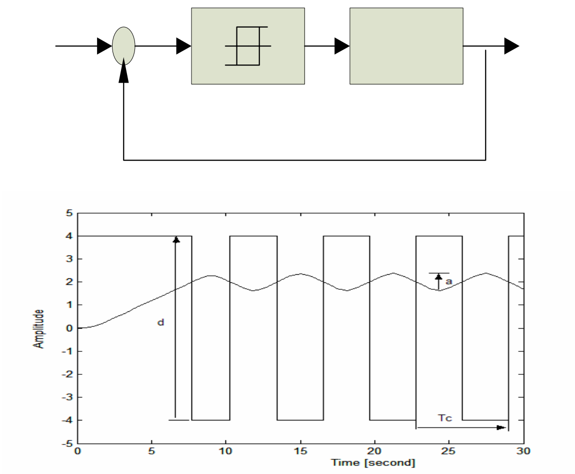


Figure 5.2 The relay feedback controller

5.6 Actuator Modeling

The electromechanical dual acting on the fin stabilizer shaft system was utilized two DC servomotors ac actuators. Output shaft of the DC servomotor is directly connected to a series of gear reducers to initiate the fin shaft's movement. Based on this arrangement, the DC servomotor enables to control the shaft's movements directly.

5.6.1 Modeling of DC Servomotor

The dynamic model of the DC servomotor is represented as follow:

$$V_a = L_a \frac{di_a}{dt} + R_a . i_a + K_a \omega_a \tag{5.8}$$

$$T_m = K_m . \, i_a \tag{5.9}$$

$T_m - T_L = J_m\ (d\omega/dt)+B_m(d\theta/dt)$ (5.10)

By using LaPlace Operator $s = d/dt,$ then

$$V_a = sL_a i_a + R_a i_a + K_a \omega \quad (5.11)$$

$$V_a - K_a.\omega = s.L_a.i_a + R_a.i_a \quad (5.12)$$

$$T_m - T_L = J_m.s.\omega + B_m.s^2.\omega \quad (5.13)$$

$$\theta = \omega/s \quad (5.14)$$

Where :

V_a : Motor Voltage [V]

L_a : Motor Inductance [H]

i_a : Motor Current [A]

R_a : Motor Resistance [Ω]

K_a : Back emf constant [mV/(rad/sec)]

ω : Motor shaft angular velocity [rad/sec]

θ : Angular displacement [rad]

T_m : Motor Torque [Nm]

K_m: Torque Constant [Nm/A]

T_L : Load Torque [Nm]

Jm: Motor Inertia [Nm.sec^2]

For the detail of servo motor parameter used in this simulation, see Table 5.2.

The simulation control studies of the proposed PID controller are carried out in order to investigate its effectiveness in this position control application. In these studies, the DC servomotor as the actuator has the following important parameters as shown in Table 5.2.

Table 5.2 DC servomotor parameters

Parameters	**Values**
Motor Voltage	24 V
Armature Resistance	0.03 Ω
Armature Inductance	0.1 mH
Back emf Constant	7 mV/rpm
Torque Constant	0.0674 N/A
Nominal Torque	1.6 Nm
Nominal Speed	3000 rpm
Rotor Inertia	0.1555e-4 N.sec^2

Based on these parameters, the simulation of the system was investigated. The simulation of the controller was performed using MATLAB-SIMULINK packages. Control performance is determined based on percent overshoot (POS), and settling time t_s, and steady state error e_{ss}. Two types of input excitation: steps, and sinusoidal waveform, are used to examine the performance of the conventional PID controllers. The initial proportional gain will be set to the same value to ensure reasonable comparisons between these two controllers.

In order to obtain initial parameters of PID, the Aström-Hagglund method based on a relay feedback controller is carried out to attain the critical period of waveform oscillation (T_c) and critical gain (K_c). The relay feedback controller is used in a closed loop control application. The amplitude of the relay controller is set to 48 since the input voltage in the range of [±24 volts] is needed to drive the servo system. From simulation results, these following parameters are found: $T_c = 0.39$ *s*, $a = 2.96$, and $d = 48$ (see figure 5.3, 5.4, and 5.4 shown he results of relay feedback controller).

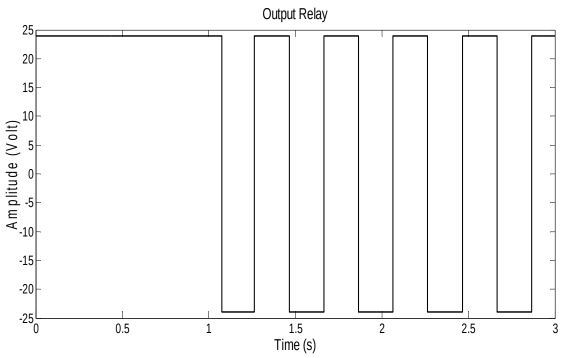


Figure 5.3 Relay output

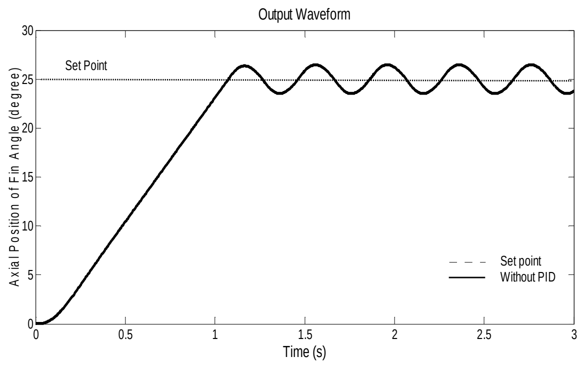


Figure 5.4 Waveform of oscillation

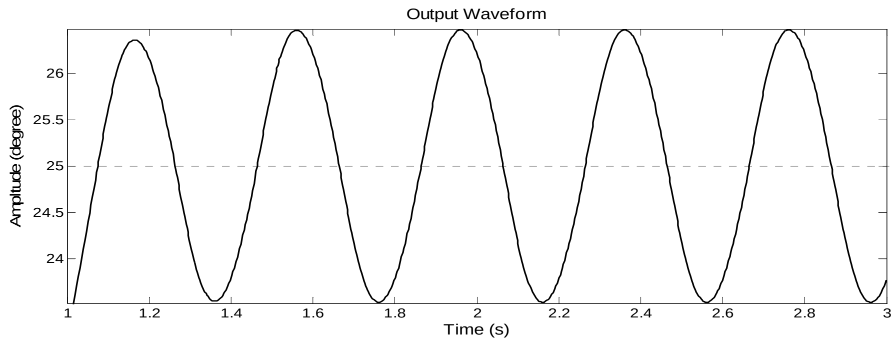


Figure 5.5 Zoomed waveform of oscillation

By using equation (5.7), the critical gain (K_c) is 20.657. Then, the Ziegler-Nichols formula (see table 5.1) is applied to find the values of K_p, T_i, and T_d. Finally, by using these values and equations (5-5) and (5-6), the three parameters of PID can be specified as follows: K_p= 12.395*,* $K_i = 63.562$*,* and $K_d = 0.604$. From these data, it can be seen that the value of the integral gain *(K_i)* is much bigger compare to other gains. By closely looking at the small amplitude of waveform oscillations, it can be seen that the servo system exhibits a small steady state error of about 5.92 % that was shown the output response of the conventional PID when the step input of 25 degree is applied, see figure 5.6. This condition can be understood, since the servo system utilizes gear reducers with a total ratio of about (750:1) to supply the rotation of motor shaft force, hence slowing down the fin shaft movement significantly. Based on this fact, it is reasonable to say that the integral gain was not used for controlling this kind of servo system, since the system behavior has already had a small tolerable steady state error.

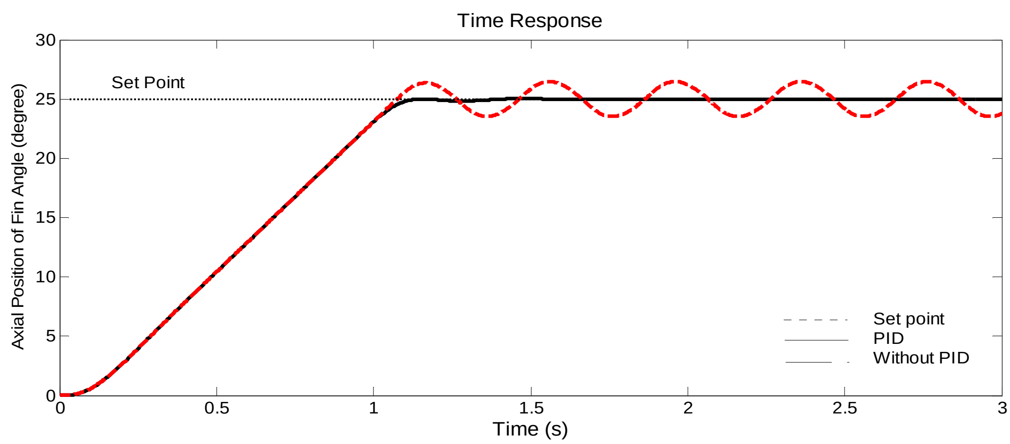


Figure 5.6 Output responses when step input is applied

In addition, Figure 5.6 shows the output responses of the sinusoidal wave excitation. From these figures it can be seen that the PID has smaller position error clearly compared to the without PID one. Thus, the PID gives good tracking trajectory with good response and minimum (tolerable) error as shown in above figures.

Overall, the PID controller performs better than without controller especially in terms of settling time and tracking. In terms of percent overshoot and steady state error, both controllers have good performance.

5.7 Application of PID Controller to Multi-Hull Motion Control

As mentioned in Chapter (4), based on the assumption that a twin-hull vessel has a longitudinal centre-plane of symmetry and neglecting the effect of surge motion due to the high slenderness ratio of the demi-hulls, the coupled motion equations for the heave and pitch modes can be written;

$$\left(M_{33}+A_{33}\right)\ddot{\eta}_3+B_{33}\dot{\eta}_3+C_{33}\eta_3+A_{35}\ddot{\eta}_5+B_{35}\dot{\eta}_5+C_{35}\eta_5=F_3\cos\left(\omega t+\varphi_3\right) \quad (5.15)$$

$$A_{53}\ddot{\eta}_3+B_{53}\dot{\eta}_3+C_{53}\eta_3+\left(I_{55}+A_{55}\right)\ddot{\eta}_5+B_{55}\dot{\eta}_5+C_{55}\eta_5=F_5\cos\left(\omega t+\varphi_5\right) \quad (5.16)$$

$$L_{fin}=\frac{\rho}{2}V^2A_PC_{L\delta}\left(\delta\right) \quad (5.17)$$

Where ρ is fluid density, Vs vessel velocity, and A_P is projected area of the fin. $C_{L\delta}$ is the lift coefficient due to the fin angular deflection of δ with respect to the hull.

The fin moment is the lift force of each fin times the distance of the fin to the center of gravity of the vessel, l.

$$L_{fore}=\frac{\rho}{2}V^2A_fC_{L\delta}\delta_F \quad (5.18)$$

$$M_{fore}=\frac{\rho}{2}V^2A_fC_{L\delta}\delta_f l_f \quad (5.19)$$

$$L_{aft}=\frac{\rho}{2}V^2A_fC_{L\delta}\delta_A \quad (5.20)$$

$$M_{aft} = \frac{\rho}{2} V^2 A_f C_{L\delta} \delta_A l_a \tag{5.21}$$

and l is the distance from the CG to the quarter chord of each fins. The appropriate superscript or subscript of f and a indicated an association with the fore and the aft fin respectively.

5.8 Closed-Loop of Anti-Pitching Fin Control System

A block diagram of the PID controller for regulation of pitch motions for a twin-hull vessel with aft controllable fins is presented in figure 5.6 and figure 5.7, where δ_{aft} is the deflection of aft control surfaces.

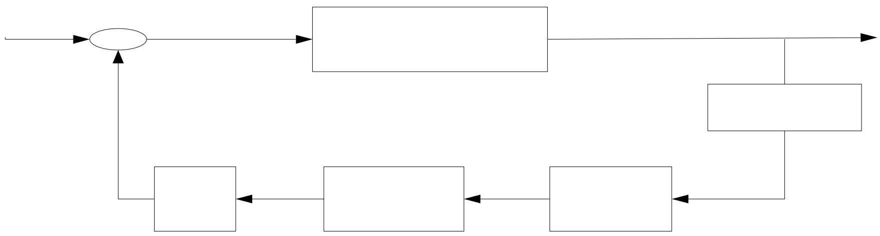


Figure 5.7, Closed-loop anti-pitching control system

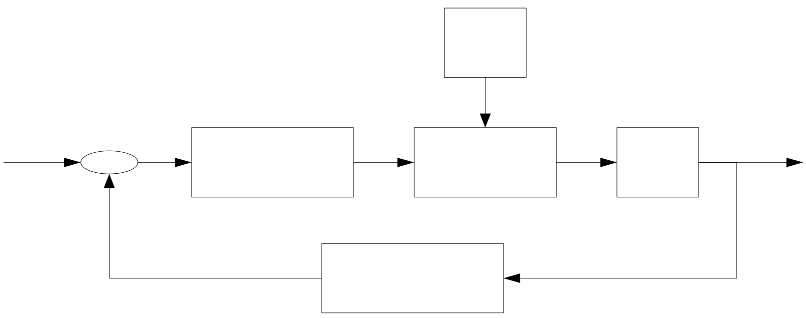


Figure 5.8, Servo motor of the anti-pitching fin

Figure 5.7 shows the closed-loop structure of the anti-pitching control system. The system uses the measured pitch and the pitch rate as feedback through the block "controller" based on future information. The control moment, M_f , is provided by a servo motor and a pair of stabilization fins via the control variable δ_c (shown in figure 5.8). A simple PID controlled servo-mechanism is used for the angular position servo of the anti-pitching fin system.

5.9 Time-Domain Simulation Program Structure

A simulation program was developed to simulation the Semi-SWATH vessel in two degrees of freedom i.e. heave and pitch motions. As consequence, 2 subroutines for the 2 degrees of freedom have to be developed. The programming structure f single degree of freedom is shown in figure 5.9. The acceleration value was obtained from total force (or moment) by using method shown in equation (5.15) and (5.16). The velocity and displacement (position) values are obtained with applying one and two integration respectively. Two loop back flow in the structure were developed to feed back the acceleration dependent value and velocity dependent value to total force value. These feed back values will contribute to the total force at the next computation time step.

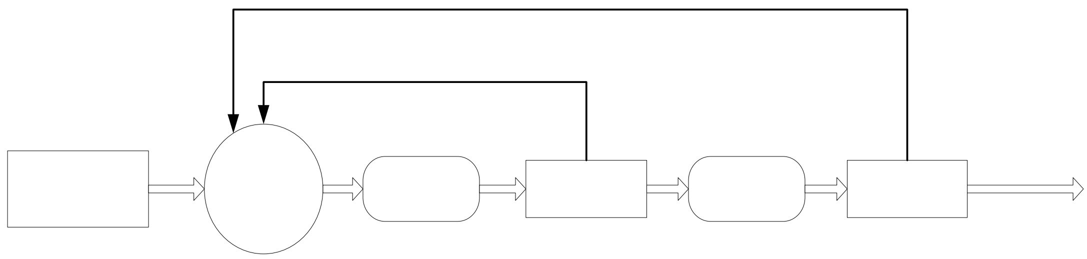


Figure 5.9 Programming structure for a single degree of freedom

The motion for axis will cross coupling with others axis during the real motions. To enable this in computational, the cross coupling flow to be included in the system for more axis. The motion values after integrations (such as; acceleration, velocity, and displacement) have to be looped to other subroutine for the next computation step. As example, the cross coupling (heave and pitch equations) for two subroutine is shown in figure 5.10.

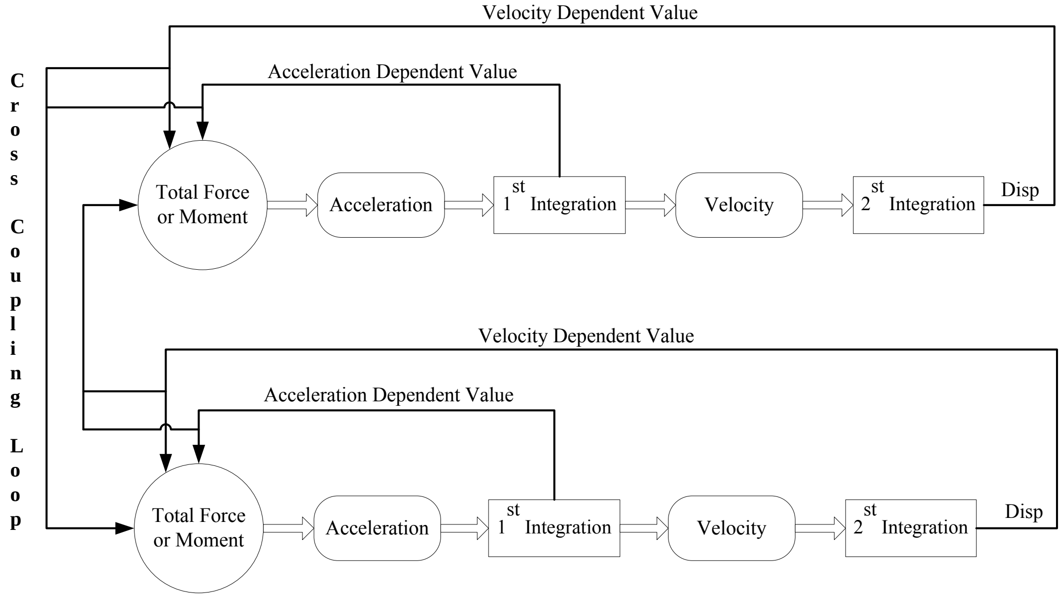


Figure 5.10 Programming structure for two degrees of freedoms

5.10 Computer Simulation

A simulation program was developed under Matlab-SIMULINK environment by using all equations involved at equation (5-25) and (5-16) respectively. The advantages of using Matlab-SIMULINK are;

- Graphical programming, which easily understand and to develop (see figure 5.11)
- Contain built in library for Time-Domain Integration
- The Matlab-SIMULINK support sub-system graphical programming, thus makes the programming for modular mathematics easier to develop.

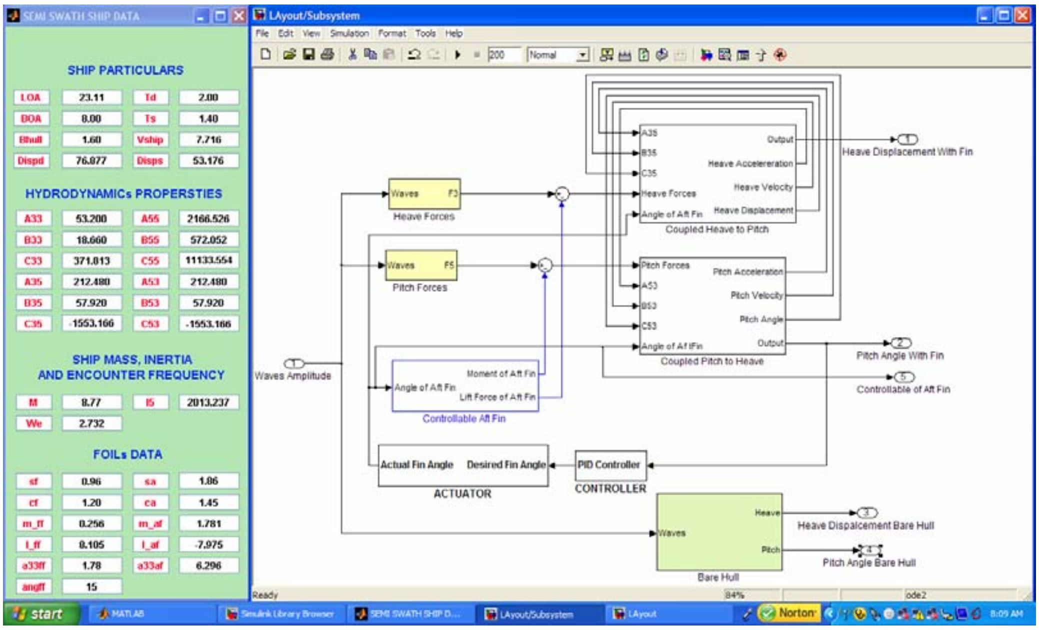


Figure 5.11 Layout of graphic programming using Matlab-SIMULINK

5.11 Simulation Condition

The simulation of the coupled heave and pitch motions on the Semi-SWATH vessel have been carried out under various conditions, see on the Table 5.3 and Table 5.4.

Table 5.3 Simulation Condition

Case No	Vessel Condition	Fin Angle (Degree) Fixed Fore	Fixed Aft	Active Aft	T (m)	Vs (knots)
1	a. Bare hull (**)	-	-	-	2.0	15
	b. Bare with fixed fore and aft fin (**) (Five cases; A, B, C, D, and E)	15	5,10,15,20 and 25	-		
	c. Bare hull with fixed fore fin and controllable aft fin using PID controller (one case; E)	15	25	0 - 25		
2	a. Bare hull (**)	-	-	-	1.4	20
	b. Bare with fixed fore and aft fin (**) (Five cases; A, B, C, D, and E)	15	5,10,15,20 and 25	-		
	c. Bare hull with fixed fore fin and controllable aft fin using PID controller (one case; E) (*)	15	25	0 - 25		

(*) Simulation condition
(**) Simulation and experiment condition

Table 5.4 Parametric of tuning PID controller

<table>
<tr><th rowspan="2">Case No</th><th colspan="2">PID Controller Gain</th><th colspan="2">Fin Angle (Degree)</th><th rowspan="2">T (m)</th><th rowspan="2">Vs (knots)</th></tr>
<tr><th>Inner Controller</th><th>Outer Controller</th><th>Fixed Fore Fin</th><th>Saturation of Controllable Aft Fin</th></tr>
<tr><td>1</td><td>$K_p = 12.395$
$K_d = 0.604$</td><td>$K_p = 275$
$K_i = 35$
$K_d = 45$</td><td>15</td><td>0 - 25</td><td>2.0</td><td>15</td></tr>
<tr><td>2</td><td>Kp = 12.395
Kd = 0.604</td><td>Kp = 220
Ki = 30
Kd = 10</td><td>15</td><td>0 - 25</td><td>1.4</td><td>20</td></tr>
</table>

5.12 Simulation Results

The simulation programming for this study was developed with a graphical output for the ship state data i.e. position of the vessel. For all simulations condition above, the heave displacement and pitch angle of the Semi-SWATH vessel have been recorded for comparison. The oscillations of Semi-SWATH vessel before and after attached anti-pitching fins were shown graphically in figure 5.12 until figure 5.41. For each case result was summarized on the Table 5.5 until Table 5.8.

5.12.1 First Simulation Results

Case 1

Parameter for T = 2.0 m, Vs = 15 knots, Inner Controller: (K_p=12.395, K_d=0.604)

Outer Controller: (K_p=275, K_i=35 and K_d=45)

Table 5.5 Semi-SWATH vessel results achieved by the application of tuning parameter of PID Controller at T=2.0 m and Vs =15 Knots

Description	**Case 1E.1**		**Case 1E.2**		**Case 1E.3**		**Case 1E.4**		**Case 1E.5**	
	WH	**λ/L_s**	**WH**	**λ/L_s**	**WH**	**λ/L_s**	**WH**	**λ/L_s**	**WH**	**λ/L_s**
	0.857	**1.8**	**0.952**	**2.0**	**1.05**	**2.2**	**1.14**	**2.4**	**1.19**	**2.5**
Heave improvement (%)	*38.83 **48.36		*28.58 **38.05		*28.98 **42.57		*30.95 **38.49		*22.50 **32.44	
Pitch improvement (%)	*38.89 **43.09		*28.18 **40.80		*34.27 **48.980		*30.25 **38.83		*24.21 **36.13	

***** Heave or pitch motions improvement due to fixed fore fin and aft fin as compared to bare hull vessel***

*****Heave or pitch motions improvement due to fixed fore fin and controllable aft fin as compared to bare hull vessel***

Case 1E.1

Saturation of aft fin angle = 0^0 to 25^0 fore fin = 15^0

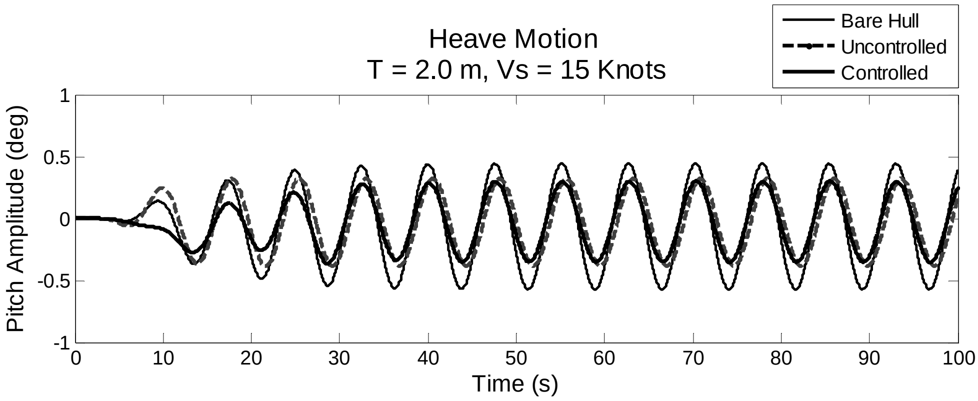


Figure 5.12 Heave motion of Semi-SWATH vessel using controllable aft fin at wave height = 0.857 m, $\lambda/L_S = 1.8$

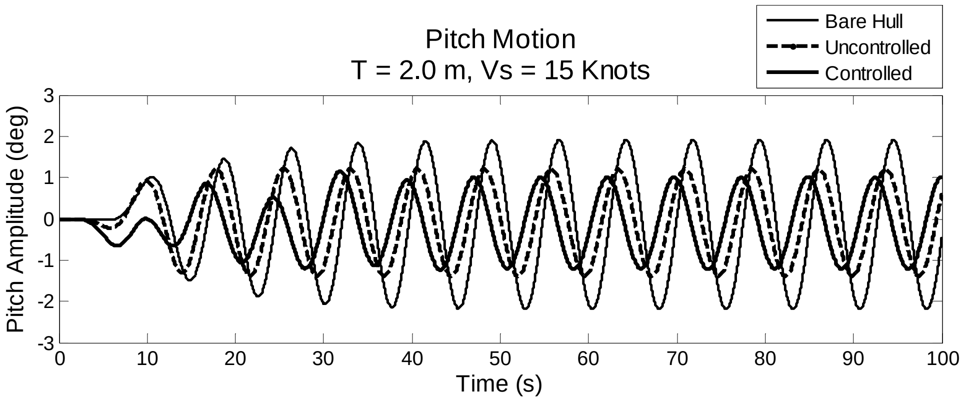


Figure 5.13 Pitch motion of Semi-SWATH vessel using controllable aft fin at wave height = 0.857 m, $\lambda/L_S = 1.8$

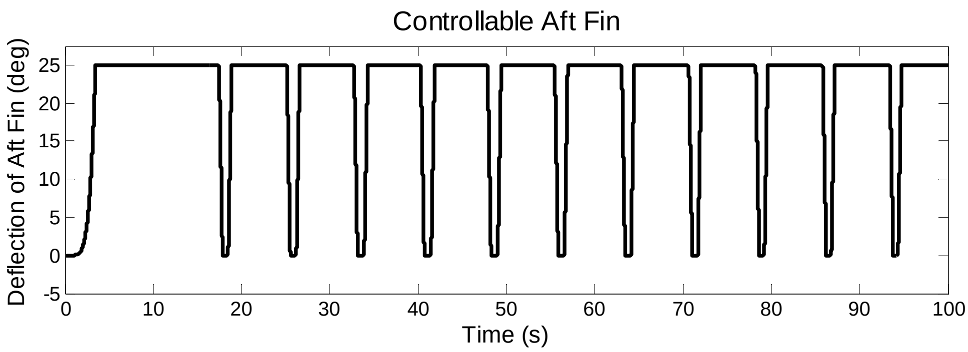


Figure 5.14 Aft fins deflection due to the application of the designed controller at vessel speed=15 Knots

Case 1E.2

Saturation of aft fin angle = 0^0 to 25^0 fore fin = 15^0

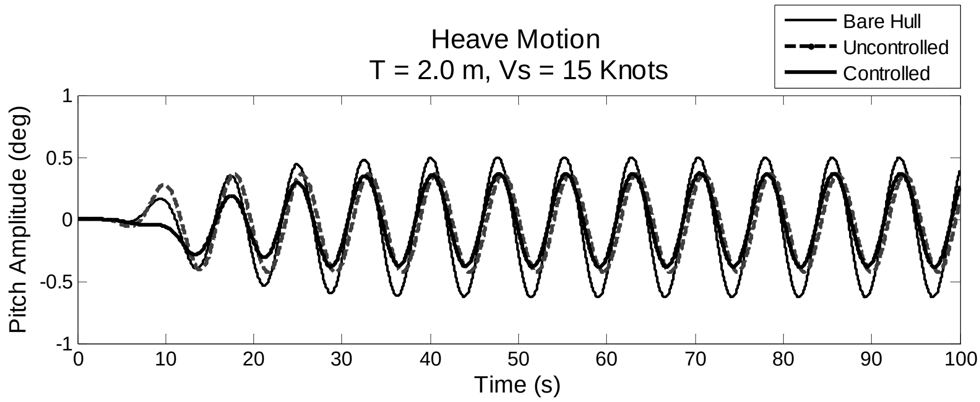


Figure 5.15 Heave motion of Semi-SWATH vessel using controllable aft fin at wave height = 0.952 m, $\lambda/L_s = 2.0$

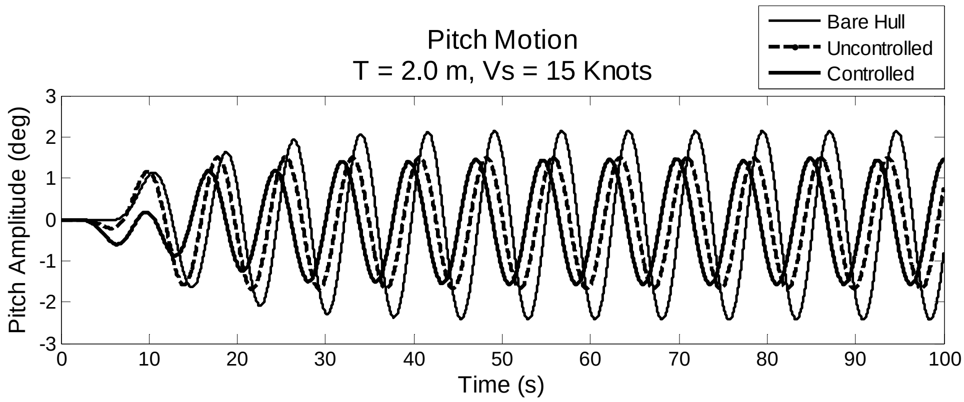


Figure 5.16 Pitch motion of Semi-SWATH vessel using controllable aft fin at wave height = 0.952 m, $\lambda/L_s = 2.0$

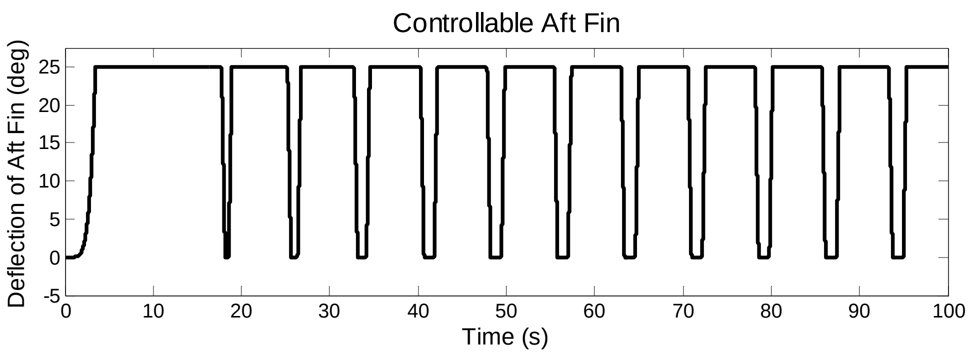


Figure 5.17 Aft fins deflection due to the application of the designed controller at vessel speed=15 Knots

Case 1E.3

Saturation of aft fin angle = 0^0 to 25^0 fore fin = 15^0

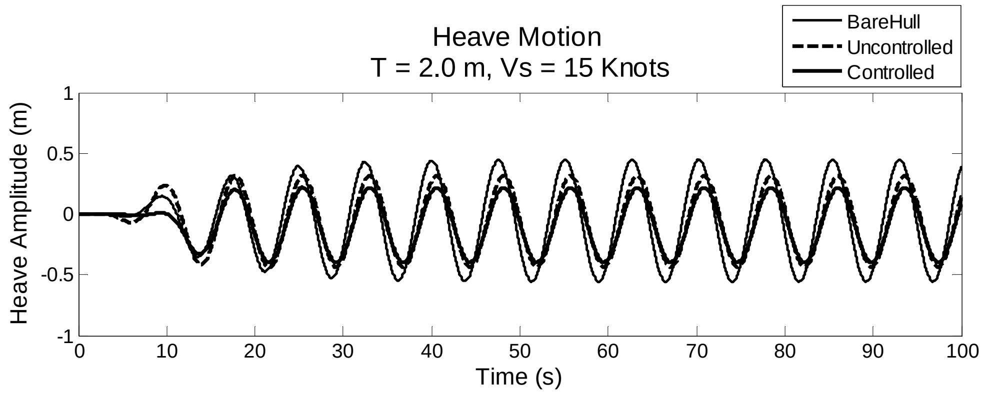


Figure 5.18 Heave motion of Semi-SWATH vessel using controllable aft fin at wave height = 1.05 m, $\lambda/L_s = 2.2$

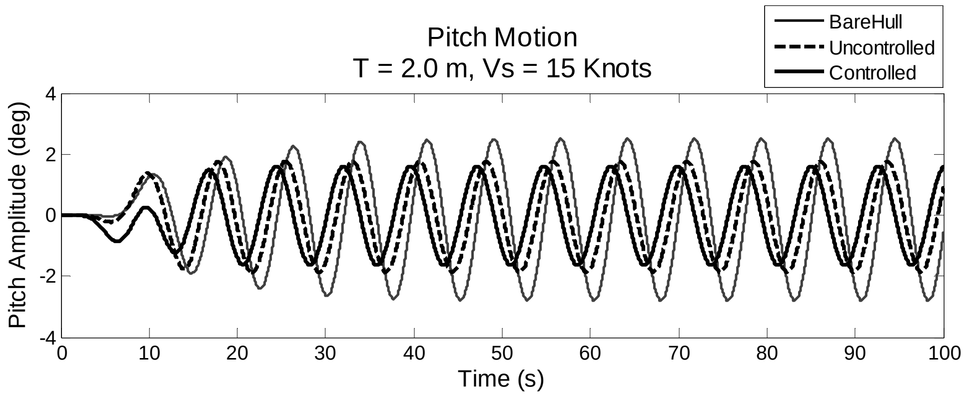


Figure 5.19 Pitch motion of Semi-SWATH vessel using controllable aft fin at wave height = 1.05 m, $\lambda/L_s = 2.2$

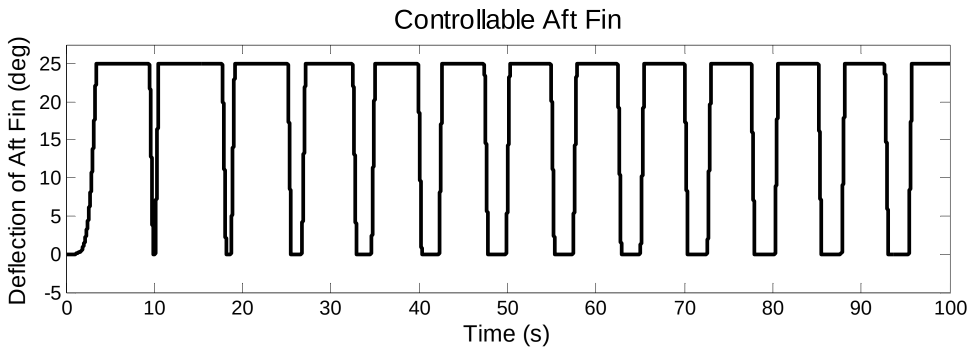


Figure 5.20 Aft fins deflection due to the application of the designed controller at vessel speed=15 Knots

Case 1E.4

Saturation of aft fin angle = 0^0 to 25^0 fore fin = 15^0

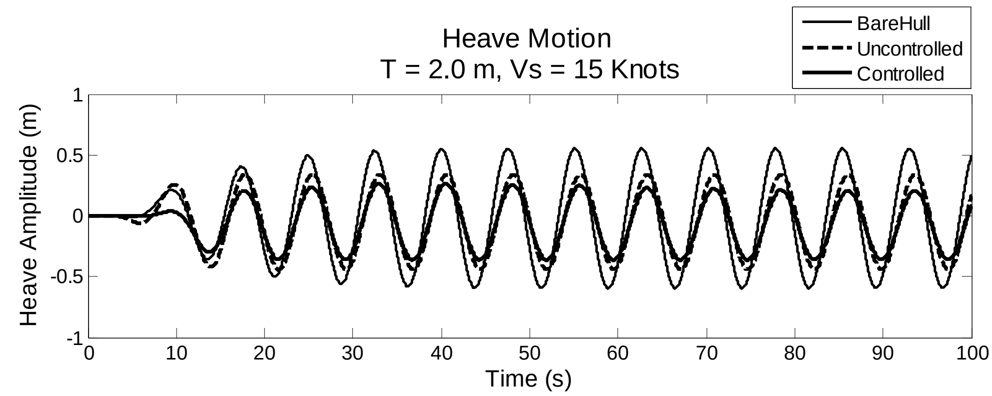


Figure 5.21 Heave motion of Semi-SWATH vessel using controllable aft fin at wave height = 1.14 m, $\lambda/L_s = 2.4$

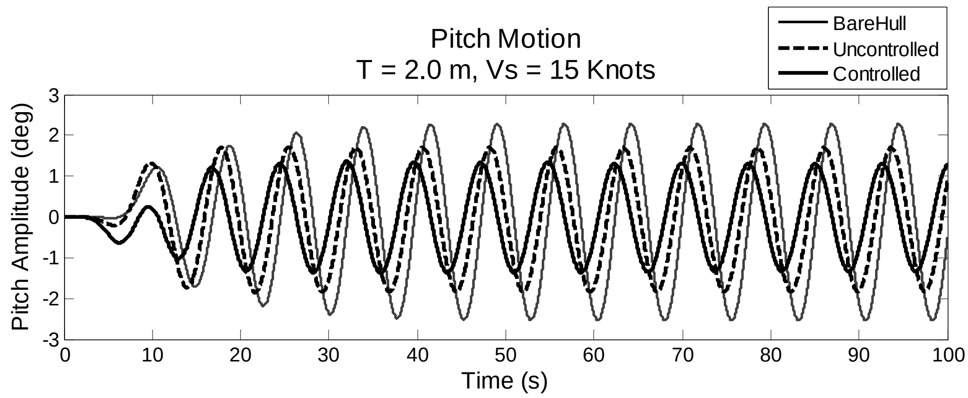


Figure 5.22 Pitch motion of Semi-SWATH vessel using controllable aft fin at wave height = 1.14 m, $\lambda/L_s = 2.4$

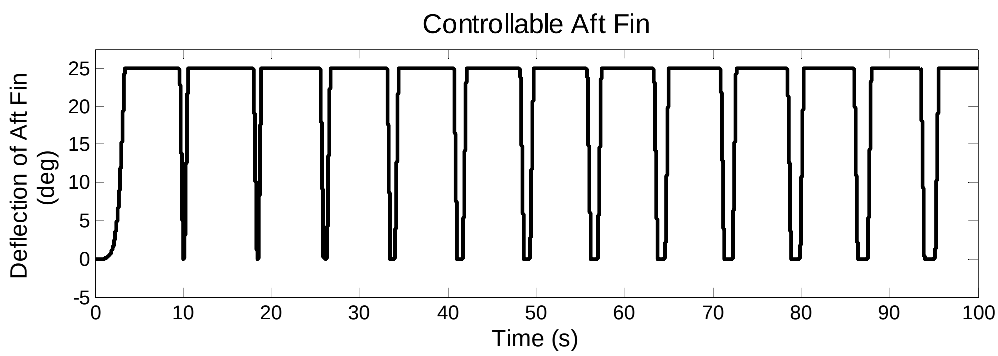


Figure 5.23 Aft fins deflection due to the application of the designed controller at vessel speed=15 Knots

Case 1E.5

Saturation of aft fin angle = 0^0 to 25^0 fore fin = 15^0

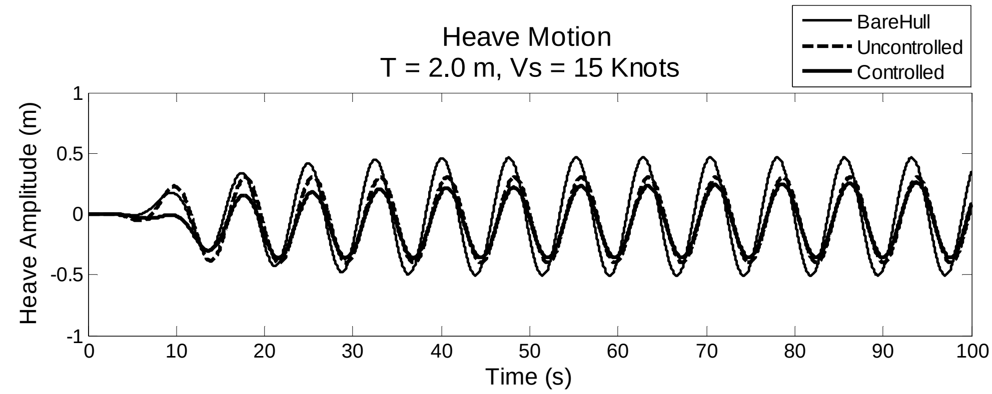


Figure 5.24 Heave motion of Semi-SWATH vessel using controllable aft fin at wave height = 1.19 m, $\lambda/L_s = 2.5$

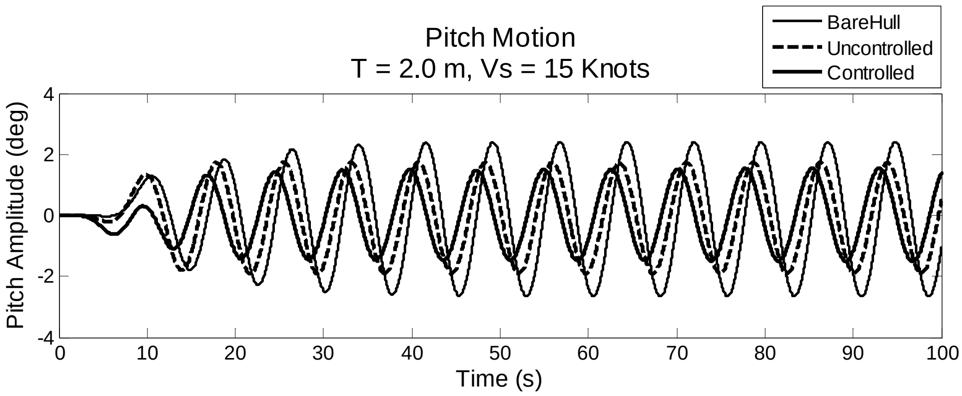


Figure 5.25 Pitch motion of Semi-SWATH vessel using controllable aft fin at wave height = 1.19 m, $\lambda/L_s = 2.5$

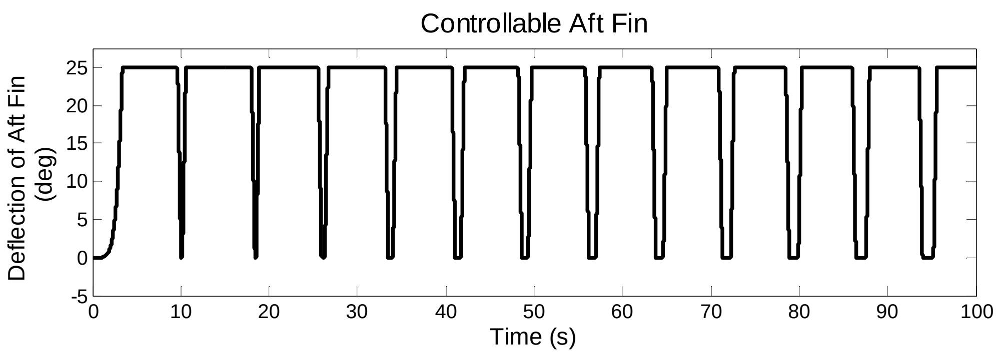


Figure 5.26 Aft fins deflection due to the application of the designed controller at vessel speed=15 Knots

5.12.2 Second Simulation Results

Case 2

Parameter for T =1.4 m, Vs = 20 knots, Inner Controller: (K_p=12.395, K_d=0.604)

Outer Controller: (K_p=275, K_i=35 and K_d=45)

Table 5.6 Semi-SWATH vessel results achieved by the application of tuning parameter of PID Controller at T=1.4 m and Vs =20 Knots

Description	**Case 2E.1**		**Case 2E.2**		**Case 2E.3**		**Case 2E.4**		**Case 2E.5**	
	WH	**λ/L_s**	**WH**	**λ/L_s**	**WH**	**λ/L_s**	**WH**	**λ/L_s**	**WH**	**λ/L_s**
	0.4285	1	0.476	1.2	0.525	1.4	0.57	1.6	0.595	1.8
Heave improvement (%)	15.393		12.211		13.663		26.775		17.882	
Pitch improvement (%)	12.473		9.628		11.264		17.152		25.771	

In Table 5.6, shown the improvement of heave and pitch motions due to application of PID Controller on the controllable aft fins (saturation of aft fin angle = 0^0 – 25^0) as compared to the fixed fore and aft fins (fore fin = 15^0 and aft fin = 25^0)

Case 2E.1

Saturation of aft fin angle = 0^0 to 25^0 fore fin = 15^0

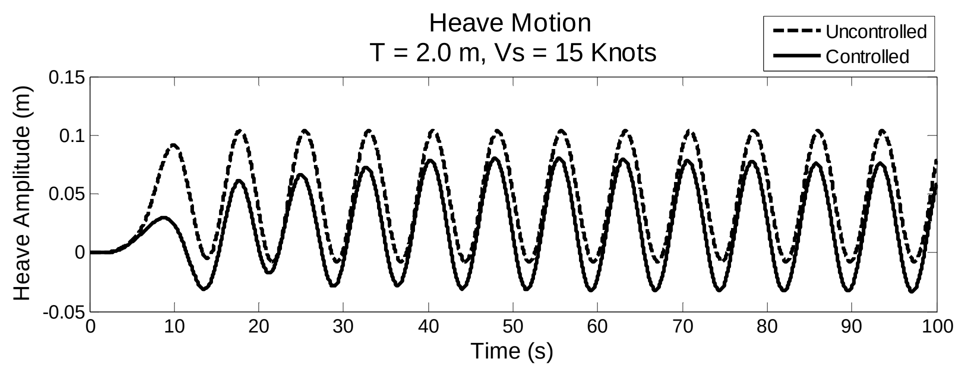


Figure 5.27 Heave motion on the Semi-SWATH vessel at wave height = 0.476 m, $\lambda/L_s = 1$

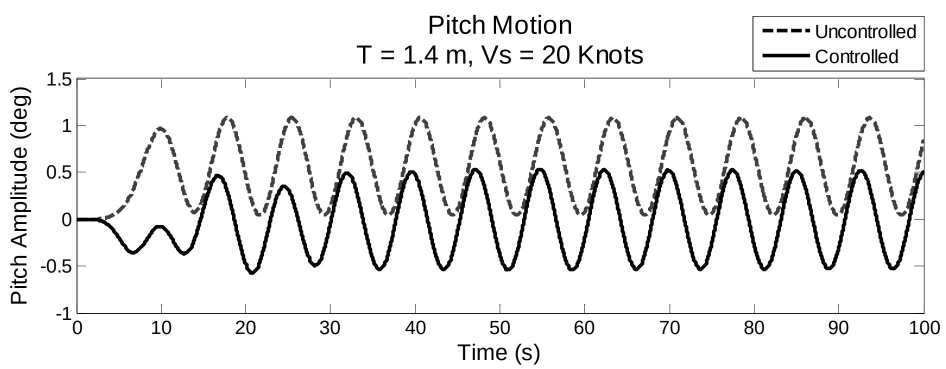


Figure 5.28 Pitch motion on the Semi-SWATH vessel at wave height = 0.476 m, $\lambda/L_s = 1$

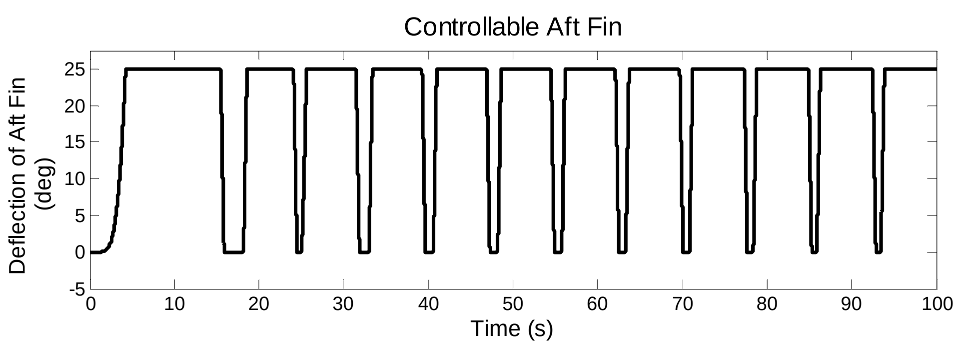


Figure 5.29 Aft fins deflection due to the application of the designed controller at vessel speed=15 Knots

Case 2E.2

Saturation of aft fin angle = 0^0 to 25^0 fore fin = 15^0

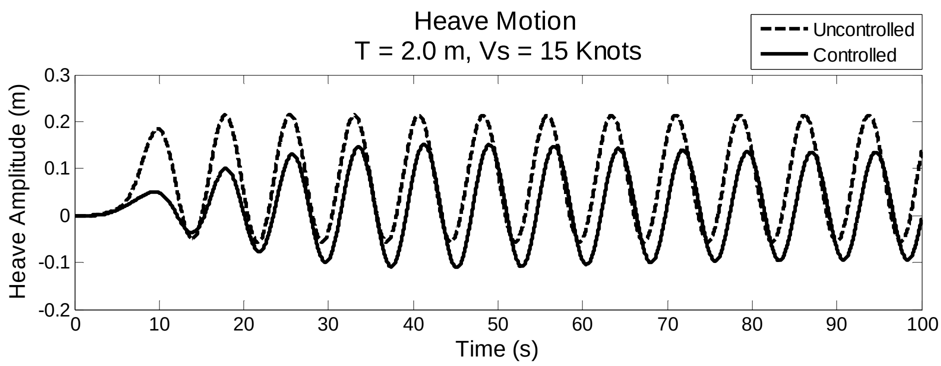


Figure 5.30 Heave motion on the Semi-SWATH vessel at wave height = 0.571 m, $\lambda/L_s = 1.2$

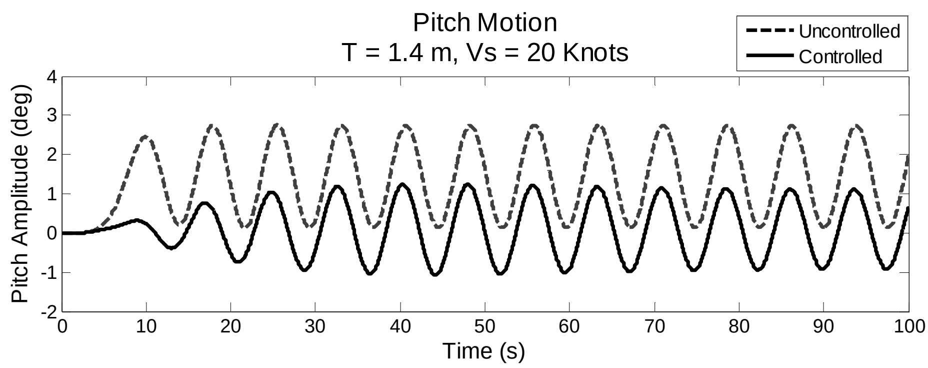


Figure 5.31 Pitch motion on the Semi-SWATH vessel at wave height = 0.571 m, $\lambda/L_s = 1.2$

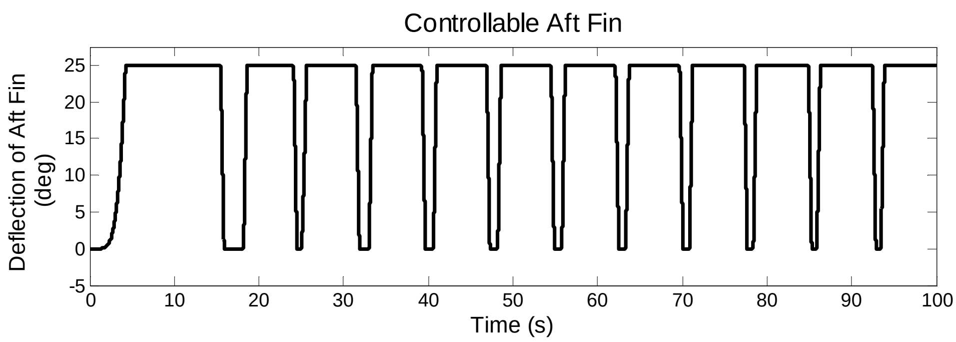


Figure 5.32 Aft fins deflection due to the application of the designed controller at vessel speed=15 Knots

Case 2E.3

Saturation of aft fin angle = 0^0 to 25^0 fore fin = 15^0

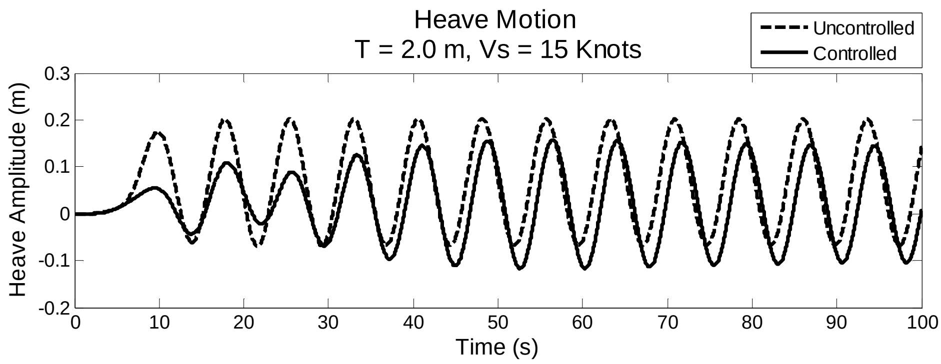


Figure 5.33 Heave motion on the Semi-SWATH vessel at wave height = 0.666 m, $\lambda/L_s = 1.4$

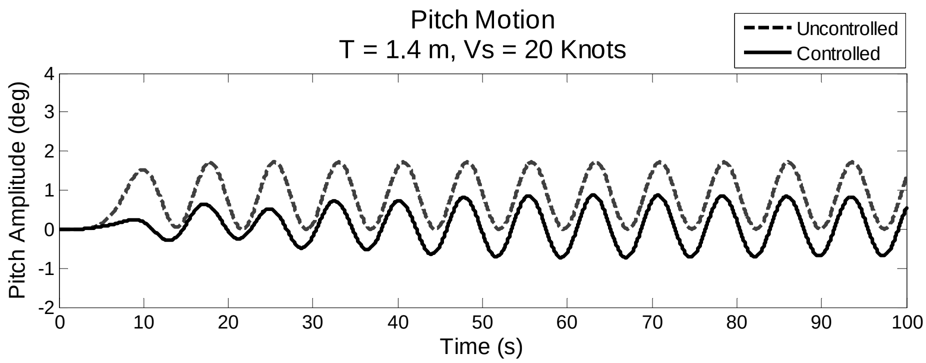


Figure 5.34 Pitch motion on the Semi-SWATH vessel at wave height = 0.666 m, λ/L_s = 1.4

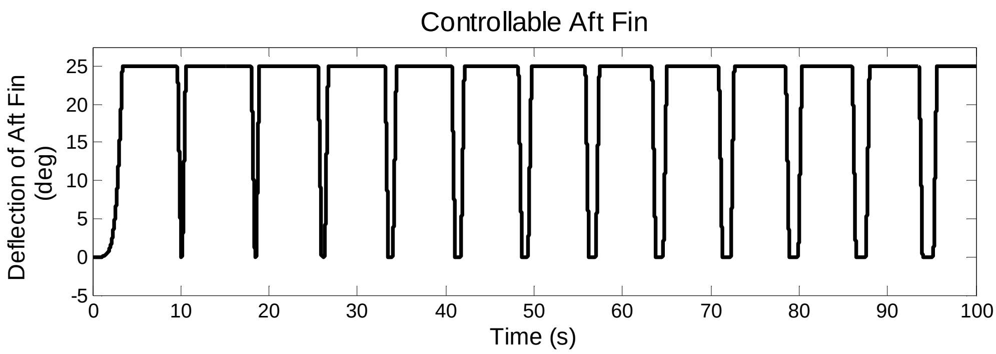


Figure 5.35 Aft fins deflection due to the application of the designed controller at vessel speed=15 Knots

Case 2E.4

Saturation of aft fin angle = 0^0 to 25^0 fore fin = 15^0

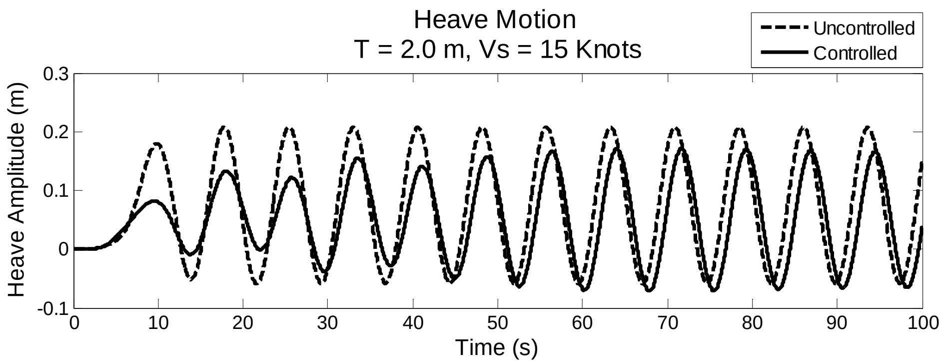


Figure 5.36 Heave motion on the Semi-SWATH vessel at wave height = 0.762 m, $\lambda/L_s = 1.6$

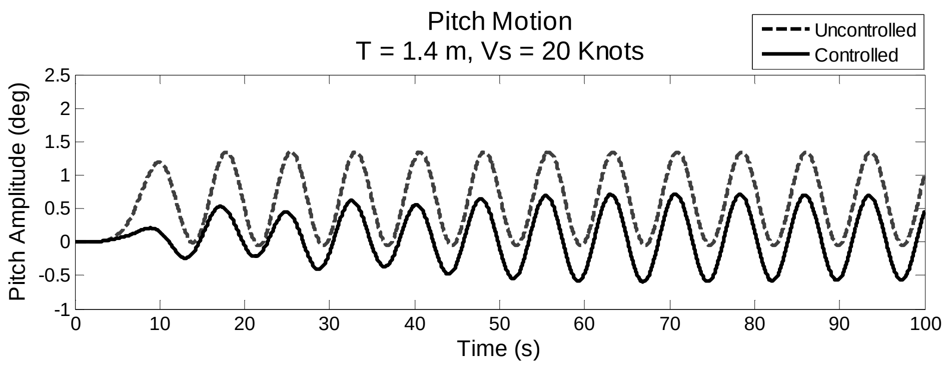


Figure 5.37 Pitch motion on the Semi-SWATH vessel at wave height = 0.762 m, $\lambda/L_s = 1.6$

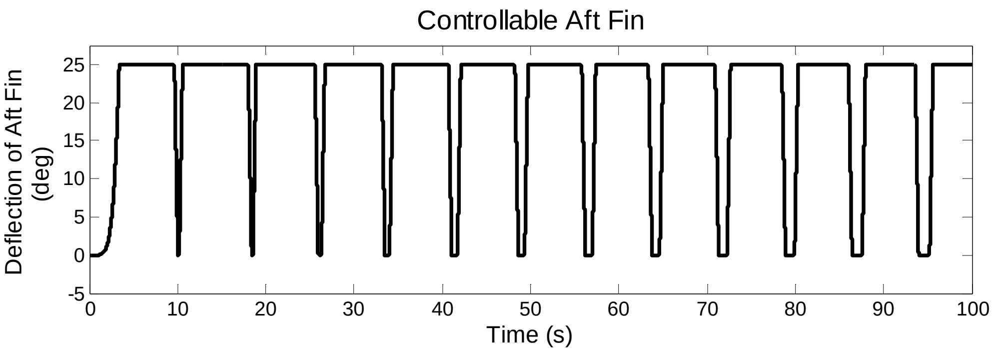


Figure 5.38 Aft fins deflection due to the application of the designed controller at vessel speed=15 Knots

Case 2E.5

Saturation of aft fin angle = 0^0 to 25^0 fore fin = 15^0

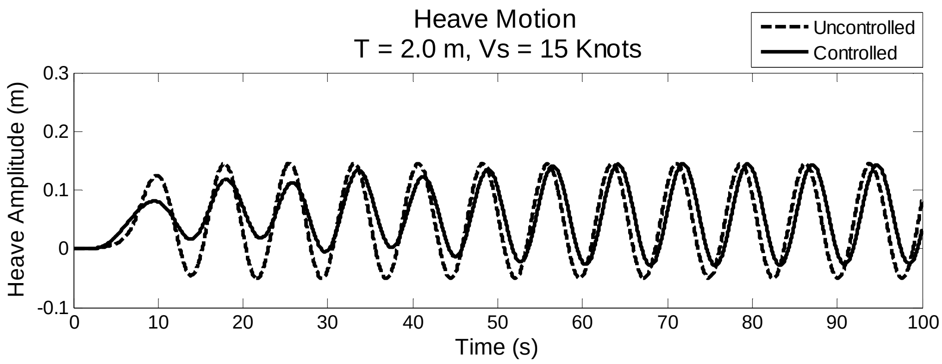


Figure 5.39 Heave motion on the Semi-SWATH vessel at wave height = 0.857 m, $\lambda/L_s = 1.8$

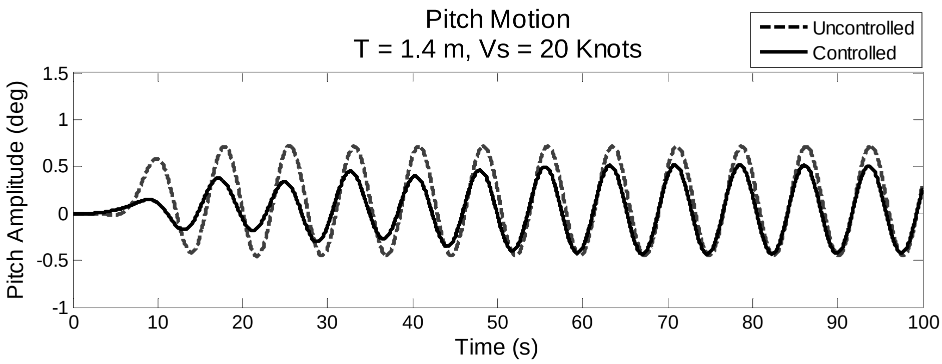


Figure 5.40 Pitch motion on the Semi-SWATH vessel at wave height = 0.857 m, $\lambda/L_s = 1.8$

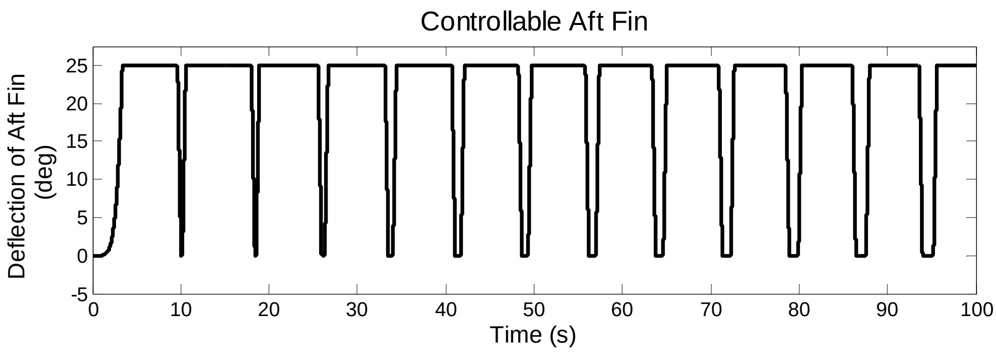


Figure 5.41 Aft fins deflection due to the application of the designed controller at vessel speed=15 Knots

5.13 Concluding Remarks

The conventional PID is initially tuned using Aström-Hagglund method and Ziegler-Nichols formula. Both of proportional and derivative gains of the PID are used to control the servomotor system. The integral gain is not used, since this servomotor system itself exhibits a small tolerable steady state error. The proposed controller utilizes a PID controller to tune the proportional gain to improve the uncontrolled system performance. The simulation results have shown that the proposed PID controller can perform improvement in terms of settling time and trajectory tracking. In terms of percent overshoot and steady state error, PID controllers perform well.

CHAPTER 6

PROCEDURE OF SEAKEEPING TEST

6.1 General

Generally, the seakeeping model tests in a towing tank are the most preferable option to validate the trusted results from the simulation programs. In addition, the quality of the tests is determined by the model scale and by the number of oscillations that the model experience in the test length (Lloyd, 1989). Consequently, it was needed a better understanding of all involved parameters of this test as well.

Accordingly, in this chapter is emphasized to describe the seakeeping model test properties such as objective of experiment, the model ship, test facility, and the procedures employed in designing the experiment.

6.2 Objective of The Experiments

Experimental seakeeping test was carried out on the Semi-SWATH model in regular head seas condition. The towed Semi-SWATH model was tested both with and without fin stabilizers. The heave and pitch motion responses of the model as the output recorded data during experiments.

Therefore, the objective of this experiment is to validate the simulation program results in case of the vertical motion (heave and pitch motions) characteristics of Semi-SWATH vessel. Then, the motion responses of heave and pitch including the significant status of the behavior of the vessel will be evaluated. Next the author present to the situation of these experiments on the RAOs of the Semi-SWATH vessel with and without fin stabilizers. Finally, the responses of the model to the same speed and waves will be compared correspondingly.

6.3 Model Test Preparation

The experiment was conducted by using the scaled model of Semi-SWATH with the scale factor of 1:10 the particulars of model will be described in sub-chapter 6.3.1. The static preparation of the model consists of establishing its weight distribution to corresponding to the full scale ship. In other words, the ratio of the masses of homologous sections of the full-scale vessel and the model should be the cube of the scale ratio. From the Froude's laws, the scaling relationships governed only consider inertia and gravity forces. Although Froude's law does not satisfy the scaling laws concerning fluid compressibility, viscosity, surface tension, and so on, the effects of these factors are insignificant as far as seakeeping characteristics are concerned.

In order to have the ship weight distribution correspond in model scale, generally the following conditions must be satisfied (for the detail see Table 6.1):

- Total weight or displacement.
- Net longitudinal position of the center of gravity for trim purposes.
- Radius of gyration

For model to have weight distribution in model scale, the model should be ballasted by arranging weights correctly to scale so the model at draught corresponding to full scale ship. The center of gravity of the model should be determined in order to corresponding to ships center of gravity. The correct height position of the center of mass of model is achieved by a heeling test of the oscillation table.

After the model ready in the needed condition researcher will proceed to next process with make the experiment set-up. In this step, researcher must make sure all equipment involve in the experiment are in good condition to make sure the result of the experiment is available to used.

6.3.1 Model Test Particulars

The semi-SWATH model was designed and decided to make the models 2.31m long, giving a scale of 1:10. It had buoyancy distribution to the Semi-SWATH hull at the scaled nominal displacements. This guaranteed the same scaled nominal wetted lengths and draughts in the variable loadings of the model test conditions. For the detail of model test particular was summarized in the Table 6.1. In addition, the variable draughts of the model test were conducted for two draught operations i.e. a light draught by 14 cm (corresponding to 53.176 kg) and a full-load draught by 20 cm (corresponding to 76.877 kg). This arrangement was set up using ballasted weight.

In this experiment, the seakeeping test will be conducted both to the bare hull model and after attached fin stabilizers. The principle dimensions corresponding to the fin stabilizers are shown in Table 6.2. Furthermore, the completed Semi-SWATH model configurations after installed of fin stabilizers were shown respectively in figure 6.1 and 6.2.

Table 6.1 The principle dimension of Semi-SWATH model test

Description	Dimension
Scale of model	1 : 10
LOA, Length of main hull (m)	2.31
Hull spacing (m)	0.8
Maximum beam diameter of main hull lower hull (m)	0.16
T_d, Deep draught (m)	0.2
T_s, Shallow draught (m)	0.14
Δ_d, Displacement at deep draught (kg)	76.877
Δ_s, Displacement at shallow draught (kg)	53.176
GM_L (m)	0.911
GM_T (m)	0.389
Length of struts at SWATH mode (m)	0.925
Maximum struts thickness at SWATH mode (m)	0.116
Radius of gyration for pitch (m)	0.578
LCG, Longitudinal center of gravity abaft (m)	0.089

Table 6.2 The principle dimension of stabilizer fins

Description	Fore Fin	Aft Fin
c, Chord (m)	0.096	0.145
s, Span (m)	0.120	0.186
Location* (m)	0.350	1.95
Depth** for deep draught (m)	0.151	0.151
Depth** for shallow draught (m)	0.092	0.092
Maximum Thickness (m)	0.015	0.023

Scale of model (1 : 10)
* Distance from the main hull nose to the quarter-chord point
** Distance from the waterline to the chord line

Figure 6.1 Plane View of Semi-SWATH Model

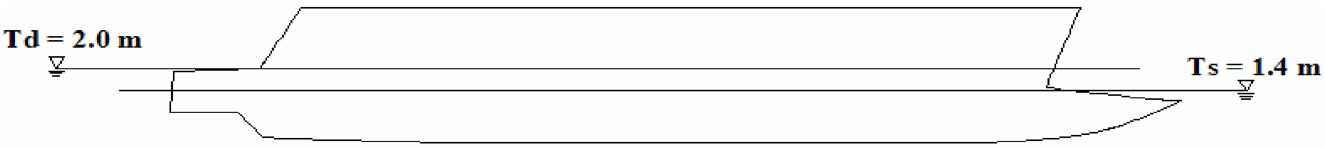


Figure 6.2 Side View of Semi-SWATH Model

6.4 Experiments Apparatus and Facility

6.4.1 Towing Tank

The Towing Tank at the Marine Technology Laboratory, Universiti Teknologi Malaysia has dimensions of 120 m length, 4 m width and, 2.5 m depth. Now the tank is being used actively for various types of model testing. The detail of Towing Tank facility of Marine Technology Laboratory is shown in figure 6.3 and figure 6.4

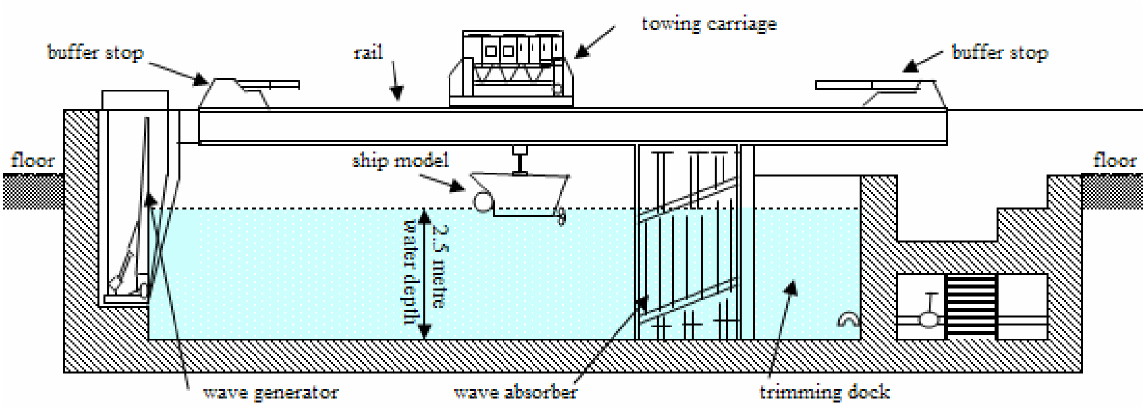


Figure 6.3 Side view of Towing Tank

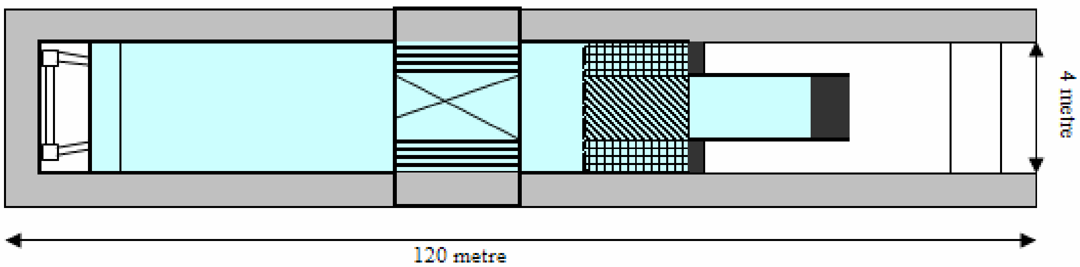


Figure 6.4 Plane view of Towing Tank

6.4.2 Towing Carriage

The maximum speed of Towing Carriage is 5 m/s. At maximum acceleration, 1 m/sec^2, the carriage can achieve a minimum measuring time of 10 seconds at the maximum speed.

6.4.3 Wave Generator

For Seakeeping model testing, the waves are generated by a wave flap at the end of the Towing Tank in UTM with are to generating a long crested regular and random waves, parallel to the wave flap. The user needs to calculate the desired waves for model test by wave calculation software before generate the waves through wave generator. The wave calculation software can be operated from the terminals and transform into desire waves and wave spectrum through the flap actuators. The capabilities of wave generators to generate regular waves are at period range 0.5 sec to 2.5 sec with a wave height corresponding to a maximum steepness of 1/10 in a period range of 0.5 to 1.7sec. The created wave is absorb by a wave absorber at the other end of the tank.

6.5 Experiments Setup

The experiment arrangement for heave and pitch motion of twin-hull model test is identically as same as mono-hull model test was shown in figure 6.5. To measure heave and pitch motion for model, the model will attach to the towing carriage by air struck with the help of the base plate at LCG of the model. The couple of this motion is measured by potentiometer. The model will tow by towing carriage for seakeeping test with forward speed. The heave and pitch motion will be measure by potentiometers. The potentiometer is connected to the D.A.A.S of towing tank and the signals are simultaneously digitize and record on paper chart.

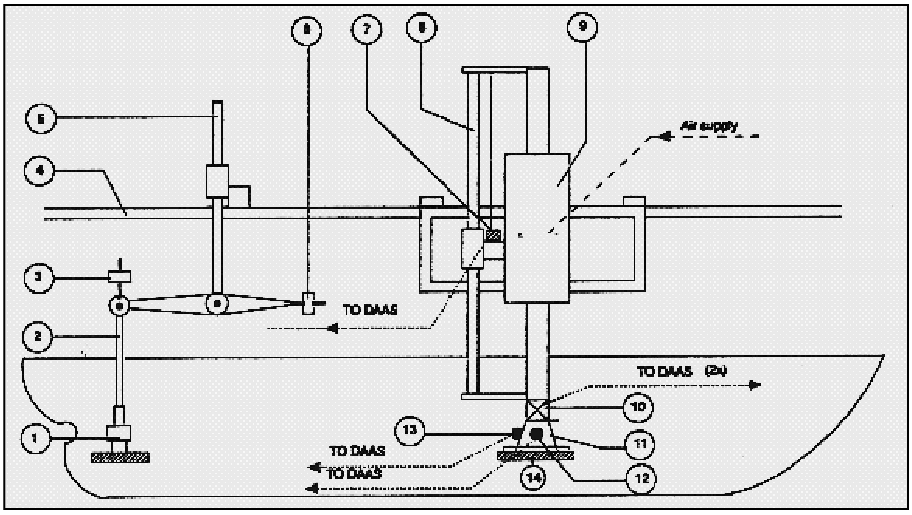


Figure 6.5 The arrangement of airstrut and towing guide with ship model (MARIN, 1997b)

The resistance and movement dynamometer consists of two main parts such as a vertical cylinder moving in low friction aerostatic bearing designated as airstrut and a towing guide. In the Figure 6.5, the airstrut is shown on the right and the towing guide on the left and the following components are indicated:

1. Base plate for towing guide
2. Towing guide
3. Balancing weight
4. Measuring rail of the carriage
5. Vertical adjustment to the towing guide
6. Balancing weight
7. Heave potentiometer
8. Bar with auxiliary air bearing
9. Main air bearing
10. Longitudinal and transverse force transducer
11. Gimbals
12. Pitch potentiometer
13. Roll potentiometer
14. Base plate with quick release coupling for airstrut

During the test, the gimbals attached to the base plate of the model consists the pitch potentiometer to measure the pitch degree and the heave potentiometer was used to measured the heave amplitude of the model when the regular waves was generated. The both potentiometer was connected to the potentiometer conditioner respectively in the signal conditioning system (SCS) box of the Data Acquisition and Analyzing System (DAAS) for transferring the signal for the motion. A wave probe located at the front edge at the towing carriage was used to measure the wave height throughout the experiments. Before running the experiments, local calibration of the accelerometers, heave potentiometer, pitch potentiometer and wave probe were carried out. The DAAS stores the data picked-up by the accelerometers and the potentiometers on the hard disc and then the data were sent to an off-line computer for analysis by using the MVR Off-line Analysis Package of the DAAS. Figure 6.6 shows the block diagram of Data Acquisition Analysis System (DAAS).

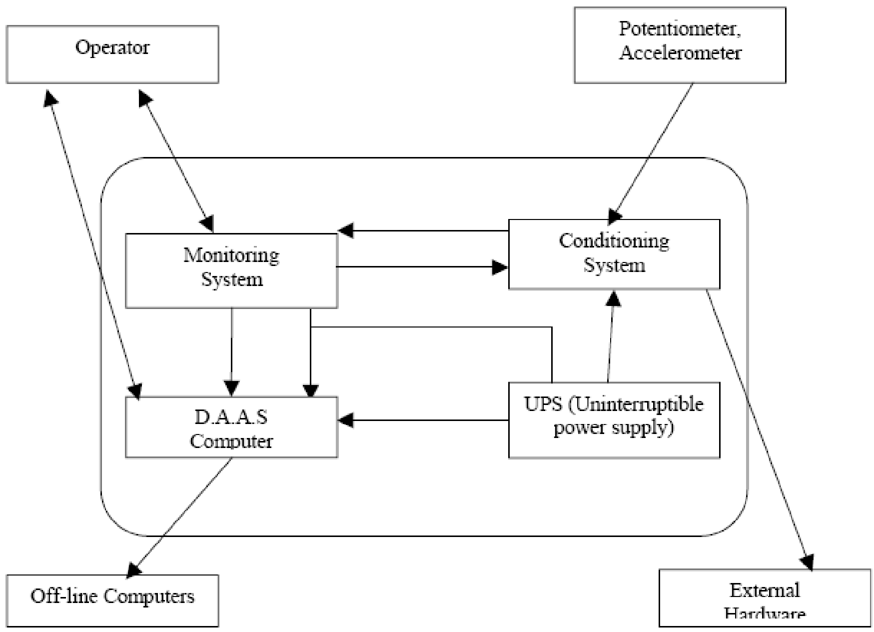


Figure 6.6 The block diagram of Data Acquisition Analysis System (DAAS)

Since the research is going to validate the seakeeping performance of Semi-SWATH vessel from the simulation program, so this could investigate the research scope to the study on the bare hull vessel and after attached fin stabilizers.

6.6 Experiment Condition

Calculation conditions for the Semi-SWATH model Seakeeping prediction are base on the requirement given as follow:

- Motion amplitudes for heave and pitch in head sea with forward speed of 20 knots. Froude Number, Fn = 0.684
- Motion amplitudes for heave and pitch in head sea with forward speed of 20 knots Froude Number, Fn = 0.512

The ship motion prediction in waves was carried out at the following environmental condition:

- A range of regular wave lengths: $0.5 \leq \lambda/L \leq 2.5$
- Steepness of the incident wave: $H/\lambda = 1/50$

For measuring the heave and pitch motion, the model is arranged to head sea condition with regular waves of various periods. The model is tested at two different Froude Number such as 0.684 and 0.512 with two variable draughts.

6.7 Description of Data Test Analysis

Since in the regular wave analysis all data should be nominally periodic in time, the basic procedure was to perform a harmonic analysis on all data channel for extracting the fundamental harmonics and for determining the transfer functions between the wave input and motion response output.

The heave displacement, pitch degree, accelerations and wave amplitude signals in time-domain was picked up by the heave potentiometer, pitch potentiometer and accelerometers were recorded. Using the MVR Off-line Analysis Package of the DAAS then processed the raw signals. By using the MVR software, the raw signals picked up were processed by the option named plotsim subjected in Output to plot the signals of all the measurements in time-domain. The option To

ASCII subjected in Tools also in the MVR software was used to transfer the data in type of Tab.

In the data test analysis, the wave input and motion response output of heave and pitch will be analyzed using Response Amplitude Operators (RAOs). This term is to describe and determine the significant heave amplitude and determine significant pitch amplitude divide by wave amplitude. Presented quantities are made non-dimensional, as follows: heave and pitch divided by wave amplitude and Standard International Metric Units were used throughout. The formulation of the RAOs is presented in equation 6.1.

$$\frac{u_a}{\xi_a} = \frac{\text{motion amplitude}}{\text{wave ampitude}} \tag{6.1}$$

Where; u_a can be defined as heave amplitude (m) or pitch amplitude (degree).

Finally, the heave and pitch responses per unit wave height are similarly plotted for every cycle of seakeeping tests. Included in the diagrams are curves showing the ratio of the heave and pitch motions of the Semi-SWATH model with fins and without fins. The advanced analysis of the motion responses the RAO results are plotted against to the wavelength and ship length ratio ($\frac{\lambda}{L_{Ship}}$).

6.8 Concluding Remarks

The seakeeping test of Semi-SWATH model is carried out in Towing Tank of Marine Technology Laboratory and restricted only to regular head condition. In this experiments, the range ratio of wavelengths to the model length ($\frac{\lambda}{L_{Ship}}$) is about 0.5-2.5 with wave height to wavelength ratio at the experiments was 1/25. The measured data from all channels were digitized at a rate of 25 samples per second on each channel. Where, the total number of encounter cycles collected on each run was typically between 5 and 10 times. Then, all data channels are extracted using MVR software. Finally, the result data analysis of heave and pitch responses will be plotted per unit wave height with fins and without fins as the Response Amplitude Operators (RAOs) against to the wavelength and ship length ratio ($\frac{\lambda}{L_{Ship}}$).

CHAPTER 7

VALIDATION

7.1 General

In this chapter, a comparison between simulation results with the experimental data provided by the Semi-SWATH model test at Towing Tank of Marine Technology Laboratory, Universiti Teknologi Malaysia is carried out to validate the simulation program. Two responses are considered in the comparison; those are heave and pitch motions.

7.2 Comparison of Experimental and Simulation Results

For comparison, the seakeeping model test of Semi-SWATH with 76.877 tones and 53.176 tones displacement (deep draught and shallow draught respectively) had been chosen for the investigation of ship motion in waves. There are two cases specified in the comparison. The summary of the comparison condition is shown in Table 7.1

For all the conditions both simulations and experiments, the operational draught is taken as 1.4 m (shallow draught) and 2.0 m (deep draught). Other conditions had been taken are; wave length to ship length ratio is taken as $\lambda_W / L = 1 - 2.5$ and wave height to length ratio is taking as $H_W / \lambda_W = 1/25$ and angle of attack for aft and fore fins. They were taken as the main references to be compared inherent to the heave and pitch motions of Semi-SWATH vessel. The required hydrodynamic coefficients are estimated through published literatures by Maimun (2001).

Table 7.1, Comparison of simulation and experiment condition for Semi-SWATH vessel

Case	Course	Speed (Knots)	Draught	Condition	Fixed Fin Angle Aft	Fore
1A,1B,1C,1D and 1E	Head Sea	15	Deep	with and without fins	5, 10, 15, 20, and 25	15
2A,2B,2C,2D and 2E	Head Sea	20	Shallow	with fins	5, 10, 15, 20, and 25	15

The summary of the comparison between simulations and experiments of motion responses both heave and pitch are shown in Table 7.2 to Table 7.9 and Table 7.15 to 7.22. The heave and pitch responses per unit wave height of all case 1 and 2 are similarly plotted. It was shown in Figure 7.1 to 7.14 and Figure 7.65 to 7.78. Included in the diagrams are curves showing the ratio of the motion of the model with fins to that without fins see figure 7.1 to 7.4. Besides that, all time histories for all cases 1 and 2 were shown in Figure 7.15 to 7.64 and Figure 7.79 to 7.128.

In overall, it is found that the comparison between the simulation and experimental results shows that the pitch and heave amplitude between simulation and experiment are found to be quite similar or can be predicted successfully. Its means, for all cases 1 and 2, the theoretical results found generally agree with the experimental results quite well for Semi-SWATH vessel model running in head sea waves ($\theta=180^0$). But, the amplitudes obtained from experiments for case 1 and 2 generally are higher than simulations values.

For all cases 1 and 2, all simulation and experiment results show that the maximum of motion responses both heave and pitch were obtained. For all cases 1, the peak motion responses both heave and pitch motions wave excitation were occurred at wave height 0.952 m with wave length to ship length ratio as 2.0. It can be seen at figure 7.1 to 7.4. For all cases 2, the peak of heave motion responses generally were occurred at wave height 0.571 m with wave length to ship length ratio as 1.2. It can be seen at figure 7.65. But, for the peak of pitch motion responses were occurred at wave height 0.666 m with wave length to ship length ratio as 1.4. It can be seen at figure 7.66.

The results of the model tests for various angle of fins compared to bare hull vessel in case 1 showed that the motions of heave and pitch were reduced by a maximum about;

1. Case 1A
   Case 1A.4; heave = 28.522%, pitch = 26.134%
2. Case 1B
   Case 1B.1; heave = 34.02%, pitch = 22.00%
3. Case 1C
   Case 1C.1; heave = 32.14%, Case 1C.5; pitch = 37.95%
4. Case 1D
   Case 1D.5; heave = 39.18%, pitch = 36.52%
5. Case 1E
   Case 1E.2; heave = 37.965%, Case 1E.1; pitch = 35.81%

7.3 Case 1

7.3.1 Maximum and Minimum Values of Heave and Pitch Motion at T=2.0m and Vs=15 Knots

Table 7.2 Summary of heave motion values (experimentally) at various angles of fins

HEAVE MOTION (EXPERIMENT)											
WH-λ/Ls		**0.857**	**1.800**	**0.952**	**2.000**	**1.050**	**2.200**	**1.140**	**2.400**	**1.190**	**2.500**
Fin Angle		Max (m)	Min (m)	Max (m)	Min (m)	Max (m)	Min (m)	Max (m)	Min (m)	Max (m)	Min (m)
Aft	Fore										
5	15	0.321	-0.431	0.424	-0.552	0.441	-0.568	0.432	-0.538	0.433	-0.538
10	15	0.286	-0.408	0.383	-0.527	0.390	-0.522	0.390	-0.510	0.397	-0.508
15	15	0.265	-0.393	0.408	-0.458	0.359	-0.494	0.367	-0.492	0.373	-0.491
20	15	0.225	-0.331	0.333	-0.470	0.334	-0.461	0.332	-0.443	0.348	-0.456
25	15	0.197	-0.279	0.302	-0.416	0.318	-0.434	0.304	-0.405	0.308	-0.405
Bare Hull		0.369	-0.472	0.515	-0.641	0.559	-0.689	0.581	-0.704	0.578	-0.695

Table 7.3 Summary of heave motion values (theoretically) at various angles of fins

HEAVE MOTION (THEORY)											
WH-λ/Ls		**0.857**	**1.800**	**0.952**	**2.000**	**1.050**	**2.200**	**1.140**	**2.400**	**1.190**	**2.500**
Fin Angle		Max (m)	Min (m)	Max (m)	Min (m)	Max (m)	Min (m)	Max (m)	Min (m)	Max (m)	Min (m)
Aft	Fore										
5	15	0.272	-0.366	0.357	-0.465	0.392	-0.505	0.378	-0.471	0.389	-0.484
10	15	0.223	-0.318	0.319	-0.440	0.355	-0.474	0.351	-0.459	0.335	-0.428
15	15	0.298	-0.442	0.267	-0.377	0.318	-0.436	0.280	-0.376	0.298	-0.393
20	15	0.172	-0.253	0.273	-0.385	0.294	-0.404	0.297	-0.397	0.286	-0.375
25	15	0.148	-0.209	0.249	-0.340	0.276	-0.378	0.259	-0.344	0.243	-0.320
Bare Hull		0.356	-0.46	0.4259	-0.537	0.501	-0.619	0.5524	-0.669	0.546	-0.656

Table 7.4 Summary of pitch motion values (experimentally) at various angles of fins

PITCH MOTION (EXPERIMENT)											
WH-λ/Ls		**0.857**	**1.800**	**0.952**	**2.000**	**1.050**	**2.200**	**1.140**	**2.400**	**1.190**	**2.500**
Fin Angle		Max	Min	Max	Min	Max	Min	Max	Min	Max	Min
Aft	**Fore**	(^{0})	(^{0})	(^{0})	(^{0})	(^{0})	(^{0})	(^{0})	(^{0})	(^{0})	(^{0})
5	15	1.755	-2.014	2.221	-2.499	2.316	-2.566	2.116	-2.343	2.412	-2.637
10	15	1.554	-1.987	1.971	-2.522	2.061	-2.469	2.037	-2.492	2.056	-2.532
15	15	1.476	-1.776	1.973	-2.331	2.005	-2.340	2.003	-2.321	2.012	-2.308
20	15	1.372	-1.549	1.987	-2.210	1.935	-2.091	1.856	-1.975	1.974	-2.158
25	15	1.241	-1.360	1.861	-2.040	1.749	-1.851	1.741	-1.807	1.746	-1.861
Bare Hull		1.900	-2.151	2.374	-2.689	2.493	-2.786	2.291	-2.537	2.409	-2.537

Table 7.5 Summary of pitch motion values (theoretically) at various angles of fins

PITCH MOTION (THEORY)											
WH-λ/Ls		**0.857**	**1.800**	**0.952**	**2.000**	**1.050**	**2.200**	**1.140**	**2.400**	**1.190**	**2.500**
Fin Angle (^{0})		Max	Min	Max	Min	Max	Min	Max	Min	Max	Min
Aft	**Fore**	(^{0})	(^{0})	(^{0})	(^{0})	(^{0})	(^{0})	(^{0})	(^{0})	(^{0})	(^{0})
5	15	1.560	-1.790	1.944	-2.186	2.072	-2.296	1.852	-2.050	2.199	-2.405
10	15	1.286	-1.793	1.673	-2.474	1.848	-2.305	1.796	-2.303	1.691	-2.242
15	15	1.216	-1.462	1.776	-2.098	1.843	-2.153	1.650	-1.912	1.634	-1.876
20	15	1.238	-1.351	1.600	-1.730	1.561	-1.673	1.489	-1.585	1.577	-1.678
25	15	1.090	-1.140	1.677	-1.754	1.589	-1.634	2.054	-2.075	1.302	-1.323
Bare Hull		1.9026	-2.182	2.136	-2.415	2.3614	-2.64	2.577	-2.856	2.7098	-2.989

7.3.2 Response Amplitude Operators (RAOs) of Heave and Pitch Motion T=2.0m and Vs=15 Knots

Table 7.6 Summary of Response Amplitude Operators (RAOs) for heave motion (experimentally) at various angles of fins

Fin Angle (^{0})		**λ/Ls**	**HEAVE RAOs (EXPERIMENT) (deg/m)**				
Aft	**Fore**						
5	15	1.80	0.877	1.025	0.960	0.851	0.816
10	15	2.00	0.809	0.956	0.869	0.790	0.761
15	15	2.20	0.768	0.911	0.812	0.753	0.726
20	15	2.40	0.649	0.844	0.757	0.680	0.676
25	15	2.50	0.556	0.754	0.716	0.622	0.599
Bare Hull			0.981	1.215	1.188	1.128	1.07

Table 7.7 Summary of Response Amplitude Operators (RAOs) for heave motion (theoretically) at various angles of fins

Fin Angle (^{0})		**λ/Ls**	**HEAVE RAOs (THEORY) (deg/m)**				
Aft	**Fore**						
5	15	1.80	0.744	0.863	0.854	0.745	0.734
10	15	2.00	0.632	0.797	0.790	0.711	0.641
15	15	2.20	0.580	0.732	0.718	0.632	0.581
20	15	2.40	0.496	0.691	0.665	0.608	0.556
25	15	2.50	0.417	0.618	0.623	0.529	0.473
Bare Hull			0.953	1.173	1.176	1.091	1.031

Table 7.8 Summary of Response Amplitude Operators (RAOs) for pitch motion (experimentally) at various angles of fins

Fin Angle (^{0})		**λ/Ls**	**PITCH RAOs (EXPERIMENT) (deg/m)**				
Aft	**Fore**						
5	15	1.80	4.398	4.958	4.649	4.199	4.110
10	15	2.00	4.132	4.719	4.315	3.973	3.856
15	15	2.20	3.795	4.521	4.138	3.794	3.630
20	15	2.40	3.408	4.410	3.834	3.361	3.472
25	15	2.50	3.035	4.098	3.428	3.112	3.031
Bare Hull			4.73	5.318	5.028	4.45	4.4

Table 7.9 Summary of Response Amplitude Operators (RAOs) for pitch motion (theoretically) at various angles of fins

Fin Angle (^{0})		**λ/Ls**	**PITCH RAOs (THEORY) (deg/m)**				
Aft	**Fore**						
5	15	1.80	3.910	4.450	4.170	3.510	3.480
10	15	2.00	3.593	4.250	3.840	3.250	3.150
15	15	2.20	3.125	3.920	3.560	2.970	2.950
20	15	2.40	3.021	3.690	3.180	2.697	2.736
25	15	2.50	2.603	3.250	2.770	2.440	2.206
Bare Hull			4.7659	5.03	4.76	4.36	4.26

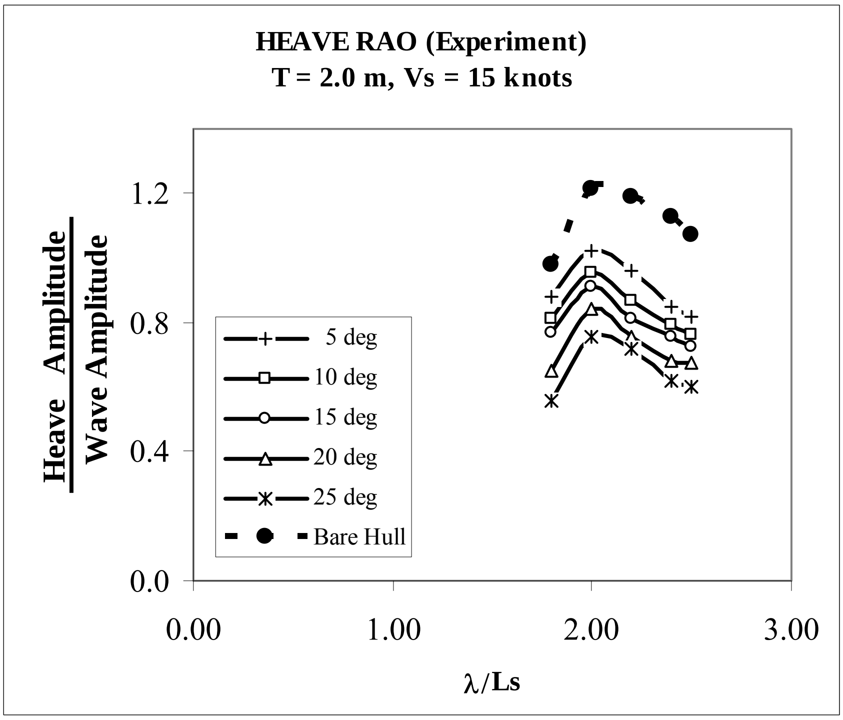


Figure 7.1 RAOs of heave for bare hull vessel and with various angles of fins (experimentally)

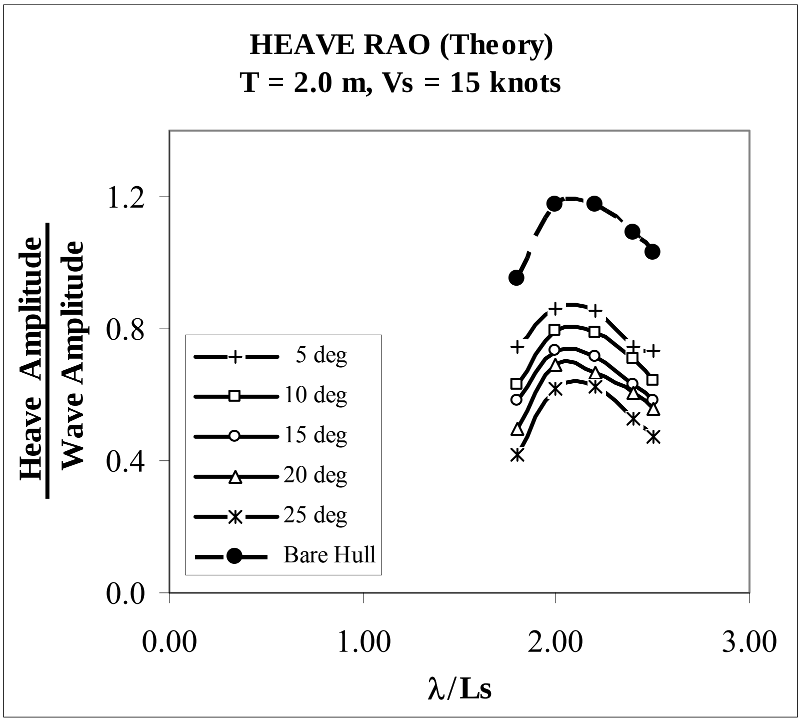


Figure 7.2 RAOs of heave for bare hull vessel and with various angles of fins (theoretically)

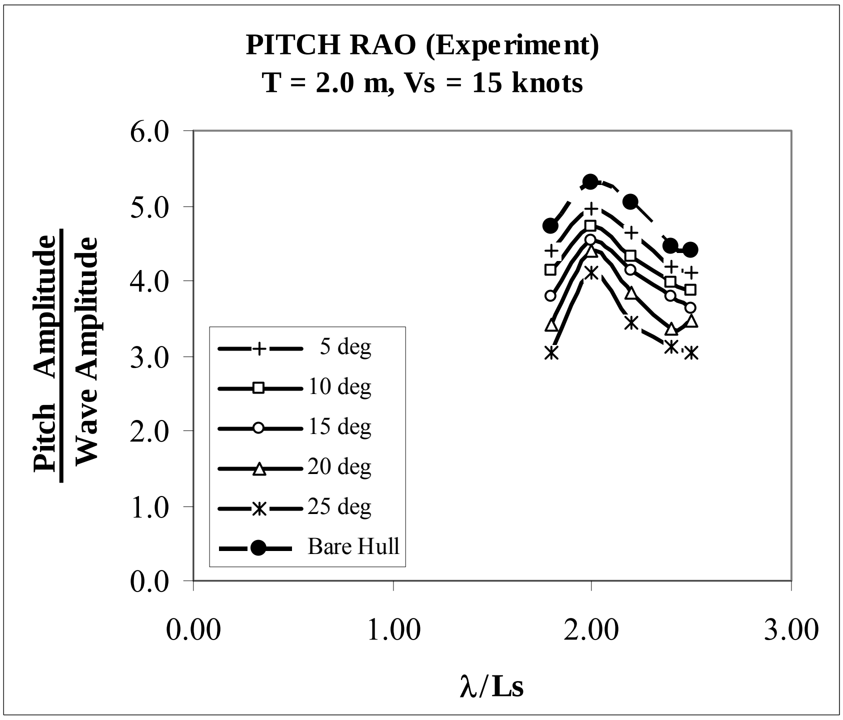


Figure 7.3 RAOs of pitch for bare hull vessel and with various angles of fins (experimentally)

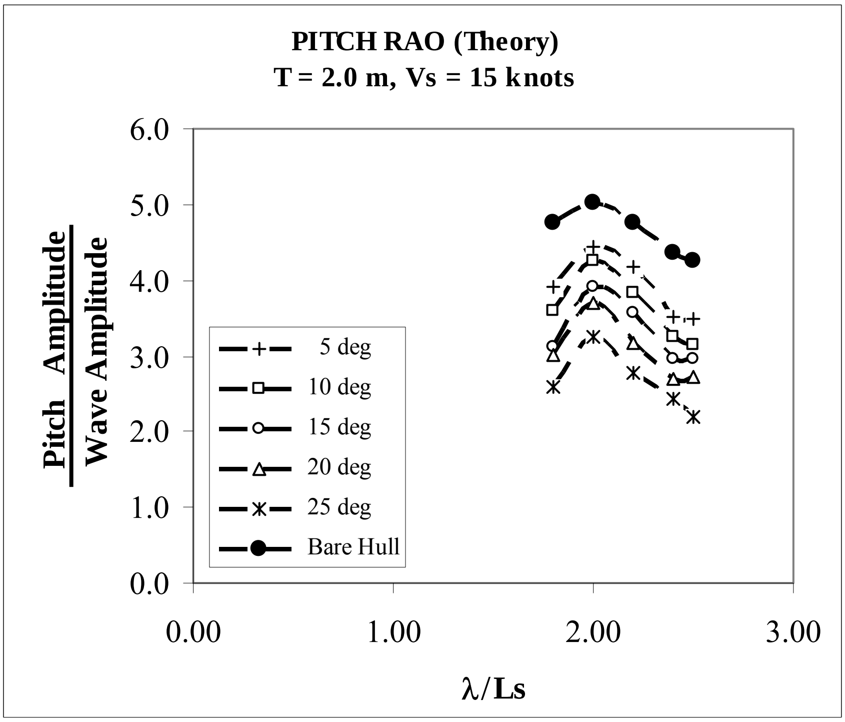
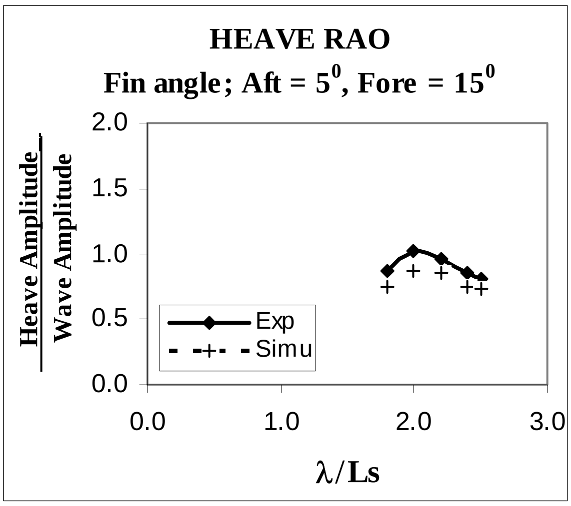


Figure 7.4 RAOs of pitch for bare hull vessel and with various angles of fins (theoretically)



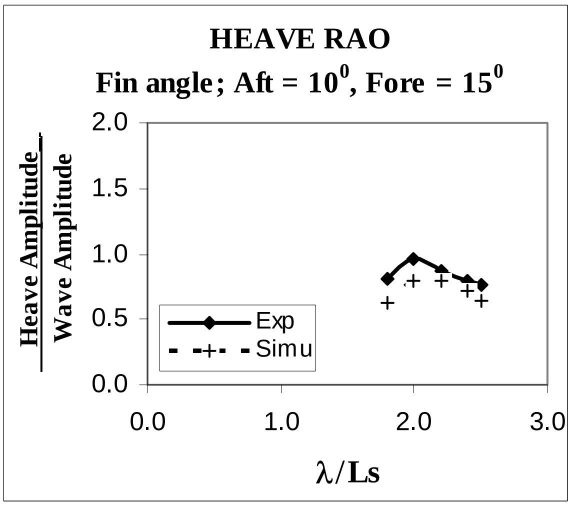


Figure 7.5 RAOs comparison of heave at fin angle; aft = 5^0 and fore = 15^0

Figure 7.6 RAOs comparison of heave at fin angle; aft = 10^0 and fore = 15^0

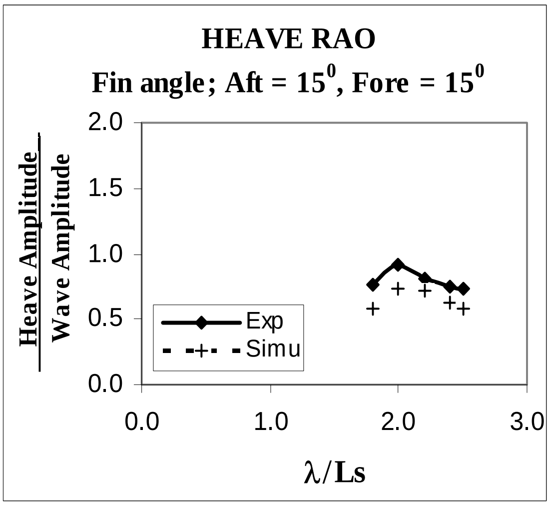


Figure 7.7 RAOs comparison of heave

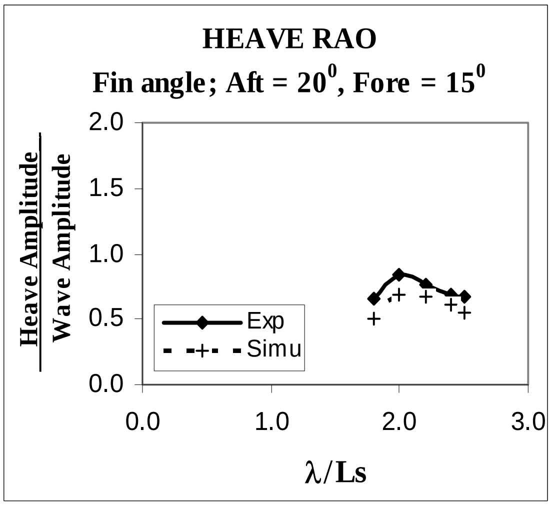


Figure 7.8 RAOs comparison of heave

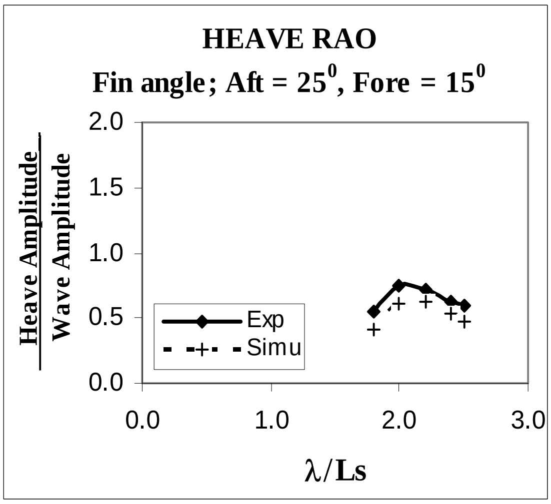


Figure 7.9 RAOs comparison of heave

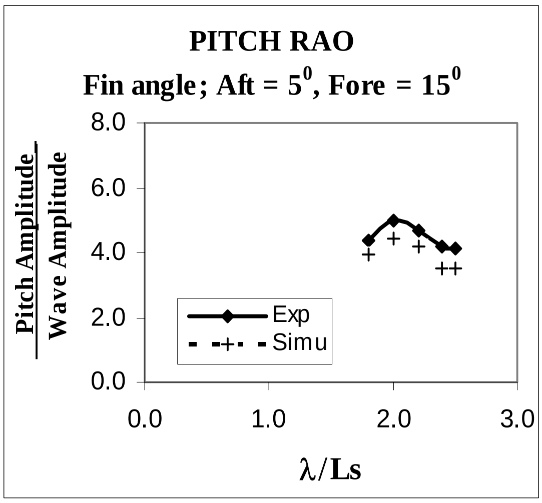


Figure 7.10 RAOs comparison of pitch

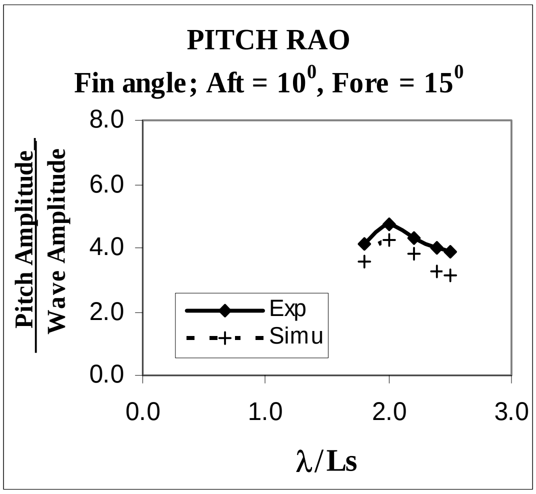


Figure 7.11 RAOs comparison of heave

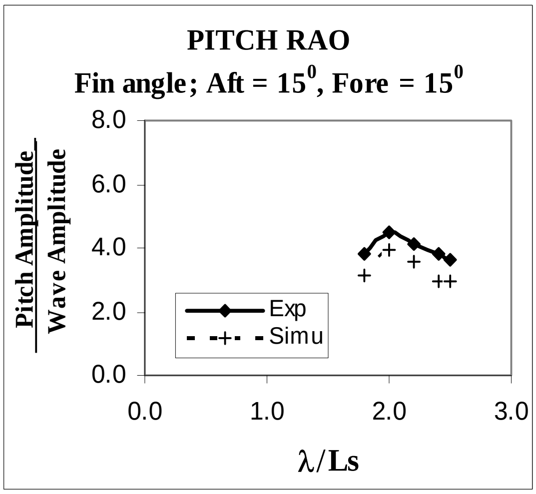


Figure 7.12 RAOs comparison of pitch

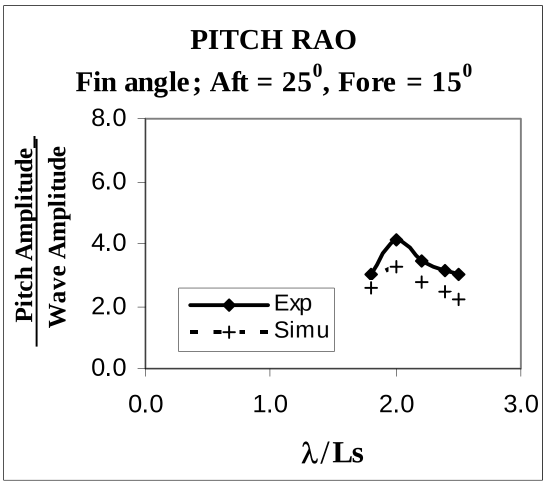


Figure 7.13 RAOs comparison of pitch

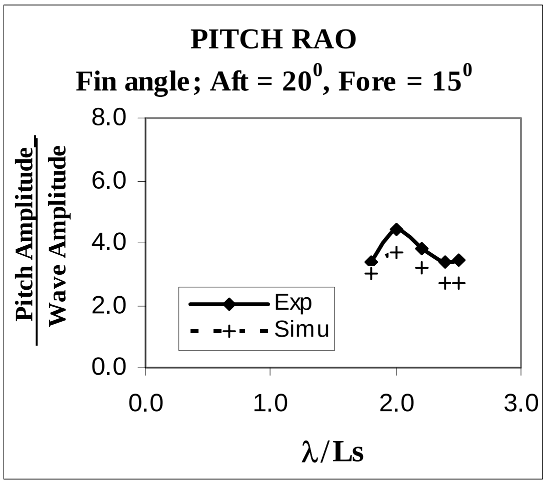


Figure 7.14 RAOs comparison of pitch

7.3.3 Time-Histories of Heave and Pitch Motion T=2.0m and Vs=15 Knots

Case 1A

Fin Angle: Aft = 5^0 and Fore = 15^0

Table 7.10 Summary of heave and pitch improvement (%) for various angles of aft fin and fixed fore fin (experimentally).

Description	**Case 1A.1**		**Case 1A.2**		**Case 1A.3**		**Case 1A.4**		**Case 1A.5**	
	WH	**λ/L_s**	**WH**	**λ/L_s**	**WH**	**λ/L_s**	**WH**	**λ/L_s**	**WH**	**λ/L_s**
	0.857	**1.8**	**0.952**	**2.0**	**1.05**	**2.2**	**1.14**	**2.4**	**1.19**	**2.6**
Heave improvement (%)	20.58		15.98		18.96		28.522		27.751	
Pitch improvement (%)	16.94		10.98		12.15		26.134		20.282	

Case 1A.1

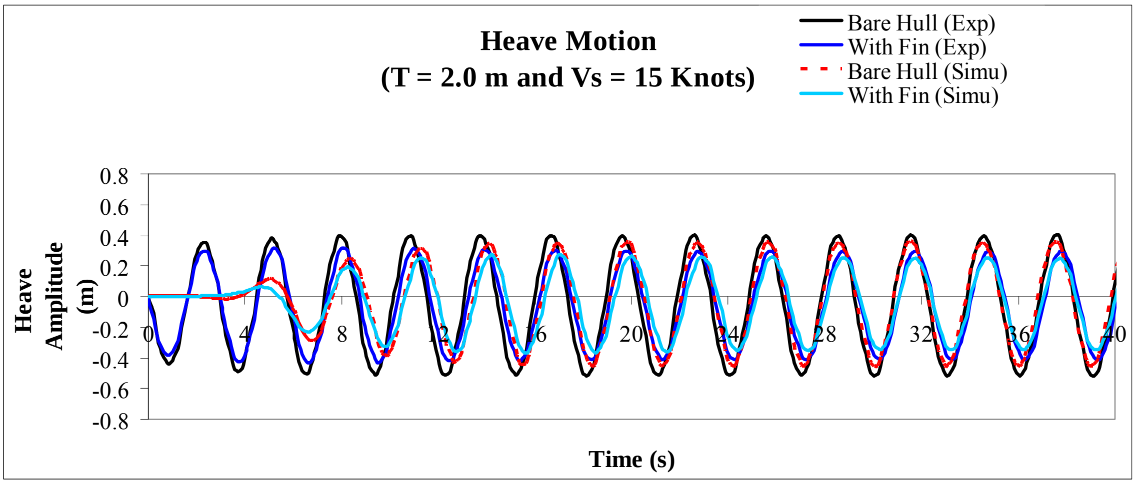


Figure 7.15 Heave motion on the Semi-SWATH vessel wave height = 0.857 m, λ/L_s = 1.8

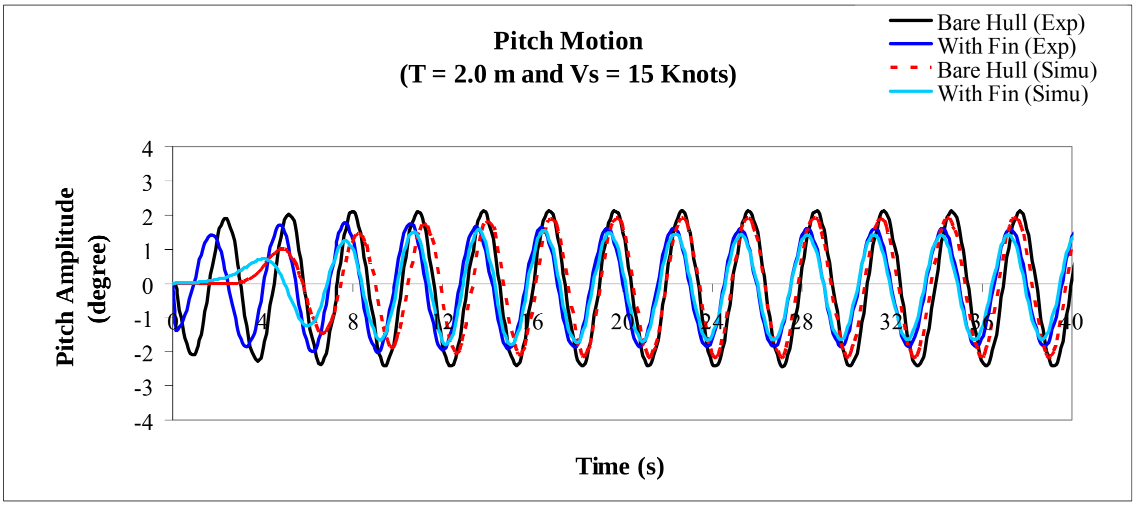


Figure 7.16 Pitch motion on the Semi-SWATH vessel wave height = 0.857 m, λ/L_s = 1.8

Case 1A.2

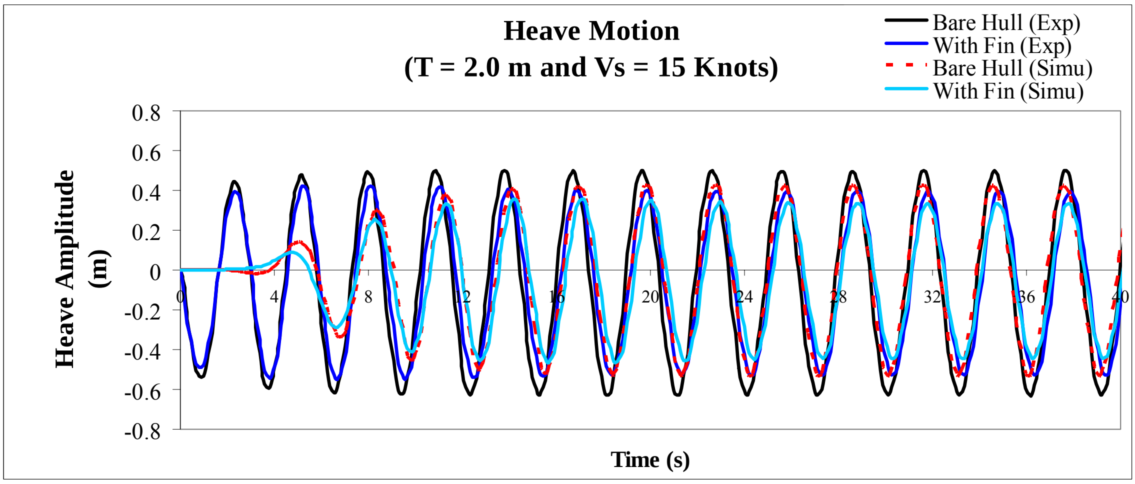


Figure 7.17 Heave motion on the Semi-SWATH vessel wave height = 0.952 m, λ/L_s = 2.0

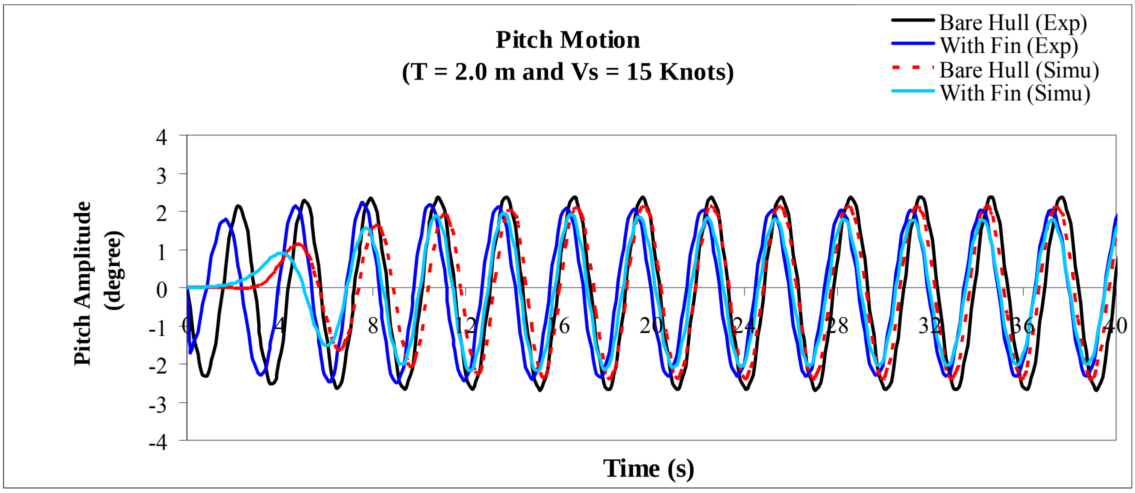


Figure 7.18 Pitch motion on the Semi-SWATH vessel wave height = 0.952 m, λ/L_s = 2.0

Case 1 A.3

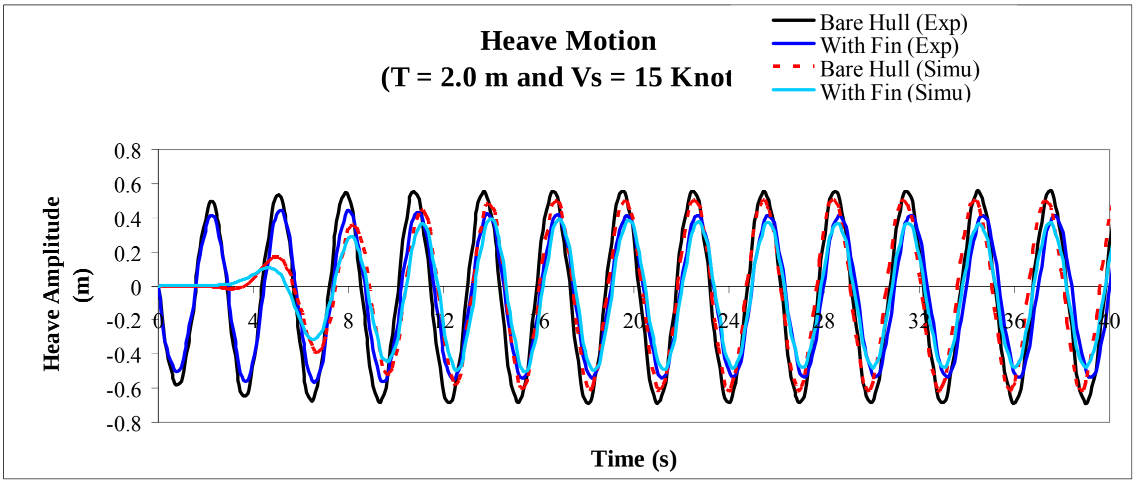


Figure 7.19 Heave motion on the Semi-SWATH vessel wave height = 1.05 m, λ/L_s = 2.2

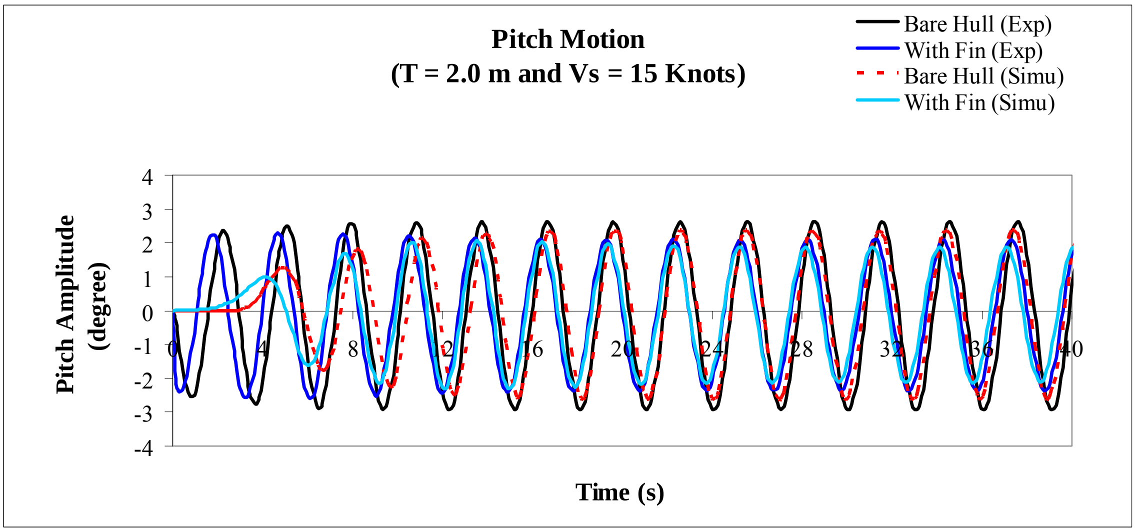


Figure 7.20 Pitch motion on the Semi-SWATH vessel wave height = 1.05 m, λ/L_s = 2.2

Case 1 A.4

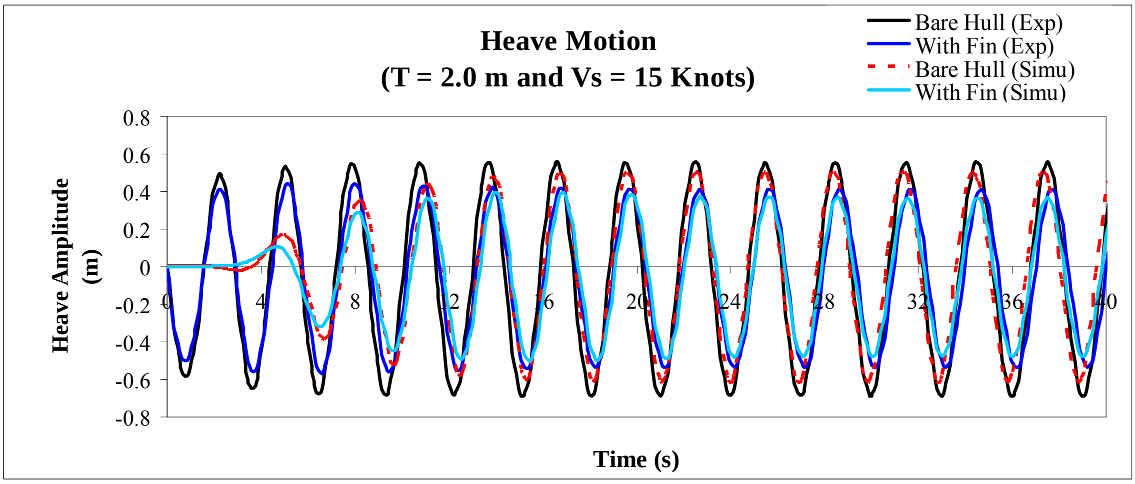


Figure 7.21 Heave motion on the Semi-SWATH vessel wave height = 1.14 m, λ/L_s = 2.4

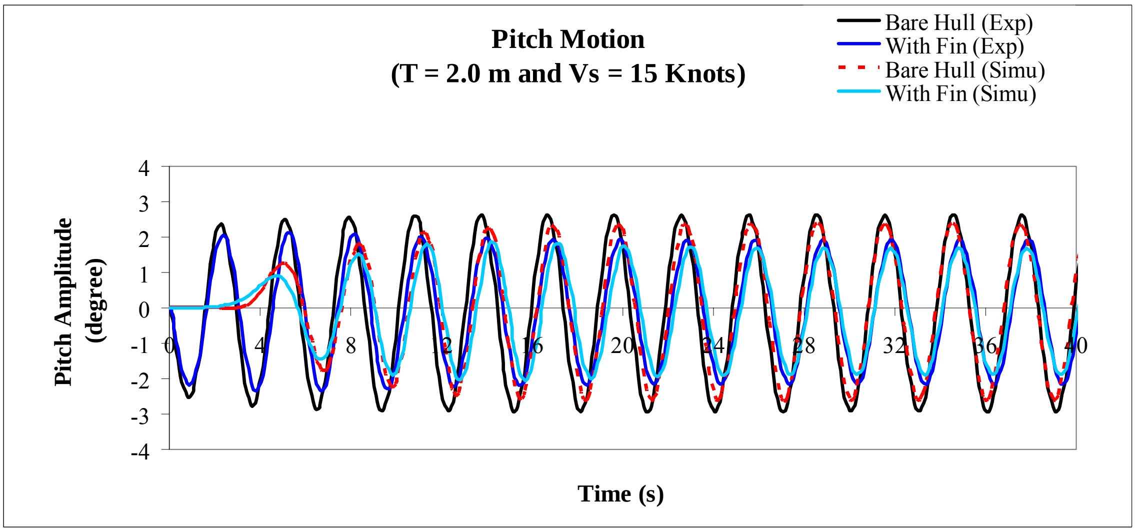


Figure 7.22 Pitch motion on the Semi-SWATH vessel wave height = 1.14 m, λ/L_s = 2.4

Case 1A.5

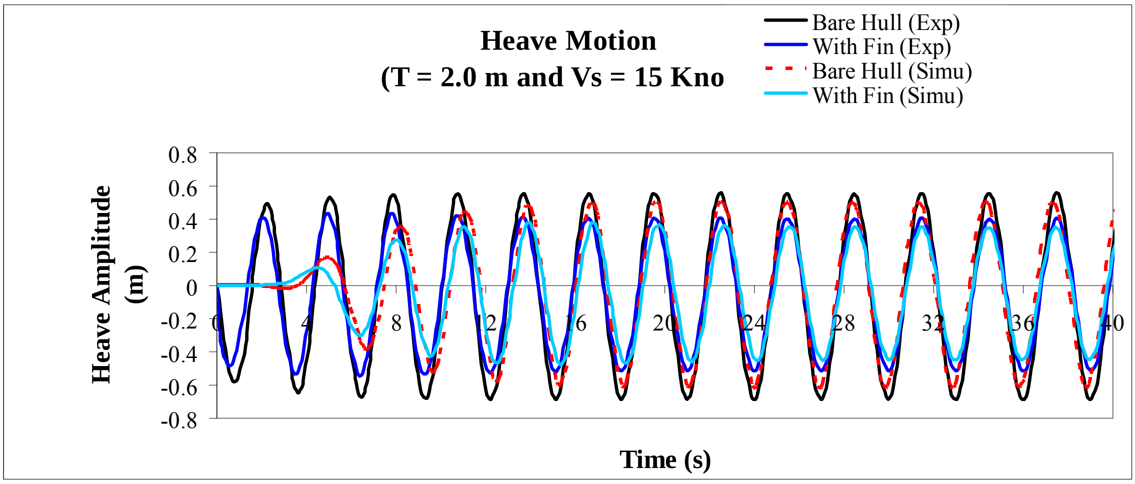


Figure 7.23 Heave motion on the Semi-SWATH vessel wave height = 1.19 m, λ/L_s = 2.6

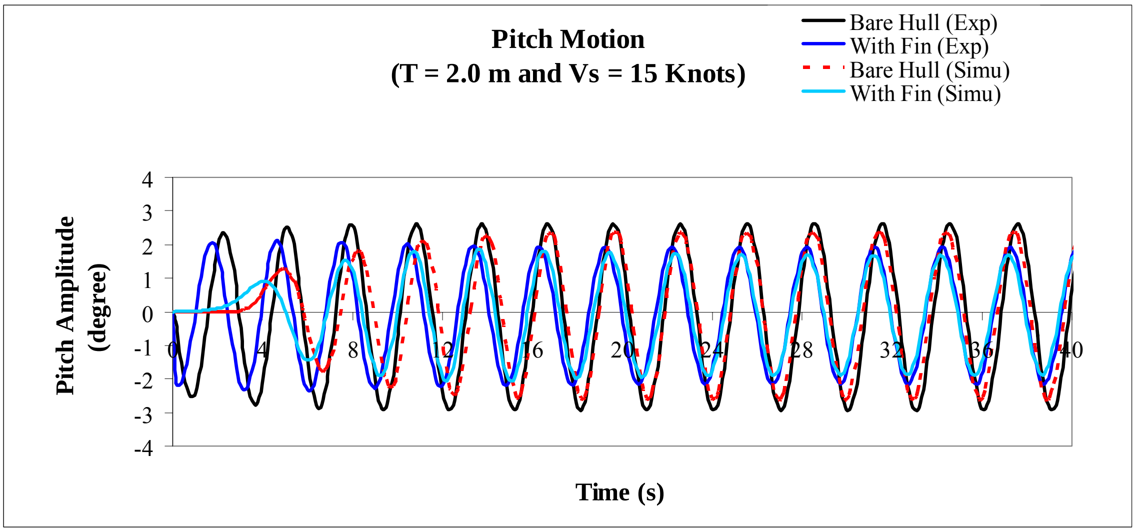


Figure 7.24 Pitch motion on the Semi-SWATH vessel wave height = 1.19 m, λ/L_s = 2.6

Case 1B

Fin Angle: Aft = 10^0 and Fore = 15^0

Table 7.11 Semi-SWATH vessel results obtained with and without using fixed fore and aft fins (experimentally).

Description	Case 1B.1 WH	λ/L_s	Case 1B.2 WH	λ/L_s	Case 1B.3 WH	λ/L_s	Case 1B.4 WH	λ/L_s	Case 1B.5 WH	λ/L_s
	0.857	**1.8**	**0.952**	**2.0**	**1.05**	**2.2**	**1.14**	**2.4**	**1.19**	**2.6**
Heave improvement (%)	34.02		28.144		26.716		23.84		32.585	
Pitch improvement (%)	22.00		11.34		18.499		16.61		19.505	

Case 1B.1

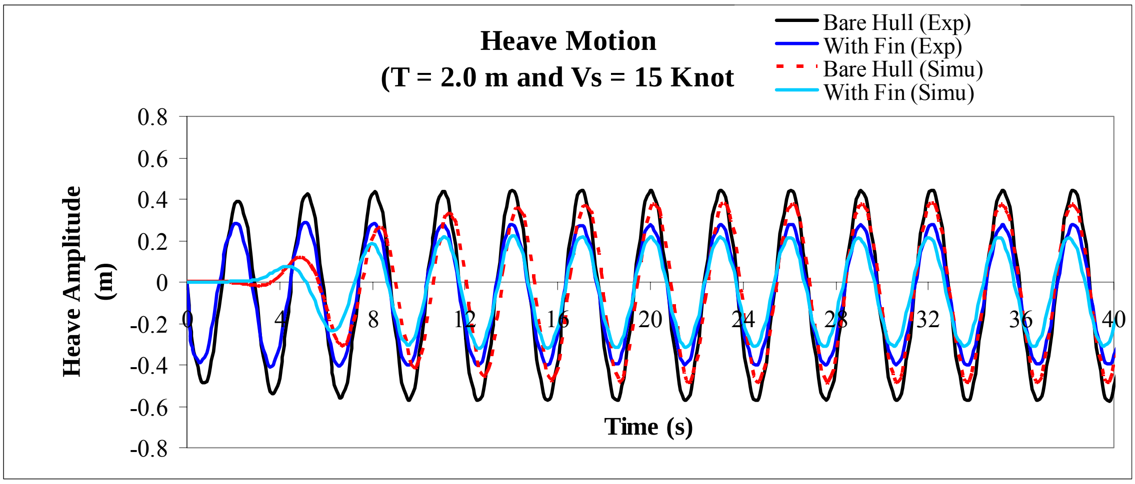


Figure 7.25 Heave motion on the Semi-SWATH vessel wave height = 0.857 m, λ/L_s = 1.8

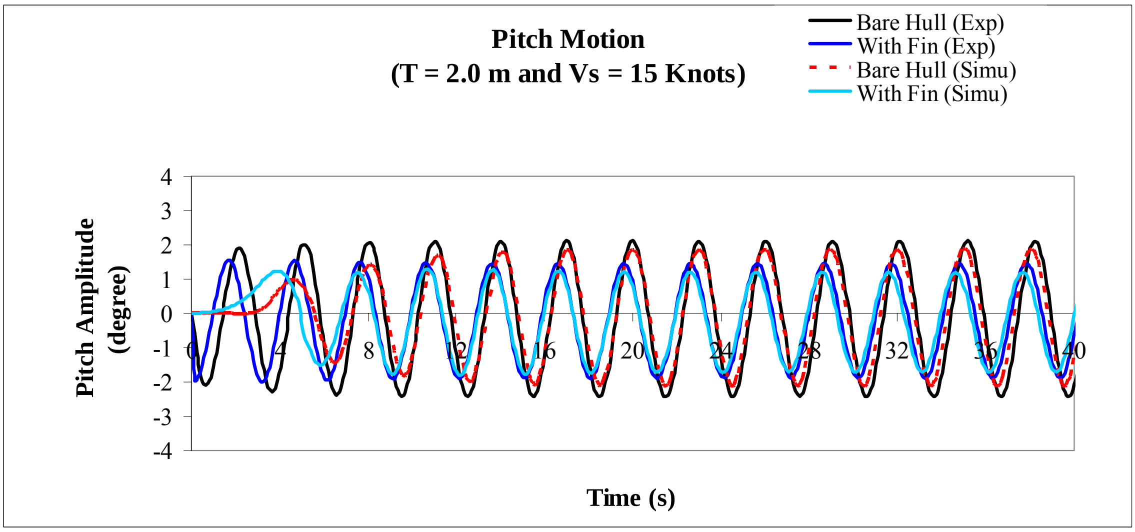


Figure 7.26 Pitch motion on the Semi-SWATH vessel wave height = 0.857 m, λ/L_s = 1.8

Case 1B.2

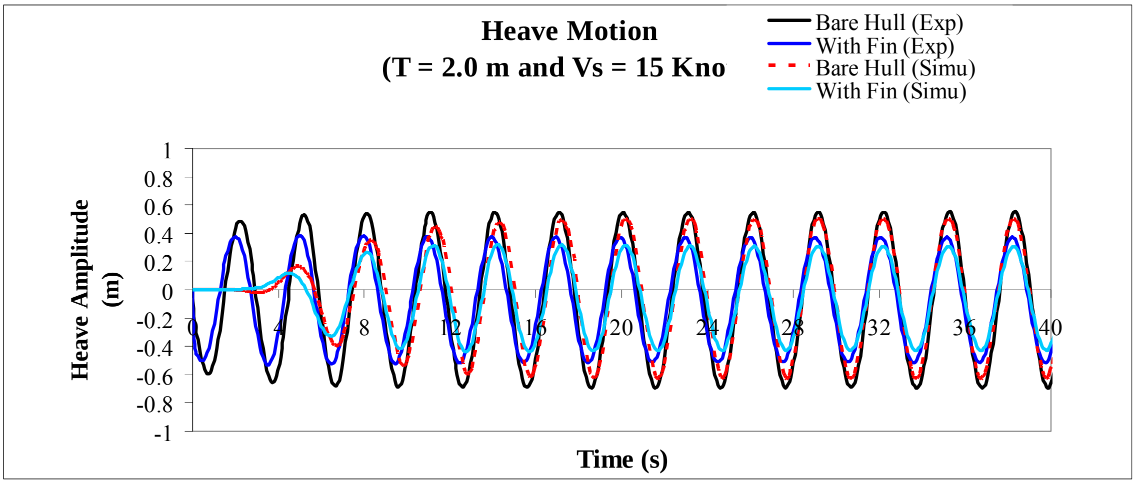


Figure 7.27 Heave motion on the Semi-SWATH vessel wave height = 0.952 m, λ/L_s = 2.0

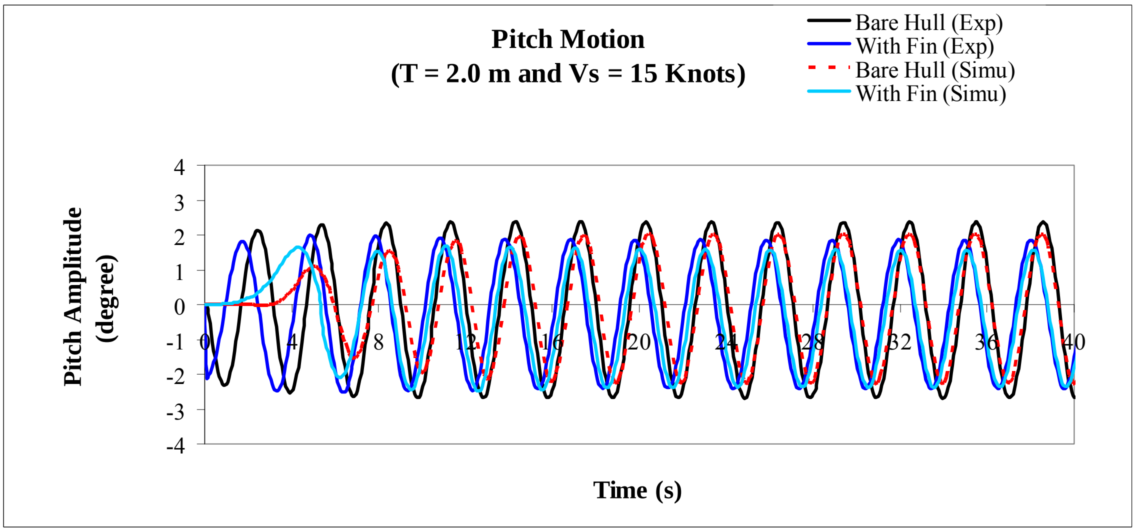


Figure 7.28 Pitch motion on the Semi-SWATH vessel wave height = 0.952 m, λ/L_s = 2.0

Case 1B.3

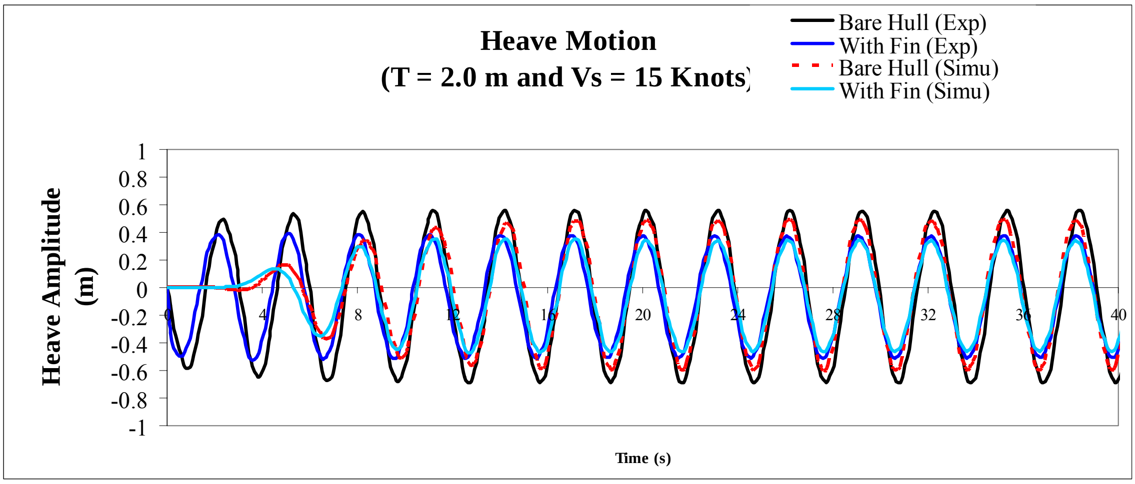


Figure 7.29 Heave motion on the Semi-SWATH vessel wave height = 1.05 m, λ/L_s = 2.2

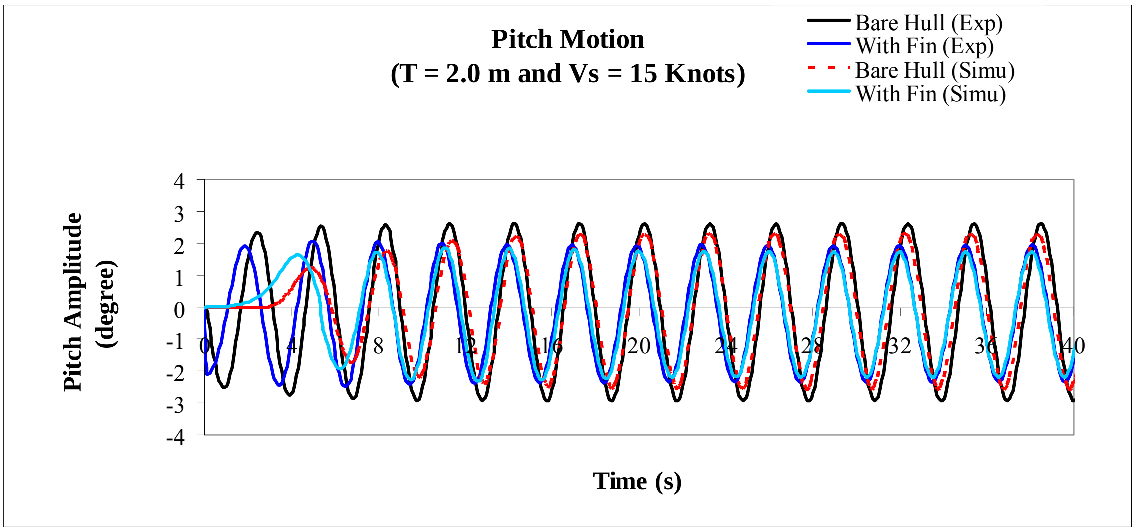


Figure 7.30 Pitch motion on the Semi-SWATH vessel at wave height = 1.05 m, λ/L_s = 2.2

Case 1B.4

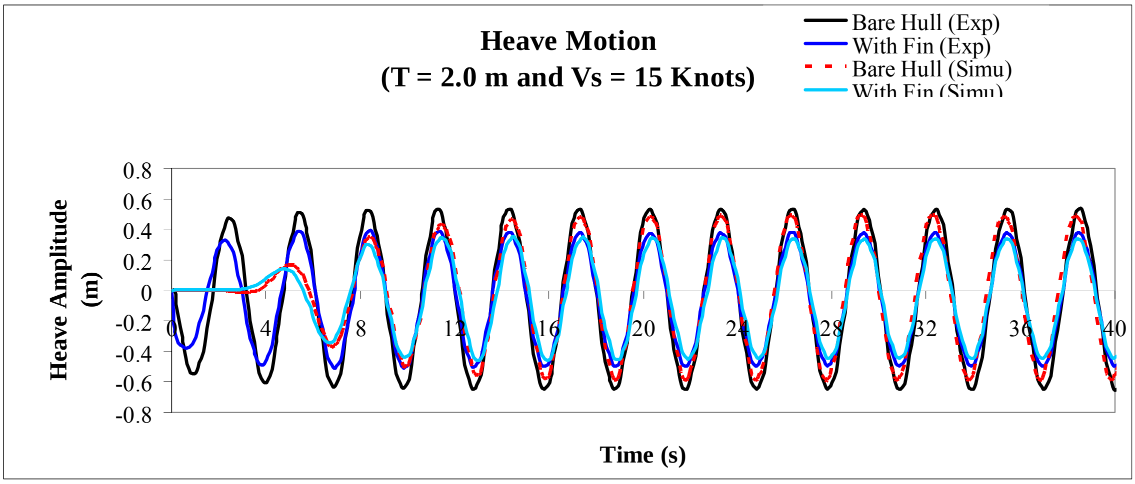


Figure 7.31 Heave motion on the Semi-SWATH vessel at wave height = 1.14 m, λ/L_s = 2.4

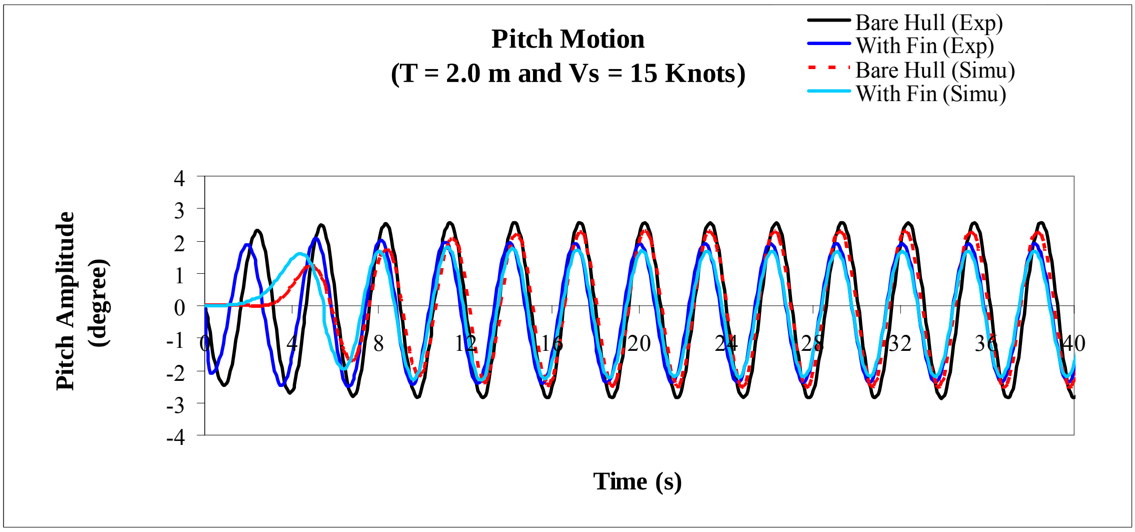


Figure 7.32 Pitch motion on the Semi-SWATH vessel at wave height = 1.14 m, λ/L_s = 2.4

Case 1B.5

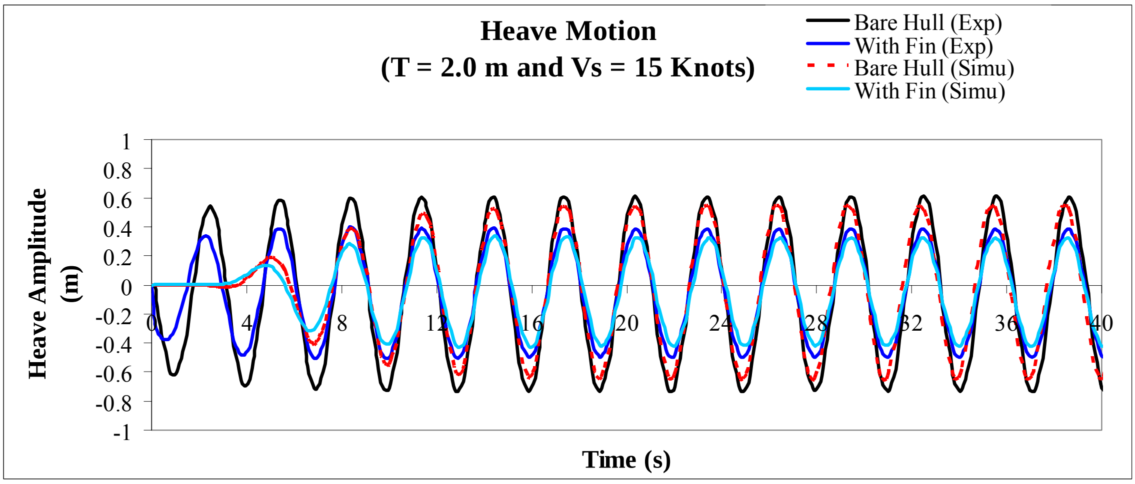


Figure 7.33 Heave motion on the Semi-SWATH vessel at wave height = 1.19 m, λ/L_s = 2.6

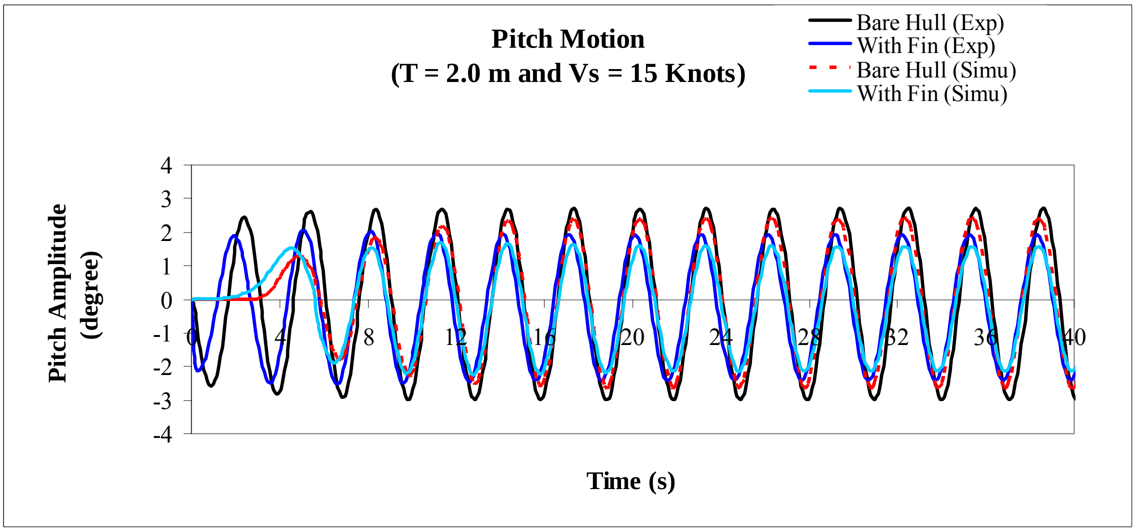


Figure 7.34 Pitch motion on the Semi-SWATH vessel at wave height = 1.19 m, λ/L_s = 2.6

Case 1C

Fin Angle: Aft = 15^0 and Fore = 15^0

Table 7.12 Semi-SWATH vessel results obtained with and without using fixed fore and aft fins (experimentally).

Description	Case 1C.1 WH	Case 1C.1 λ/L_s	Case 1C.2 WH	Case 1C.2 λ/L_s	Case 1C.3 WH	Case 1C.3 λ/L_s	Case 1C.4 WH	Case 1C.4 λ/L_s	Case 1C.5 WH	Case 1C.5 λ/L_s
	0.857	**1.8**	**0.952**	**2.0**	**1.05**	**2.2**	**1.14**	**2.4**	**1.19**	**2.6**
Heave improvement (%)	32.14		25.006		24.17		31.61		33.93	
Pitch improvement (%)	34.92		22.678		21.815		28.33		37.95	

Case 1C.1

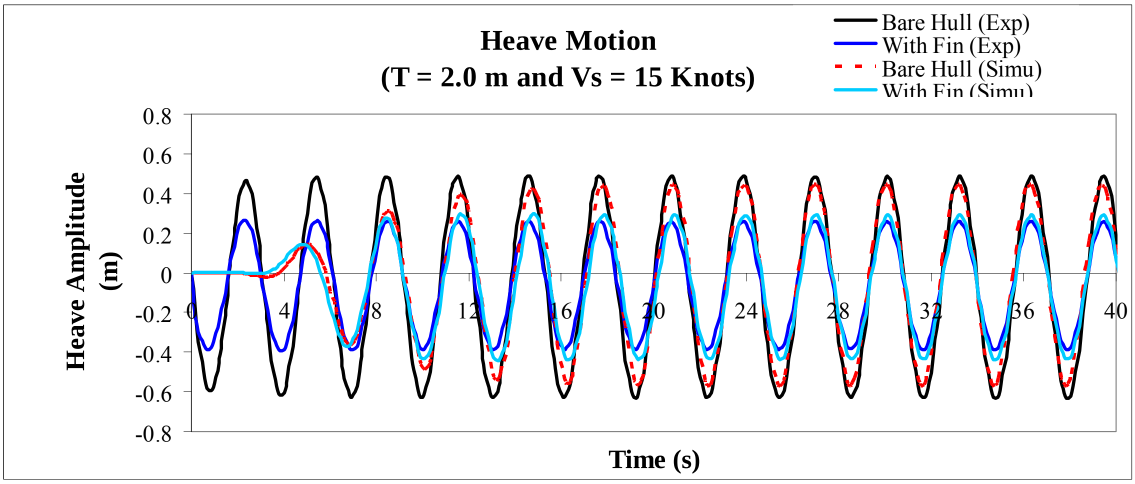


Figure 7.35 Heave motion on the Semi-SWATH vessel at wave height = 0.857 m, $\lambda/L_s = 1.8$

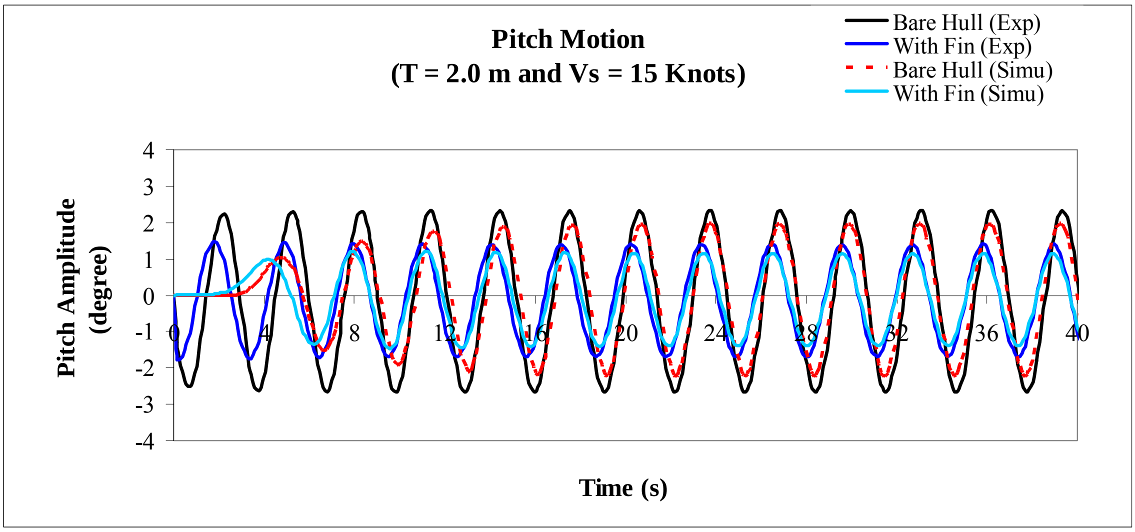


Figure 7.36 Pitch motion on the Semi-SWATH vessel at wave height = 0.857 m, λ/L_s = 1.8

Case 1C.2

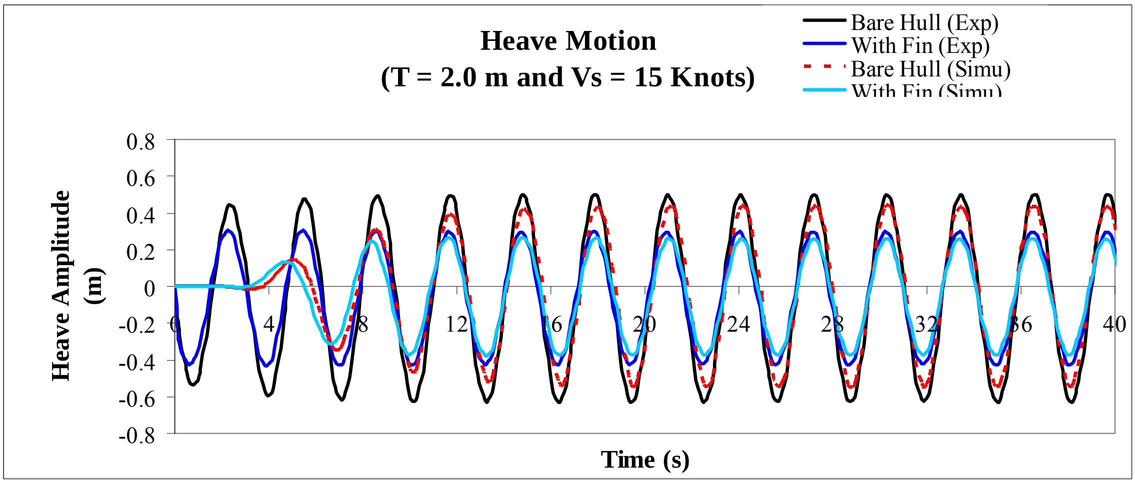


Figure 7.37 Heave motion on the Semi-SWATH vessel at wave height = 0.952 m, $\lambda/L_s = 2.0$

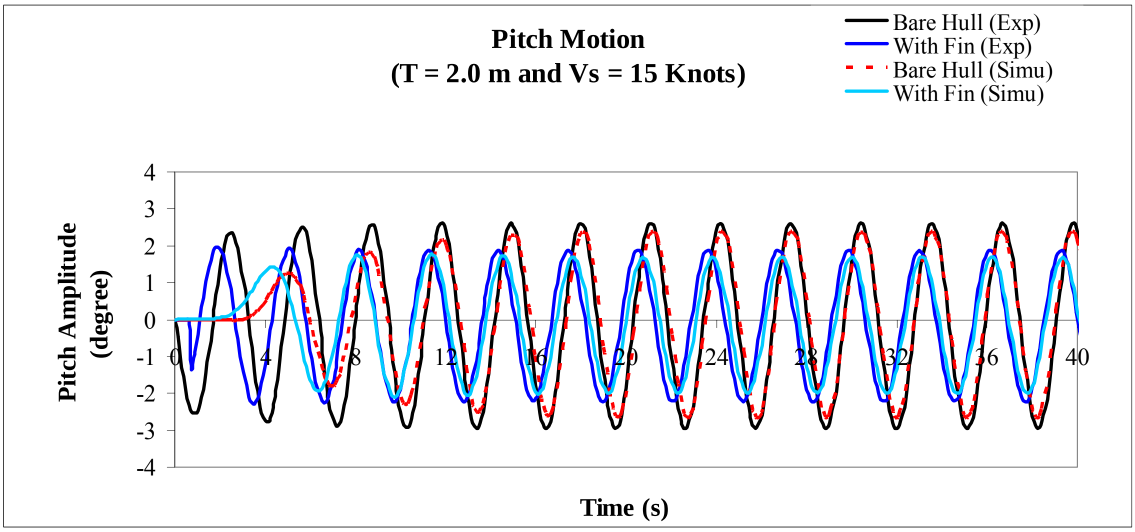


Figure 7.38 Pitch motion on the Semi-SWATH vessel at wave height = 0.952 m, λ/L_s = 2.0

Case 1C.3

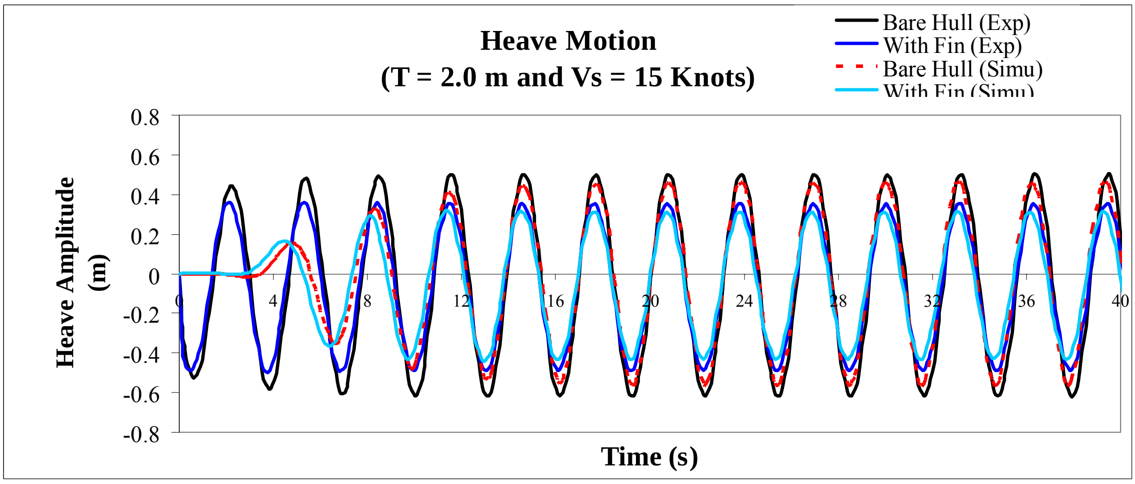


Figure 7.39 Pitch motion on the Semi-SWATH vessel at wave height = 1.05 m, λ/L_s = 2.2

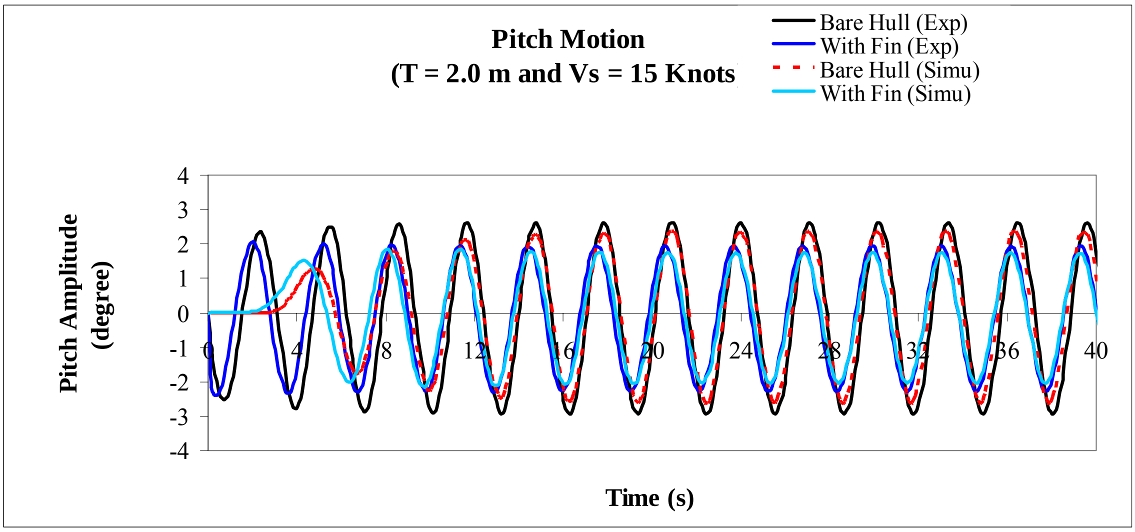


Figure 7.40 Pitch motion on the Semi-SWATH vessel at wave height = 1.05 m, λ/L_s = 2.2

Case 1C.4

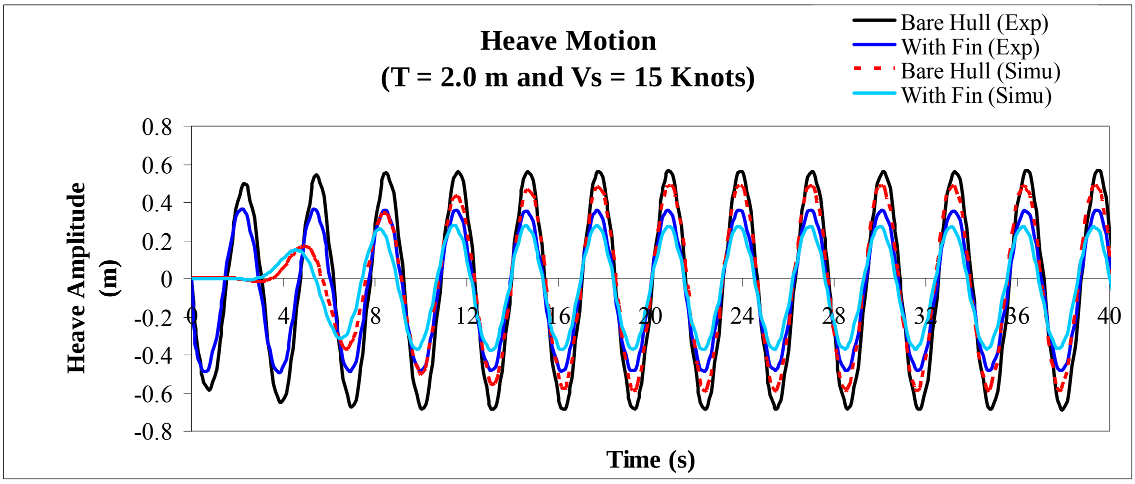


Figure 7.41 Heave motion on the Semi-SWATH vessel at wave height = 1.14 m, λ/L_s = 2.4

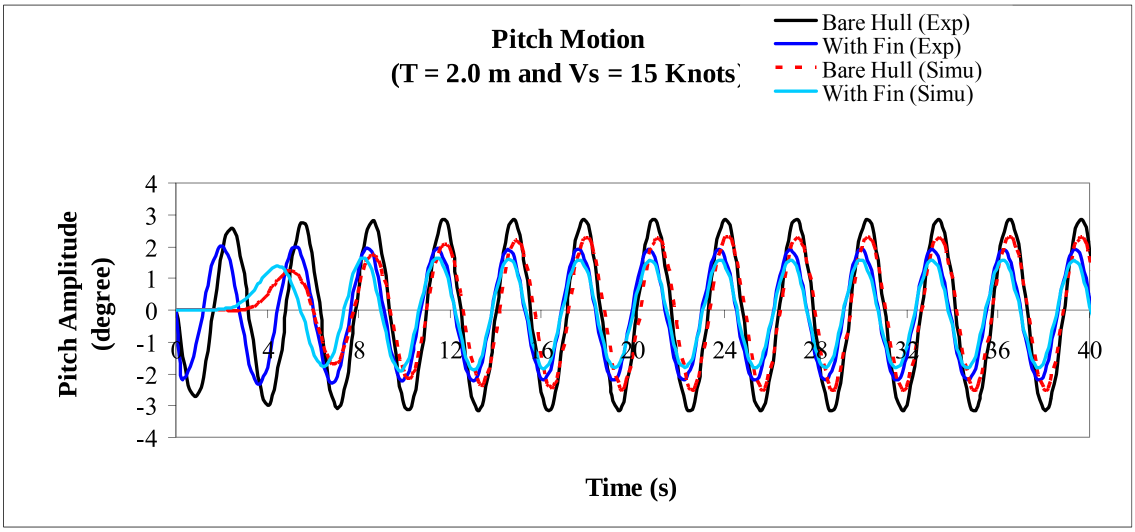


Figure 7.42 Pitch motion on the Semi-SWATH vessel at wave height = 1.14 m, λ/L_s = 2.4

Case 1C.5

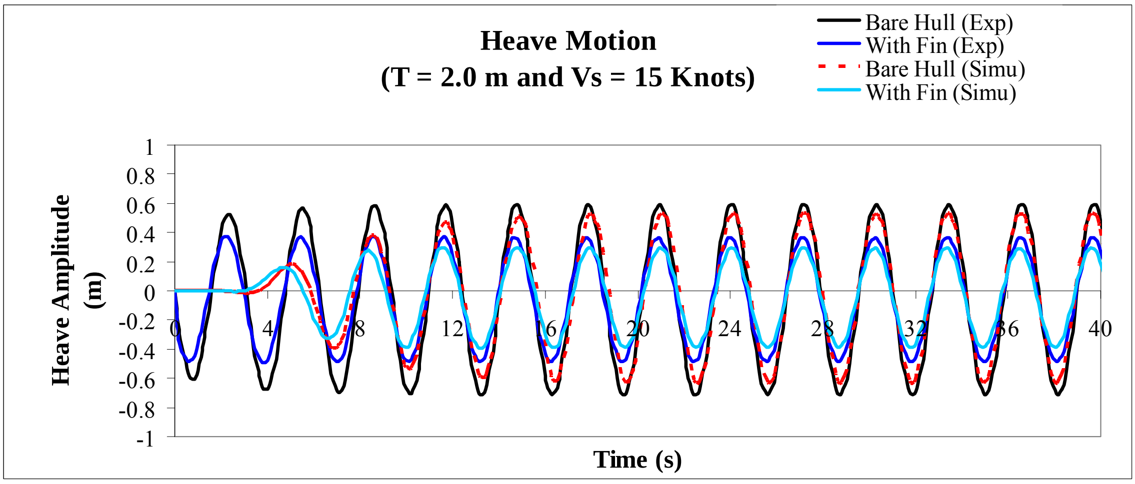


Figure 7.43 Heave motion on the Semi-SWATH vessel at wave height = 1.19 m, λ/L_s = 2.6

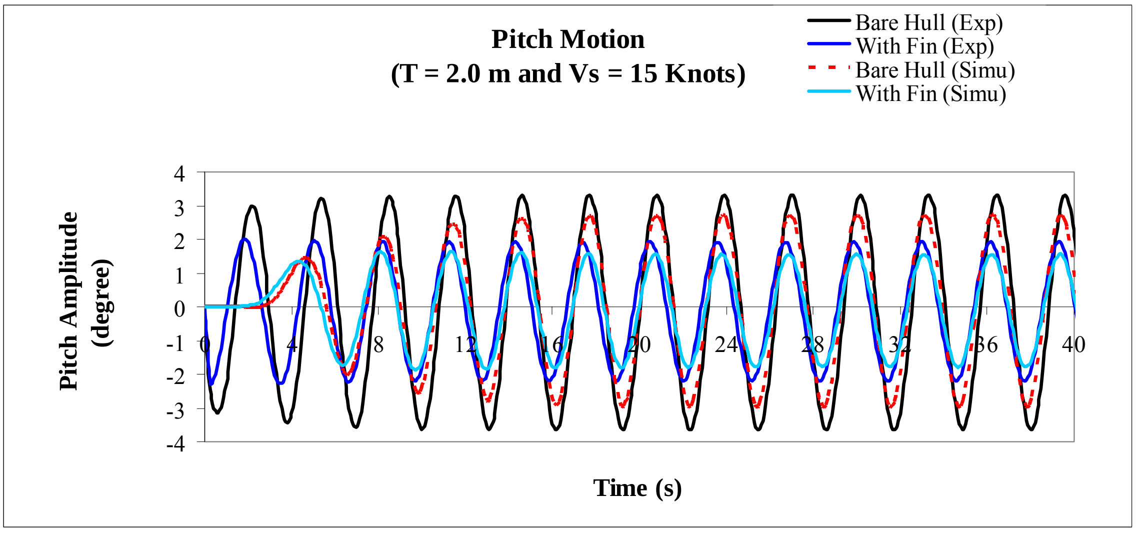


Figure 7.44 Pitch motion on the Semi-SWATH vessel at wave height = 1.19 m, λ/L_s = 2.6

Case 1D

Fin Angle: Aft = 20^0 and Fore = 15^0

Table 7.13 Semi-SWATH vessel results obtained with and without using fixed fore and aft fins (experimentally).

Description	**Case 1D.1**		**Case 1D.2**		**Case 1D.3**		**Case 1D.4**		**Case 1D.5**	
	WH	**λ/L_s**	**WH**	**λ/L_s**	**WH**	**λ/L_s**	**WH**	**λ/L_s**	**WH**	**λ/L_s**
	0.857	**1.8**	**0.952**	**2.0**	**1.05**	**2.2**	**1.14**	**2.4**	**1.19**	**2.6**
Heave improvement (%)	32.44		30.337		36.362		38.85		39.18	
Pitch improvement (%)	35.68		17.124		27.557		36.52		34.72	

Case 1D.1

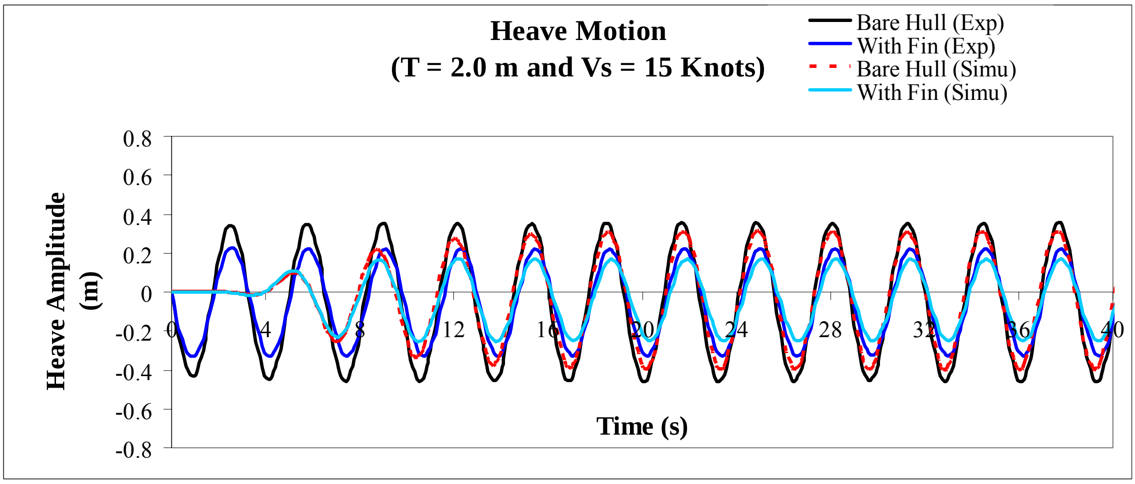


Figure 7.45 Heave motion on the Semi-SWATH vessel wave height = 0.857 m, λ/L_s = 1.8

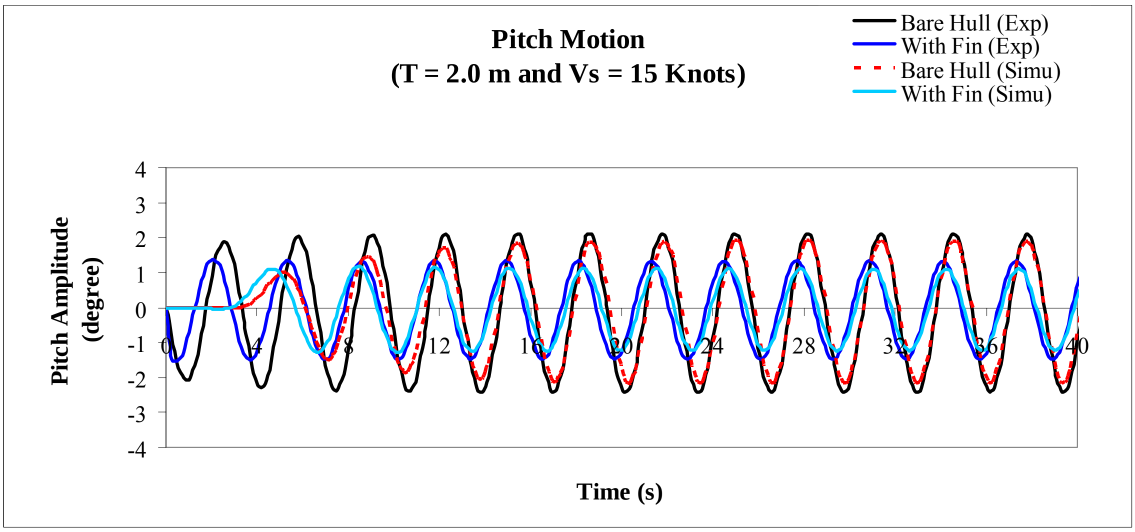


Figure 7.46 Pitch motion on the Semi-SWATH vessel vessel wave height = 0.857 m, λ/L_s = 1.8

Case 1D.2

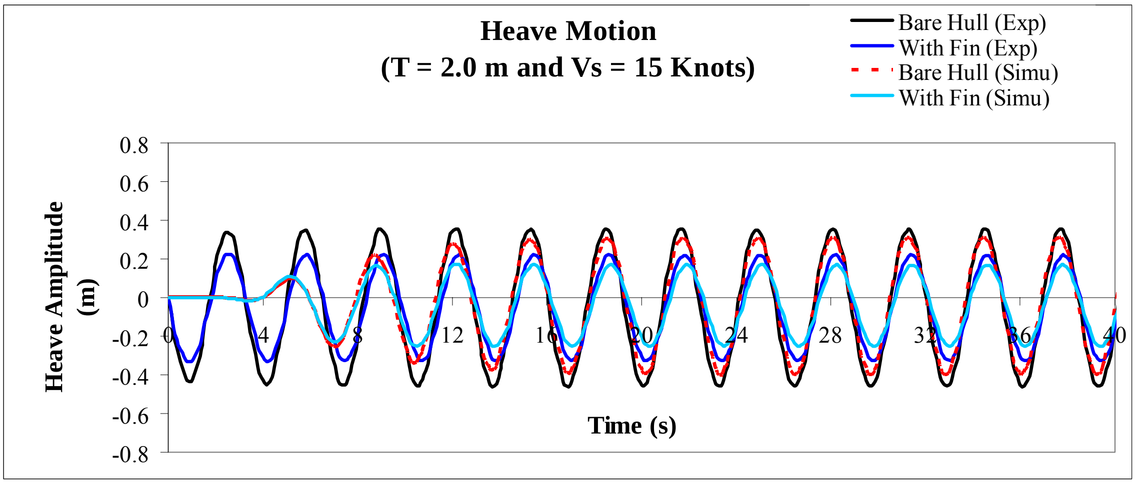


Figure 7.47 Pitch motion on the Semi-SWATH vessel at wave height = 0.952 m, λ/L_s = 2.0

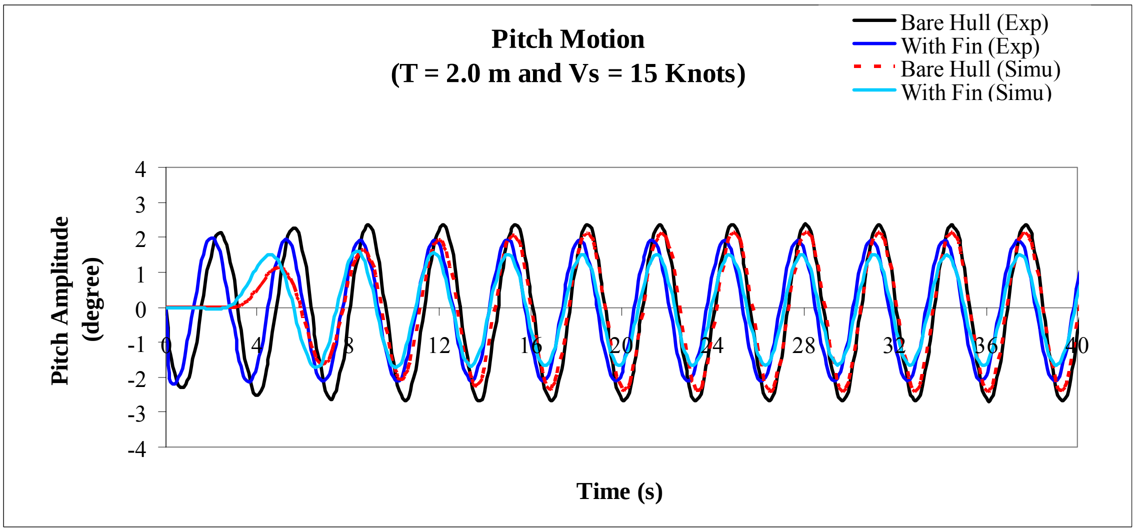


Figure 7.48 Pitch motion on the Semi-SWATH vessel with and without using fixed fore fins = 15^0 and fixed aft fins 20^0

Case 1D.3

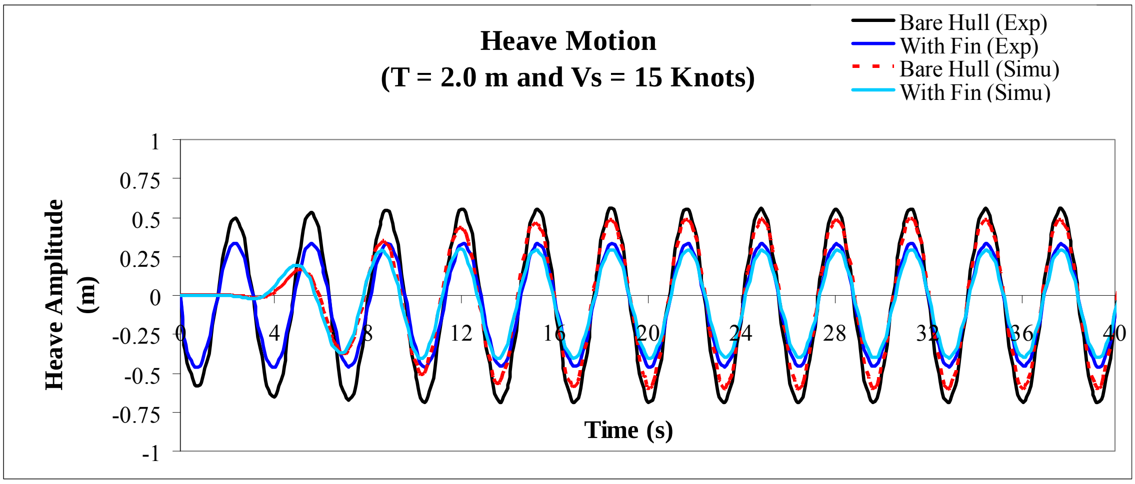


Figure 7.49 Heave motion on the Semi-SWATH vessel at wave height = 1.05 m, λ/L_s = 2.2

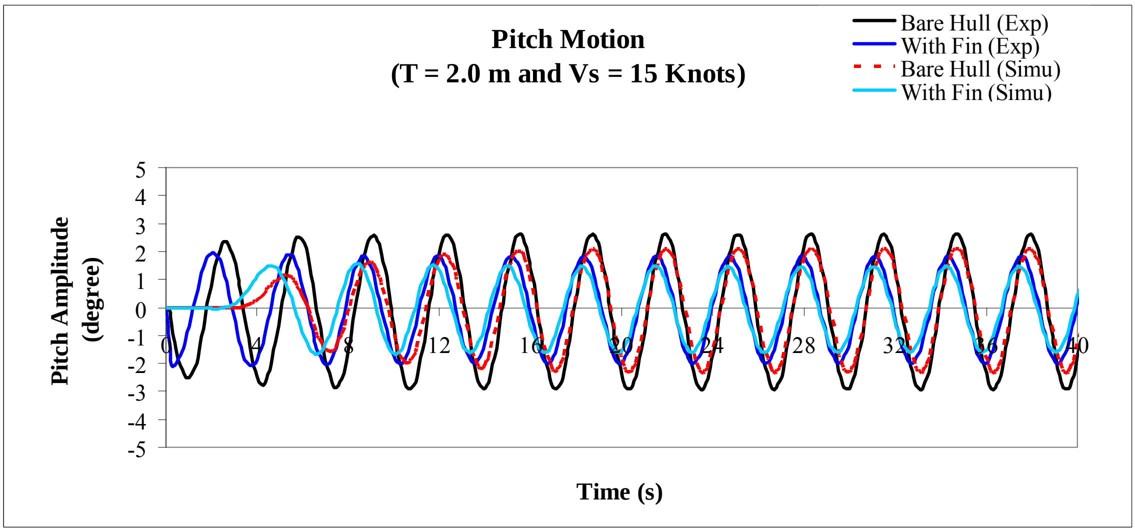


Figure 7.50 Pitch motion on the Semi-SWATH vessel at wave height = 1.05 m, λ/L_s = 2.2

Case 1D.4

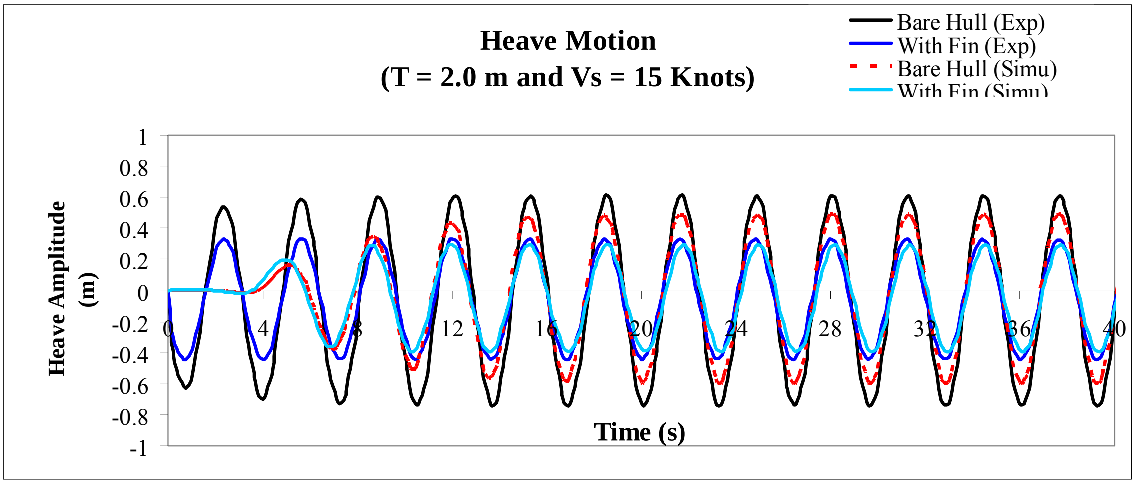


Figure 7.51 Heave motion on the Semi-SWATH vessel at wave height = 1.14 m, λ/L_s = 2.4

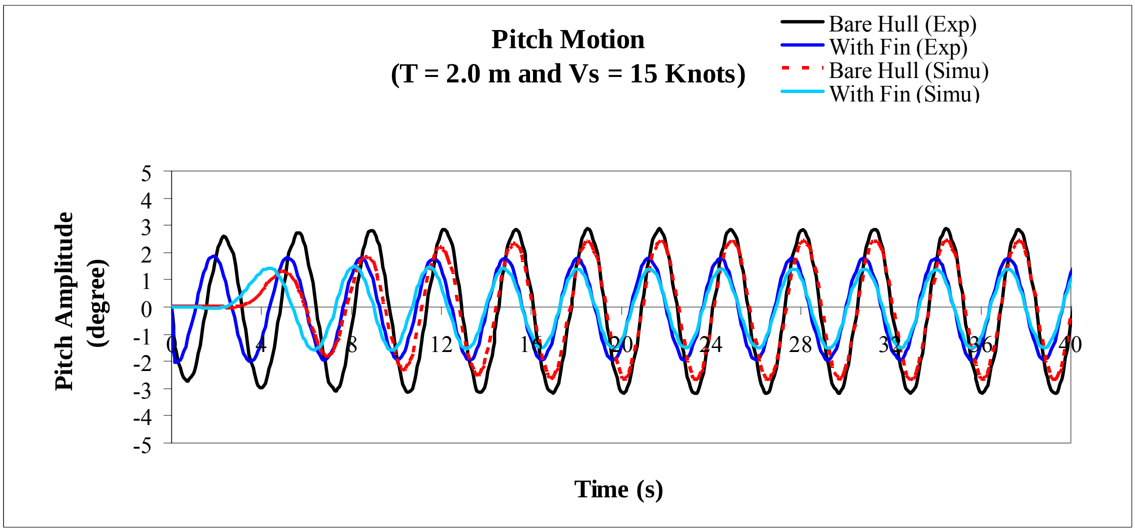


Figure 7.52 Pitch motion on the Semi-SWATH vessel at wave height = 1.14 m, λ/L_s = 2.4

Case 1D.5

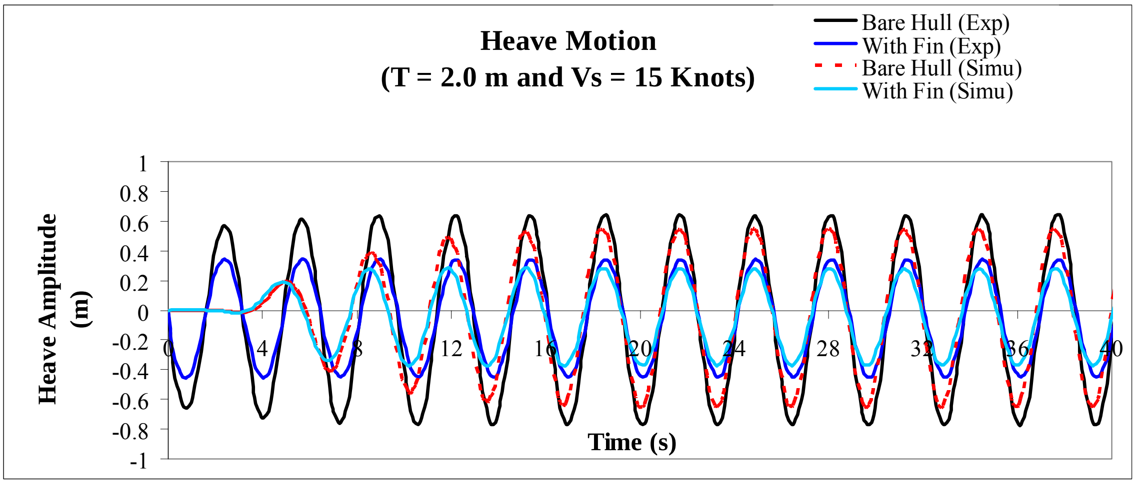


Figure 7.53 Heave motion on the Semi-SWATH vessel at wave height =1.19 m, λ/L_s = 2.6

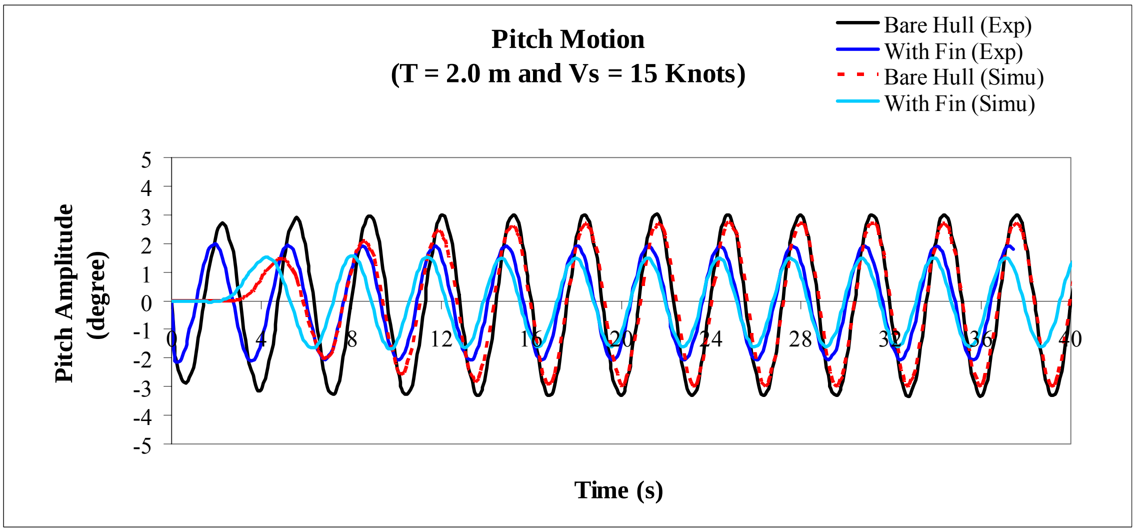


Figure 7.54 Pitch motion on the Semi-SWATH vessel at wave height =1.19 m, λ/L_s = 2.6

Case 1E

Fin Angle: Aft = 25^0 and Fore = 15^0

Table 7.14 Semi-SWATH vessel results obtained with and without using fixed fore and aft fins

Description	Case 1E.1 WH	Case 1E.1 λ/L_s	Case 1E.2 WH	Case 1E.2 λ/L_s	Case 1E.3 WH	Case 1E.3 λ/L_s	Case 1E.4 WH	Case 1E.4 λ/L_s	Case 1E.5 WH	Case 1E.5 λ/L_s
	0.857	**1.8**	**0.952**	**2.0**	**1.05**	**2.2**	**1.14**	**2.4**	**1.19**	**2.6**
Heave improvement (%)	36.26		37.965		24.636		27.97		28.02	
Pitch improvement (%)	35.81		22.944		31.814		26.52		28.784	

Case 1E.1

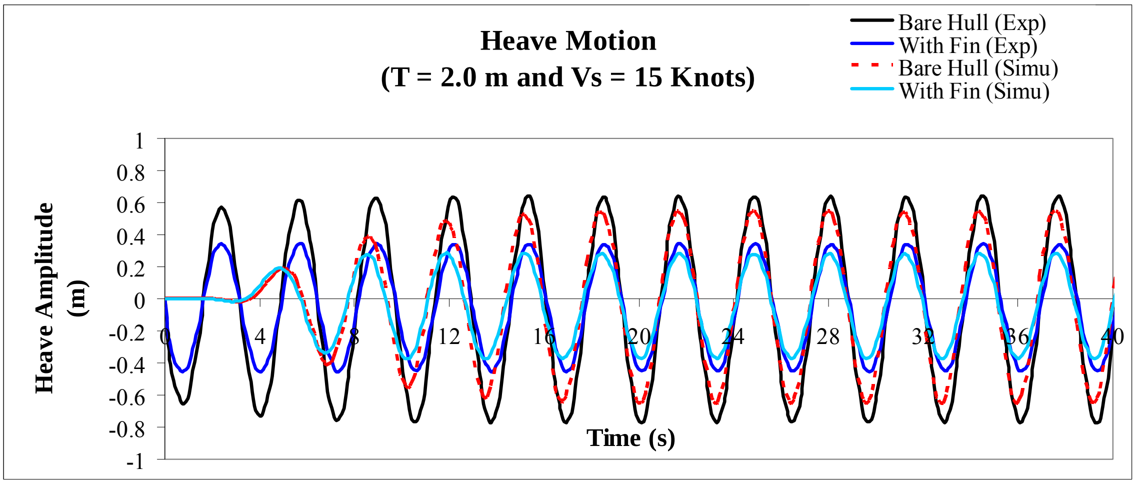


Figure 7.55 Heave motion on the Semi-SWATH vessel at wave height = 0.857 m, λ/L_s = 1.8

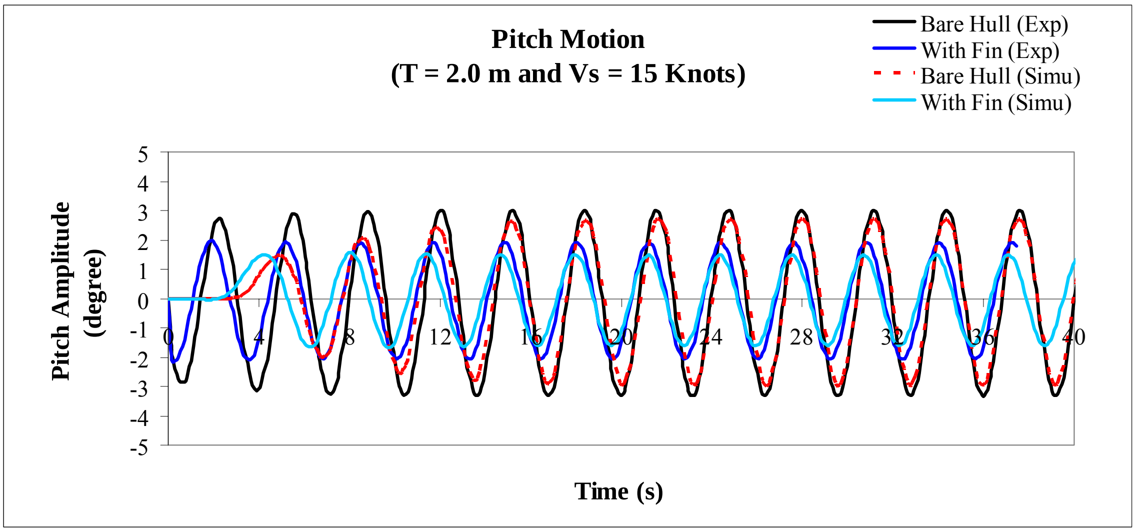


Figure 7.56 Pitch motion on the Semi-SWATH vessel at wave height = 0.857 m, λ/L_s = 1.8

Case 1E.2

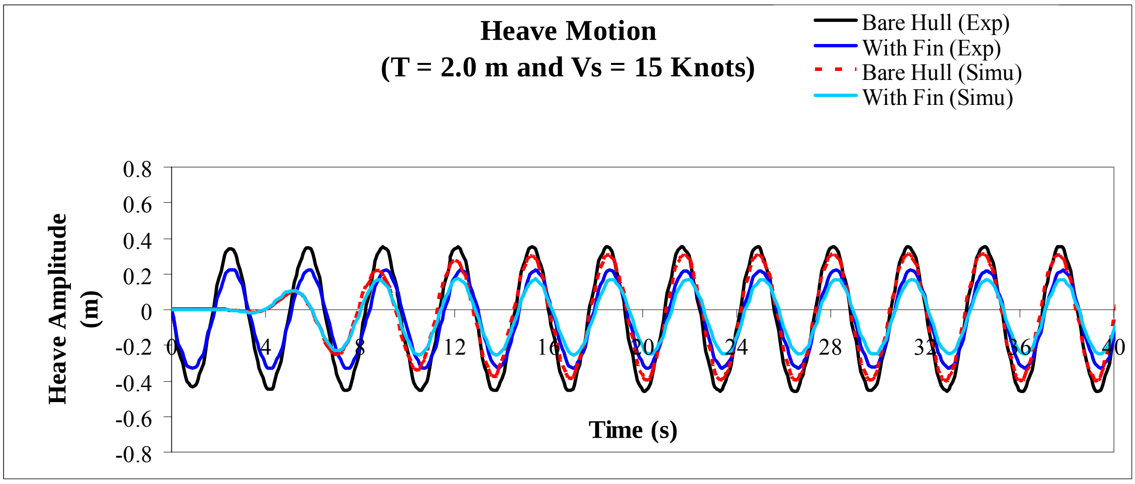


Figure 7.57 Heave motion on the Semi-SWATH vessel at wave height = 0.952 m, $\lambda/L_s = 2.0$

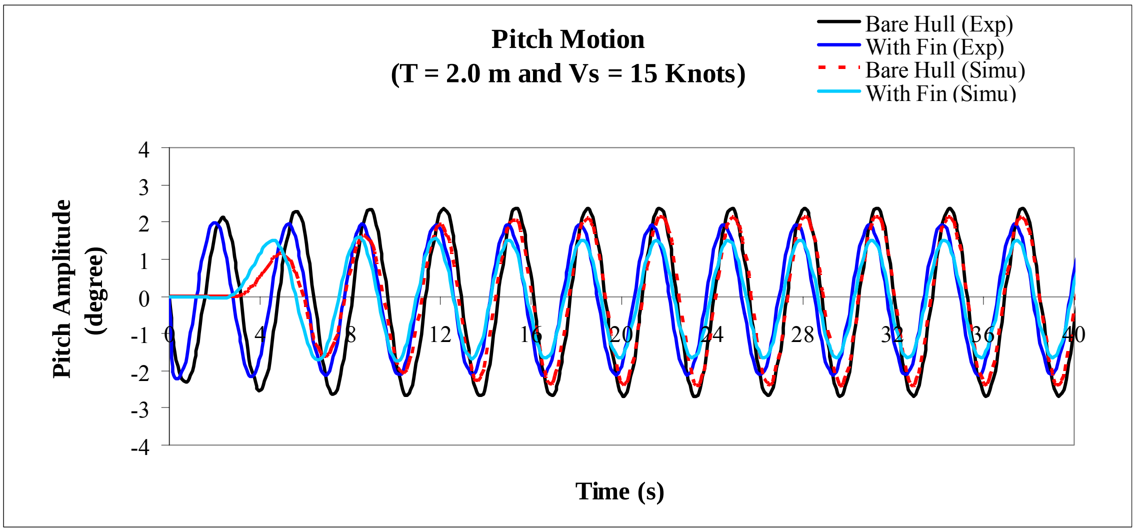


Figure 7.58 Pitch motion on the Semi-SWATH vessel at wave height = 0.952 m, $\lambda/L_s = 2.0$

Case 1E.3

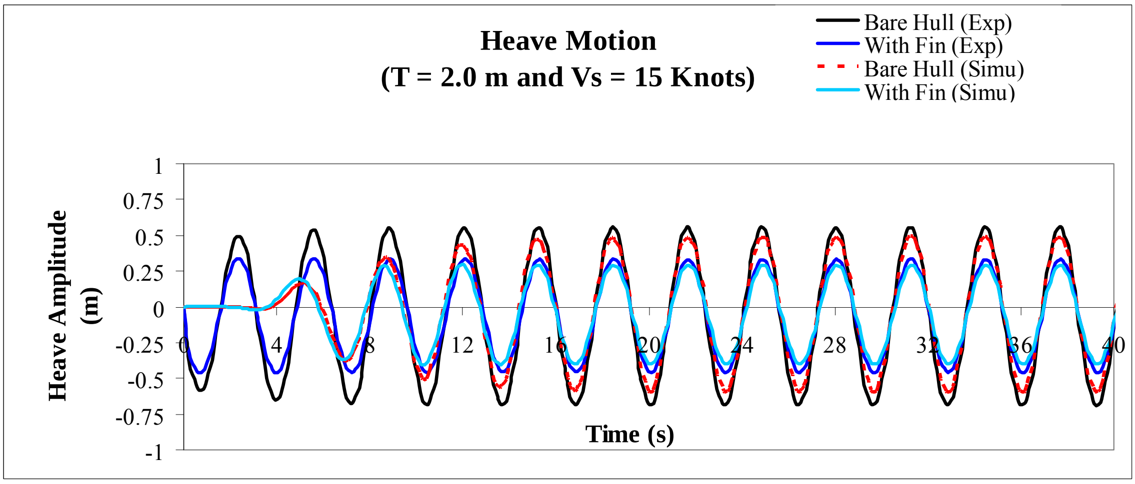


Figure 7.59 Heave motion on the Semi-SWATH vessel at wave height = 1.05 m, $\lambda/L_s = 2.2$

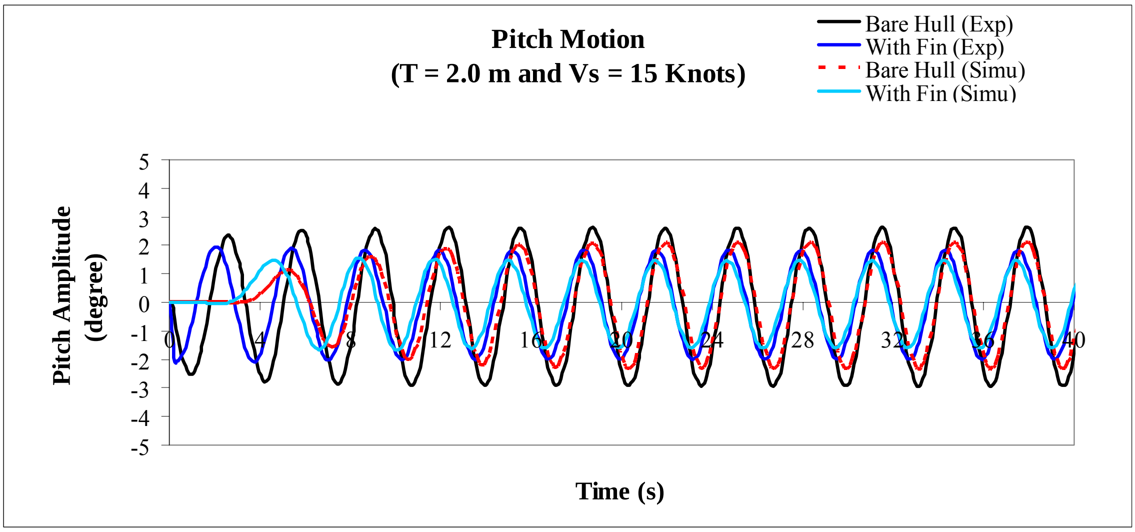


Figure 7.60 Pitch motion on the Semi-SWATH vessel at Wave Height = 1.05 m, λ/L_s = 2.2

Case 1E.4

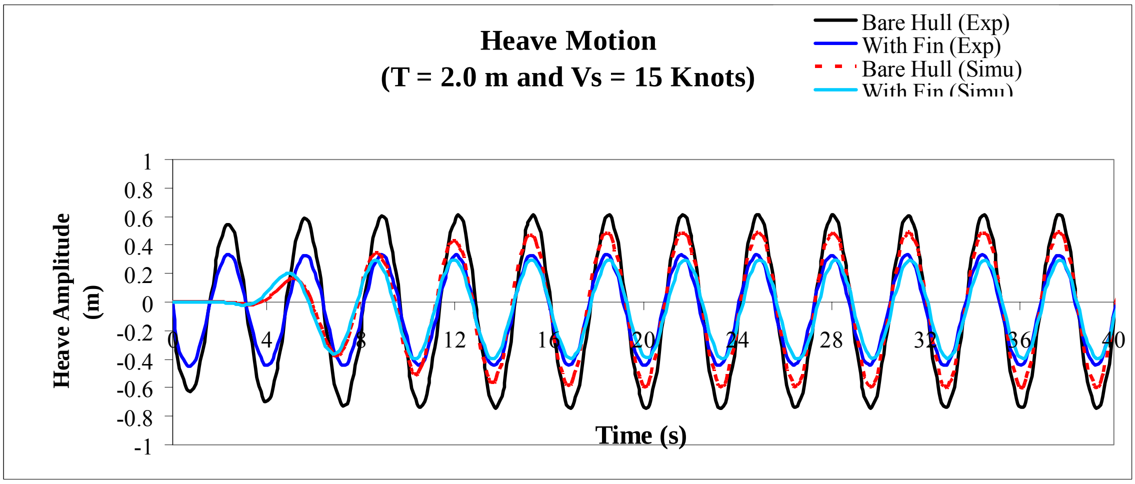


Figure 7.61 Heave motion on the Semi-SWATH vessel at wave height = 1.14 m, λ/L_s = 2.4

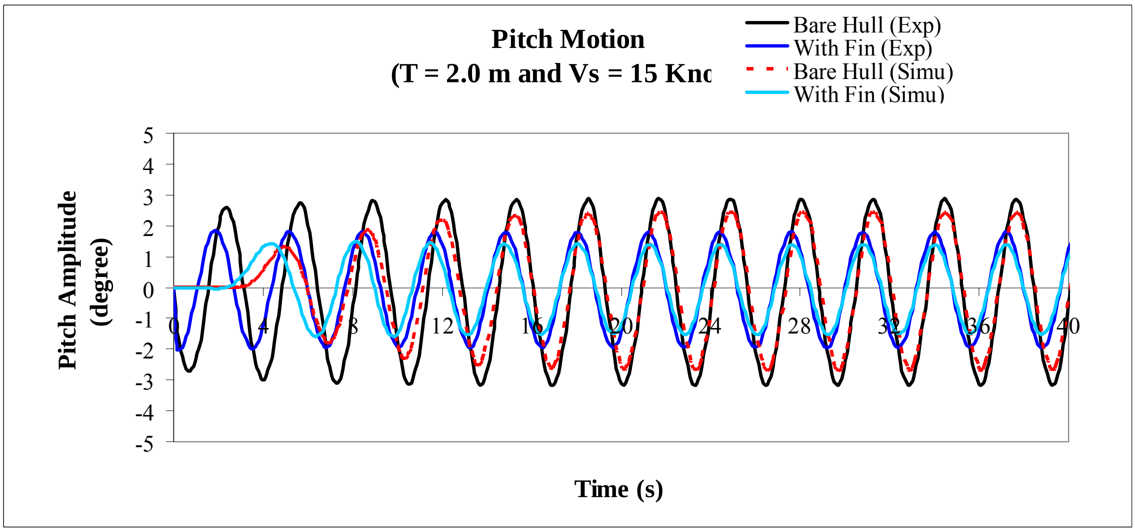


Figure 7.62 Pitch motion on the Semi-SWATH vessel at wave height = 1.14 m, λ/L_s = 2.4

Case 1E.5

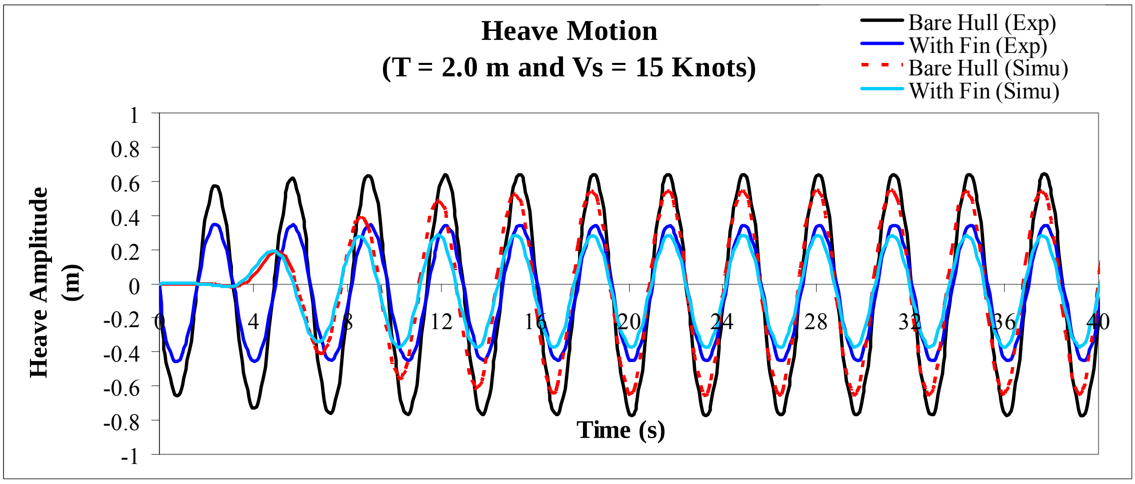


Figure 7.63 Heave motion on the Semi-SWATH vessel at wave height = 1.19 m, λ/L_s = 2.6

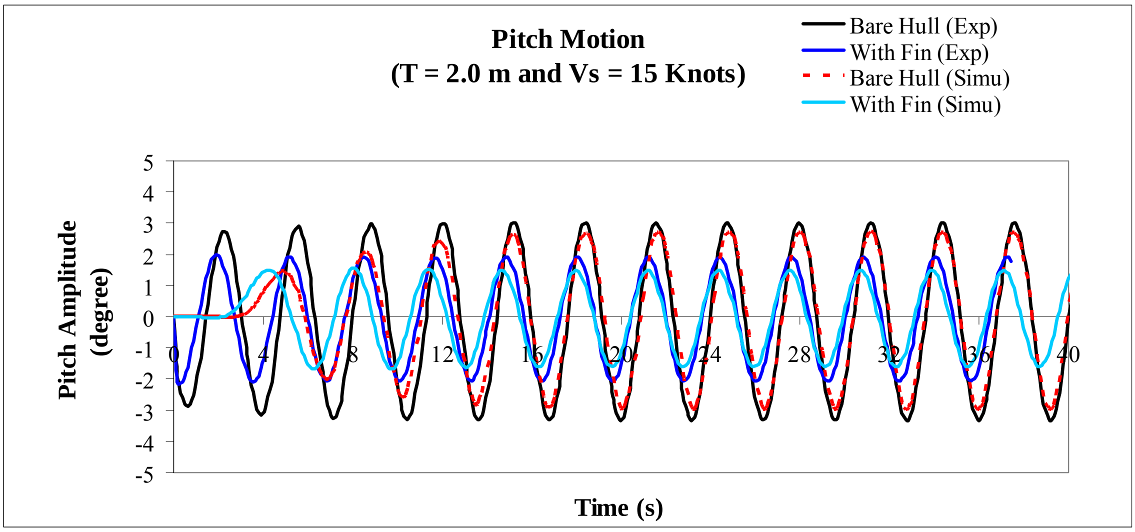


Figure 7.64 Pitch motion on the Semi-SWATH vessel at wave height = 1.19 m, λ/L_s = 2.6

7.4 Case 2

7.4.1 Maximum and Minimum Values of Heave and Pitch Motion at T=1.4 m and Vs=20 Knots

Table 7.15 Summary of heave motion values (experimentally) at various angles of fins

HEAVE MOTION (EXPERIMENT)											
WH - λ/Ls		0.429	1.000	0.476	1.200	0.525	1.400	0.570	1.600	0.595	1.800
Fin Angle (^{0})		Max (m)	Min (m)	Max (m)	Min (m)	Max (m)	Min (m)	Max (m)	Min (m)	Max (m)	Min (m)
Aft	**Fore**										
5	15	0.179	0.157	0.283	0.178	0.313	0.192	0.263	0.178	0.259	0.153
10	15	0.102	0.160	0.284	0.117	0.299	0.145	0.217	0.106	0.197	0.084
15	15	0.190	0.003	0.228	0.085	0.246	0.081	0.181	0.073	0.141	0.078
20	15	0.206	0.071	0.240	0.032	0.211	0.053	0.179	0.029	0.127	0.032
25	15	0.116	0.015	0.210	0.008	0.165	0.051	0.121	0.059	0.075	0.063

Table 7.16 Summary of heave motion values (theoretically) at various angles of fins

HEAVE MOTION (THEORY)											
WH-λ/Ls		0.429	1.000	0.476	1.200	0.525	1.400	0.570	1.600	0.595	1.800
Fin Angle $(^0)$		Max (m)	Min (m)	Max (m)	Min (m)	Max (m)	Min (m)	Max (m)	Min (m)	Max (m)	Min (m)
Aft	Fore										
5	15	0.139	0.130	0.245	0.137	0.289	0.161	0.237	0.138	0.206	0.181
10	15	0.260	0.057	0.264	0.106	0.260	0.108	0.198	0.087	0.224	0.116
15	15	0.225	0.067	0.217	0.071	0.259	0.070	0.168	0.056	0.170	0.065
20	15	0.195	0.081	0.218	0.051	0.204	0.034	0.160	0.008	0.138	0.028
25	15	0.104	0.021	0.197	0.012	0.183	0.025	0.123	0.052	0.064	0.055

Table 7.17 Summary of heave motion values (experimentally) at various angles of fins

PITCH MOTION (EXPERIMENT)											
WH-λ/Ls		0.429	1.000	0.476	1.200	0.525	1.400	0.570	1.600	0.595	1.800
Fin Angle $(^0)$		Max $(^0)$	Min $(^0)$	Max $(^0)$	Min $(^0)$	Max $(^0)$	Min $(^0)$	Max $(^0)$	Min $(^0)$	Max $(^0)$	Min $(^0)$
Aft	**Fore**										
5	15	0.839	1.222	1.234	1.606	1.536	2.127	1.622	2.134	1.456	1.977
10	15	1.149	0.522	1.385	0.940	1.732	1.250	1.797	1.189	1.389	1.107
15	15	1.111	0.168	1.451	0.720	2.158	0.461	1.986	0.377	1.792	0.211
20	15	0.954	0.153	2.029	0.185	2.606	0.373	2.510	0.430	1.574	0.016
25	15	1.094	0.172	1.455	0.085	1.744	0.027	1.228	0.262	0.702	0.440

Table 7.18 Summary of heave motion values (theoretically) at various angles of fins

PITCH MOTION (THEORY)											
WH-λ/Ls		0.429	1.000	0.476	1.200	0.525	1.400	0.570	1.600	0.595	1.800
Fin Angle $(^0)$		Max	Min	Max	Min	Max	Min	Max	Min	Max	Min
Aft	**Fore**	$(^0)$	$(^0)$	$(^0)$	$(^0)$	$(^0)$	$(^0)$	$(^0)$	$(^0)$	$(^0)$	$(^0)$
5	15	0.714	1.250	1.075	1.440	1.359	1.912	1.656	2.176	1.543	1.984
10	15	1.055	0.355	1.437	0.948	1.652	1.085	1.583	1.076	1.170	0.834
15	15	1.016	0.324	1.521	0.886	2.498	0.402	2.120	0.298	1.889	0.380
20	15	0.847	0.086	1.852	0.219	2.533	0.533	2.417	0.376	1.549	0.079
25	15	0.992	0.244	1.389	0.085	1.709	0.056	1.077	0.164	0.633	0.374

7.4.2 Response Amplitude Operators (RAOs) of Heave and Pitch Motion T=1.4 m and Vs=20 Knots

Table 7.19 Summary of Response Amplitude Operators (RAOs) for heave motion (experimentally) at various angles of fins

Fin Angle (^{0})		**λ/Ls**	**HEAVE RAOs (EXPERIMENT)**				
Aft	**Fore**						
5	15	1.00	0.78	0.968	0.962	0.77	0.692
10	15	1.20	0.61	0.843	0.845	0.57	0.471
15	15	1.40	0.45	0.657	0.624	0.44	0.368
20	15	1.60	0.32	0.573	0.503	0.37	0.267
25	15	1.80	0.24	0.459	0.412	0.32	0.231

Table 7.20 Summary of Response Amplitude Operators (RAOs) for heave motion (theoretically) at various angles of fins

Fin Angle (^{0})		**λ/Ls**	**HEAVE RAOs (THEORY) (deg/m)**				
Aft	**Fore**						
5	15	1.00	0.628	0.801	0.856	0.657	0.529
10	15	1.20	0.472	0.779	0.702	0.499	0.468
15	15	1.40	0.369	0.604	0.627	0.392	0.394
20	15	1.60	0.265	0.565	0.453	0.295	0.279
25	15	1.80	0.194	0.439	0.396	0.306	0.201

Table 7.21 Summary of Response Amplitude Operators (RAOs) for pitch motion (experimentally) at various angles of fins

Fin Angle (^{0})		**λ/Ls**	**PITCH RAOs (EXPERIMENT) (deg/m)**				
Aft	**Fore**						
5	15	1.00	4.810	5.967	6.977	6.591	5.770
10	15	1.20	3.898	4.886	5.680	5.240	4.195
15	15	1.40	2.983	4.562	4.988	4.145	3.365
20	15	1.60	2.582	3.874	4.252	3.649	2.618
25	15	1.80	2.153	2.878	3.373	2.614	1.920

Table 7.22 Summary of Response Amplitude Operators (RAOs) for pitch motion (theoretically) at various angles of fins

Fin Angle (^{0})		λ/Ls	PITCH RAOs (THEORY) (deg/m)				
Aft	Fore						
5	15	1.00	4.583	5.340	6.231	6.030	5.040
10	15	1.20	3.290	5.011	5.214	4.665	3.369
15	15	1.40	3.128	4.630	4.830	4.241	3.100
20	15	1.60	2.176	3.432	3.950	3.581	2.470
25	15	1.80	1.746	2.739	3.149	2.178	1.693

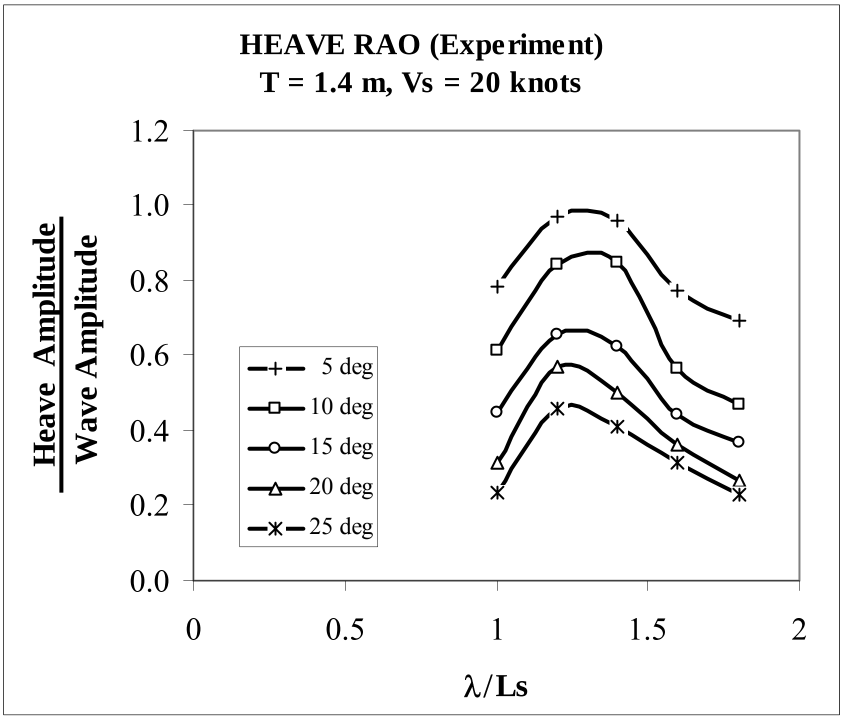


Figure 7.65 RAOs of heave for Semi-SWATH vessel with various angles of fins (experimentally)

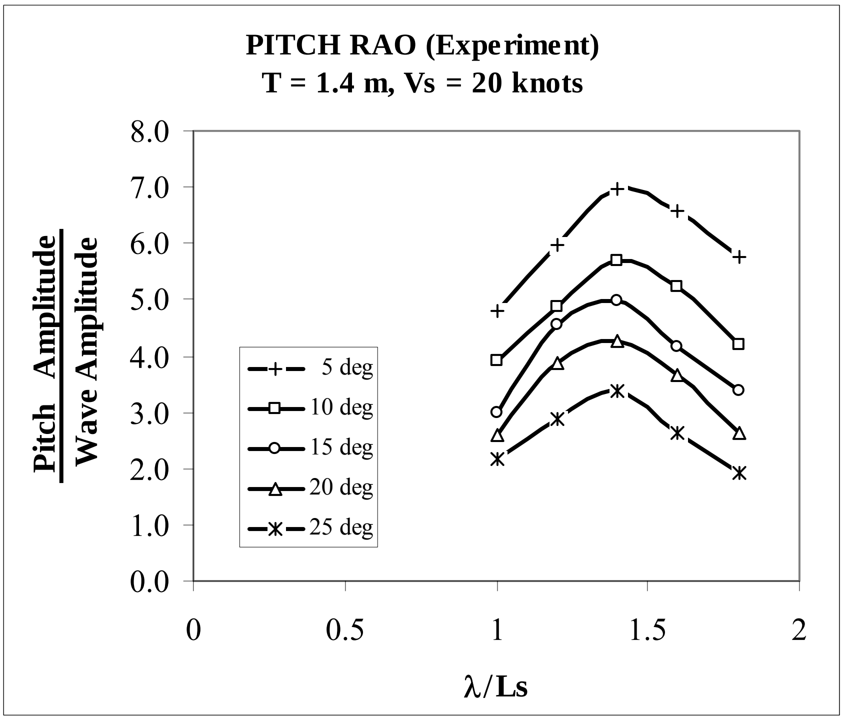


Figure 7.66 RAOs of pitch for Semi-SWATH vessel with various angles of fins (experimentally)

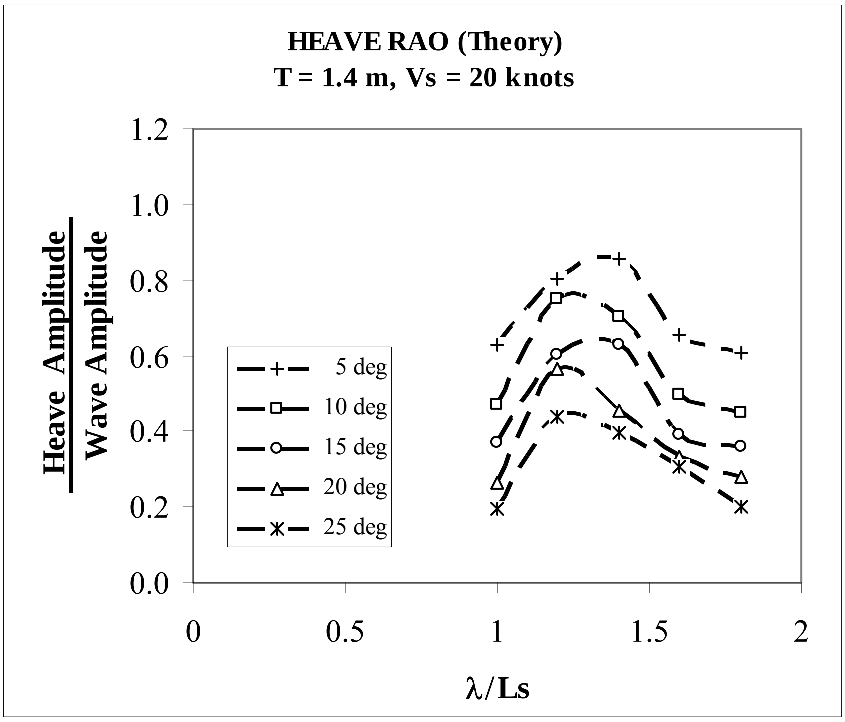


Figure 7.67 RAOs of heave for Semi-SWATH vessel with various angles of fins (theoretically)

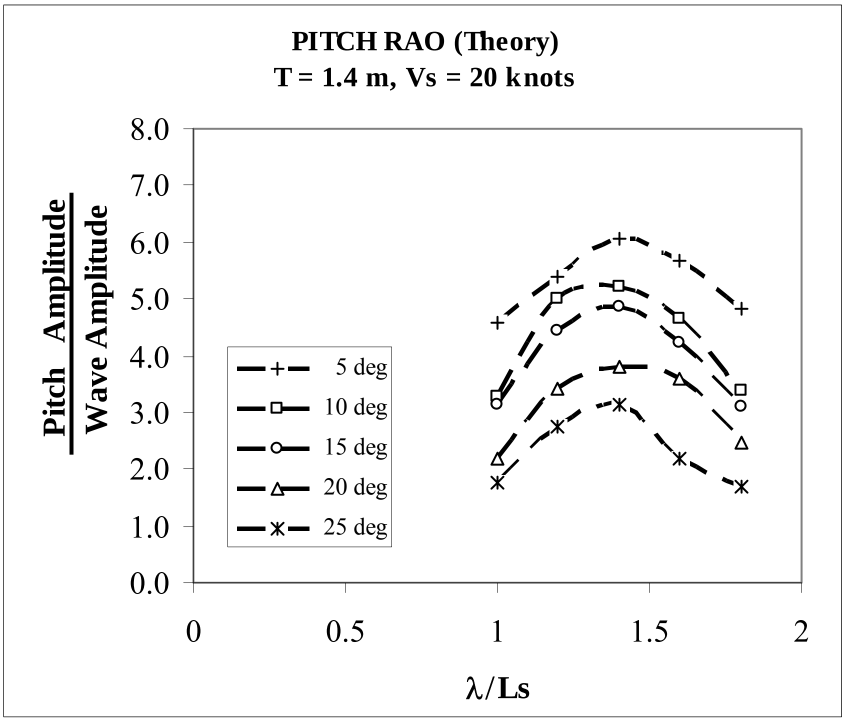


Figure 7.68 RAOs of pitch for Semi-SWATH vessel with various angles of fins (theoretically)

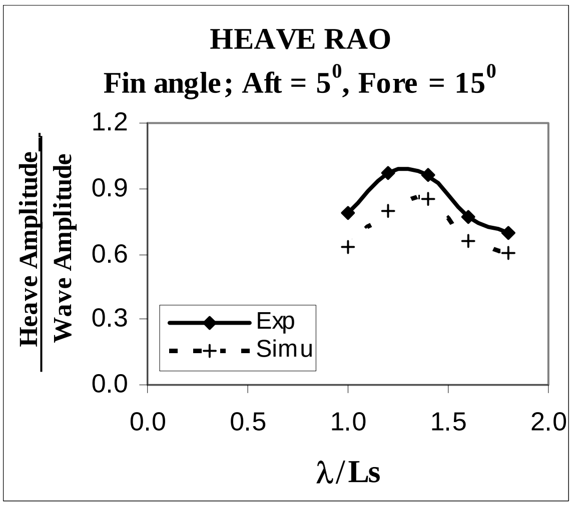


Figure 7.69 RAOs comparison of heave at fin angle; aft = 5^0 and fore = 15^0

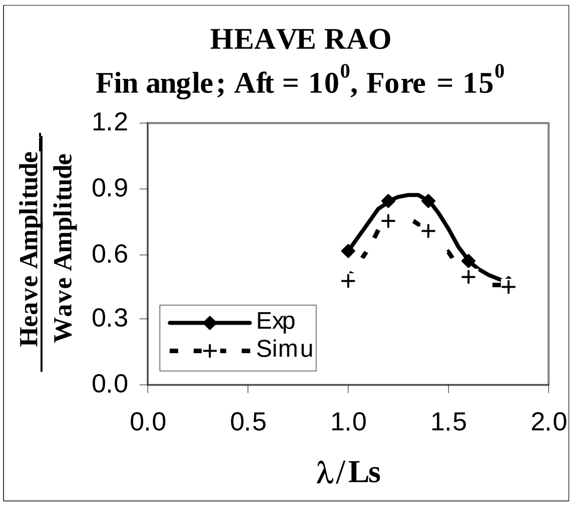


Figure 7.70 RAOs comparison of heave at fin angle; aft = 10^0 and fore = 15^0

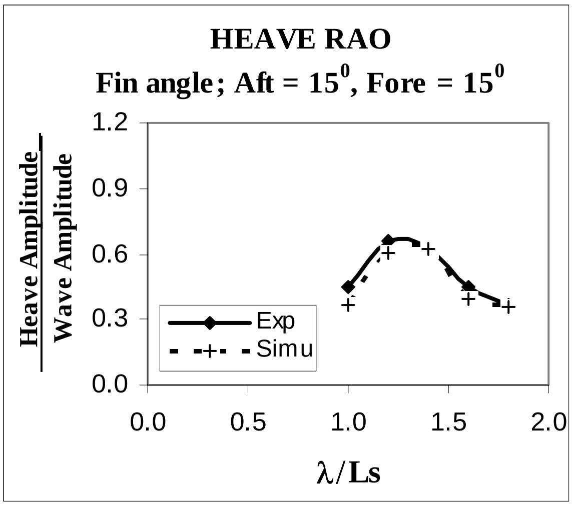


Figure 7.71 RAOs comparison of heave at fin angle; aft = 15^0 and fore = 15^0

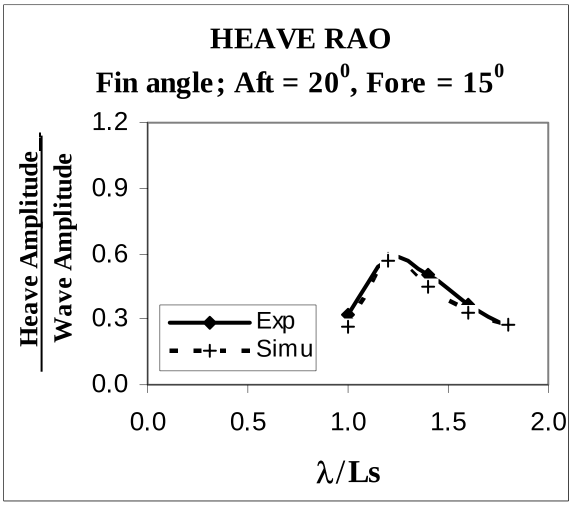


Figure 7.72 RAOs comparison of heave at fin angle; aft = 20^0 and fore = 15^0

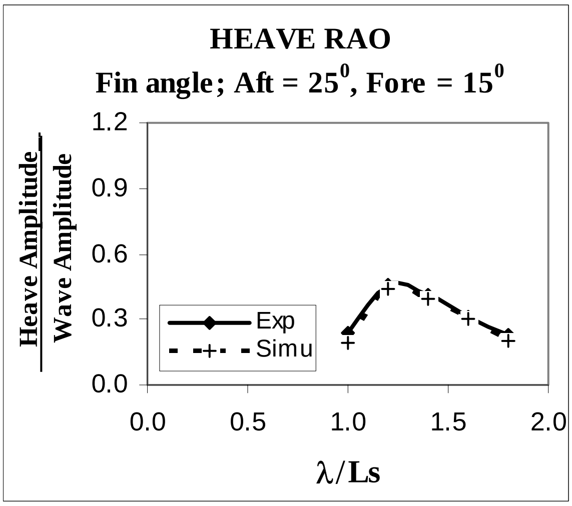


Figure 7.73 RAOs comparison of heave at fin angle; aft = 25^0 and fore = 15^0

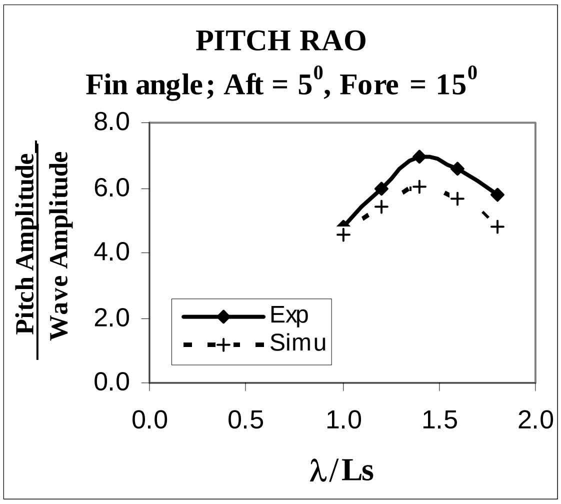


Figure 7.74 RAOs comparison of pitch at fin angle; aft = 5^0 and fore = 15^0

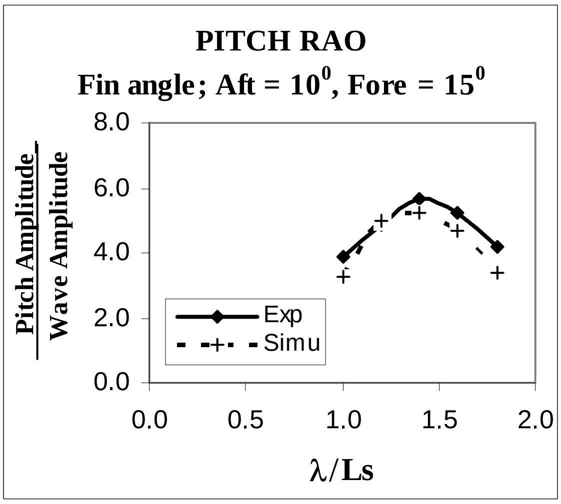


Figure 7.75 RAOs comparison of pitch at fin angle; aft = 10^0 and fore = 15^0

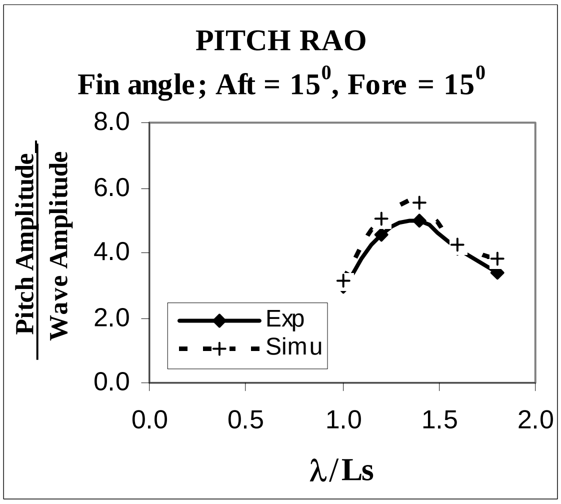


Figure 7.76 RAOs comparison of pitch at fin angle; aft = 15^0 and fore = 15^0

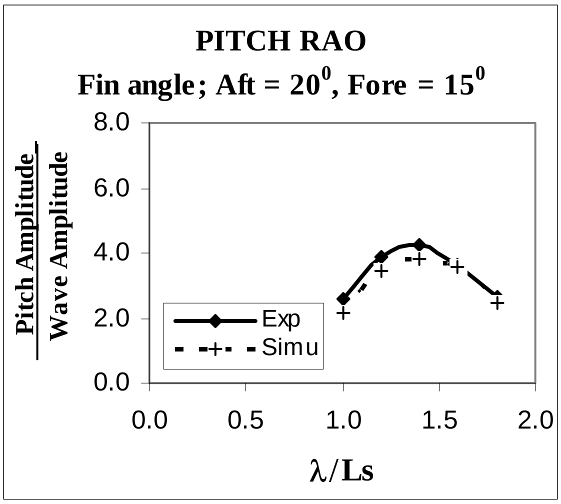


Figure 7.77 RAOs comparison of pitch at fin angle; aft = 5^0 and fore = 15^0

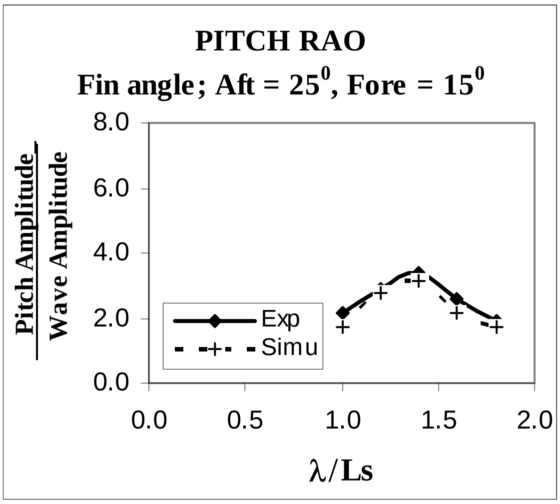


Figure 7.78 RAOs comparison of pitch at fin angle; aft = 25^0 and fore = 15^0

7.4.3 Time-Histories of Heave and Pitch Motion T=1.4 m and Vs=20 Knots

Case 2A

Fin Angle: Aft = 5^0 and Fore = 15^0

Case 2.A1

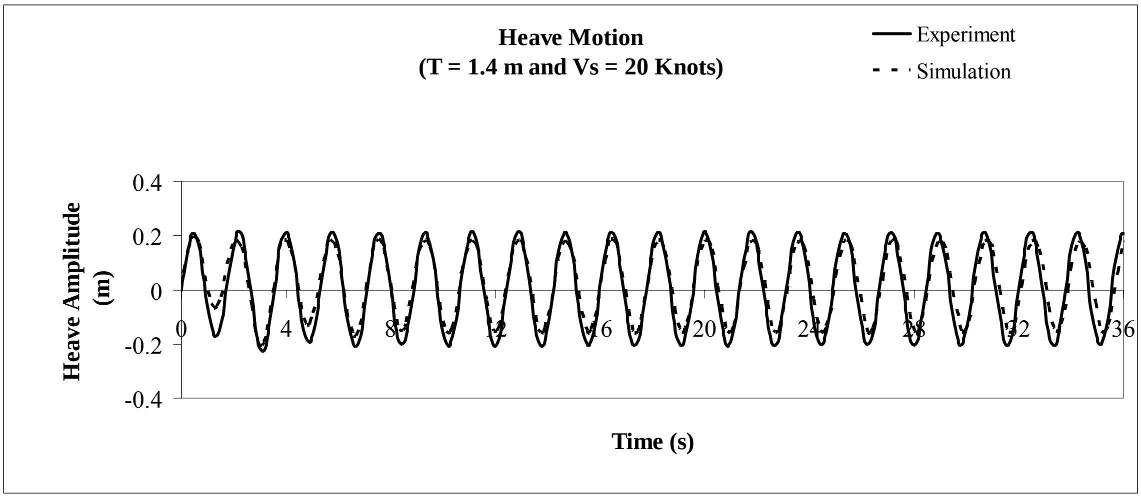


Figure 7.79 Heave motion on the Semi-SWATH vessel at wave height = 0.476 m, $\lambda/L_s = 1$

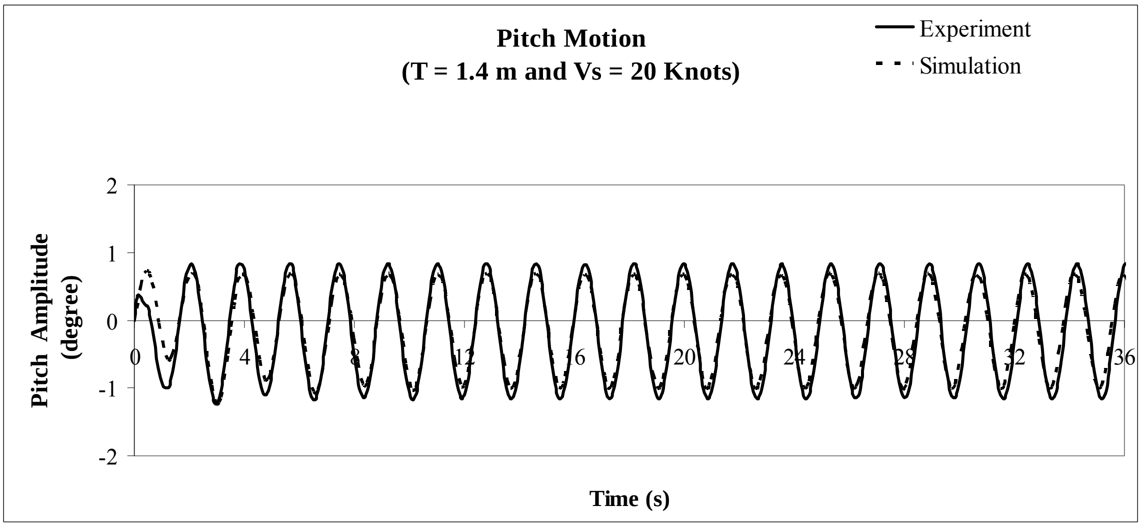


Figure 7.80 Pitch motion on the Semi-SWATH vessel at wave height = 0.476 m, $\lambda/L_s = 1.0$

Case 2.A2

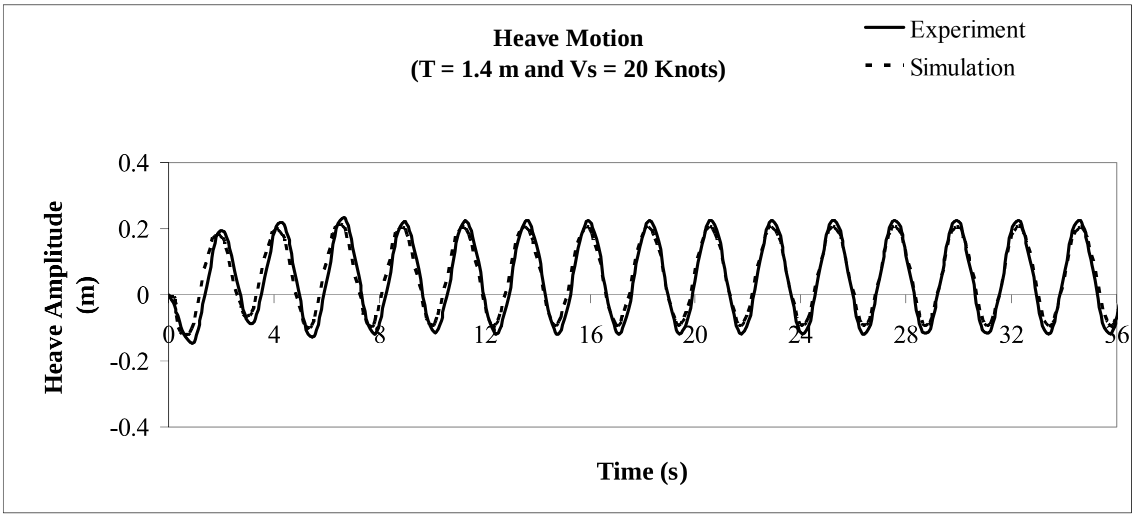


Figure 7.81 Heave motion on the Semi-SWATH vessel at wave height = 0.571 m, $\lambda/L_s = 1.2$

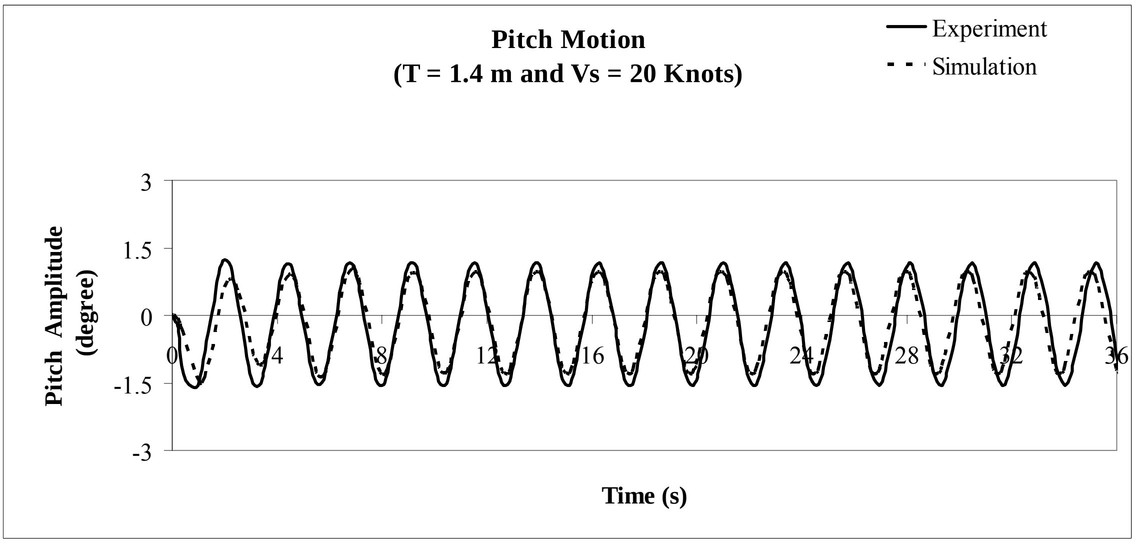


Figure 7.82 Pitch motion on the Semi-SWATH vessel at wave height = 0.571 m, $\lambda/L_s = 1.2$

Case 2.A3

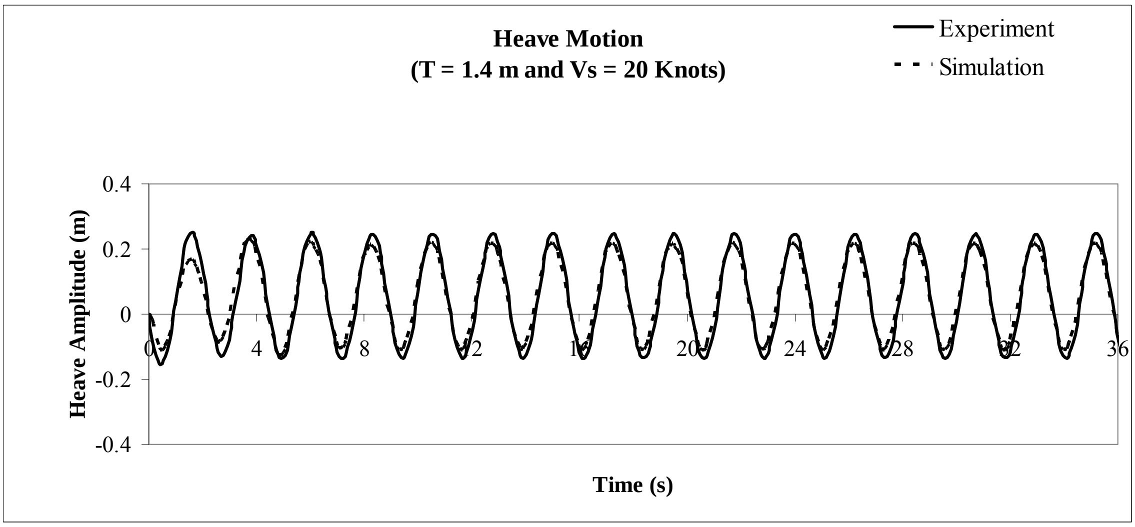


Figure 7.83 Heave motion on the Semi-SWATH vessel at wave height = 0.666 m, $\lambda/L_s = 1.4$

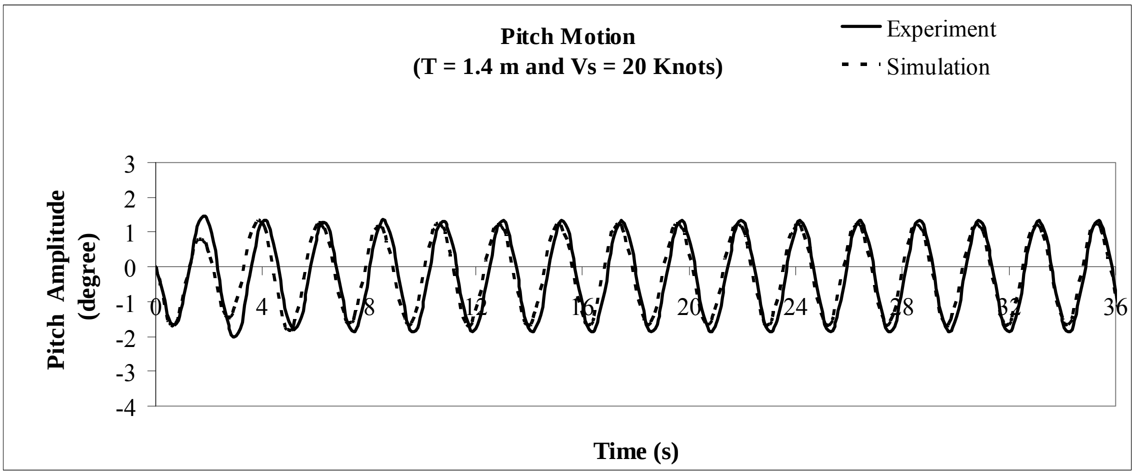


Figure 7.84 Pitch motion on the Semi-SWATH vessel at wave height = 0.666 m, $\lambda/L_s = 1.4$

Case 2.A4

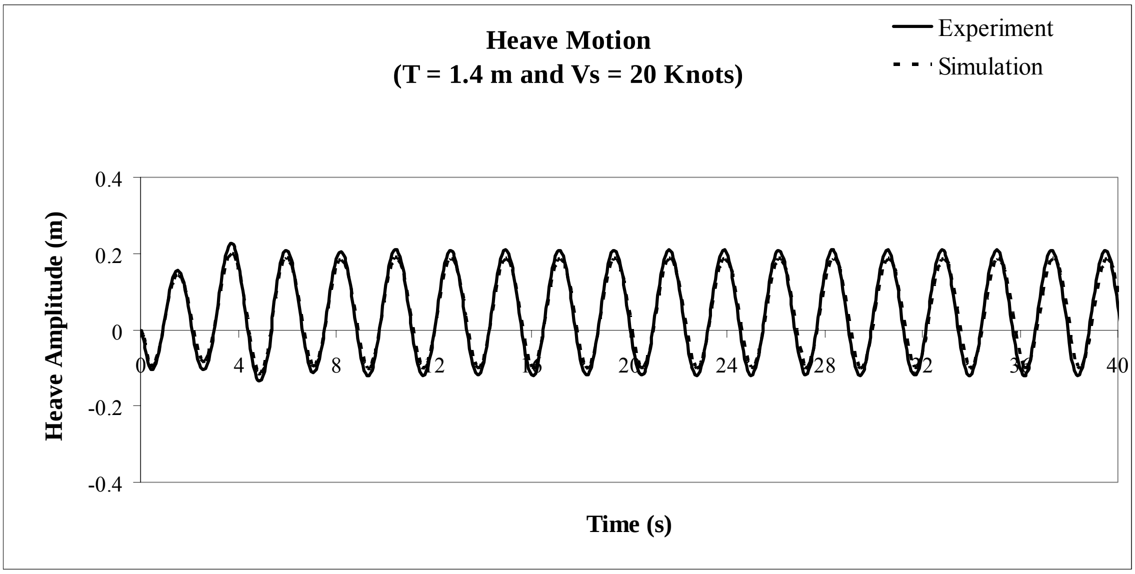


Figure 7.85 Heave motion on the Semi-SWATH vessel at wave height = 0.762 m, $\lambda/L_s = 1.6$

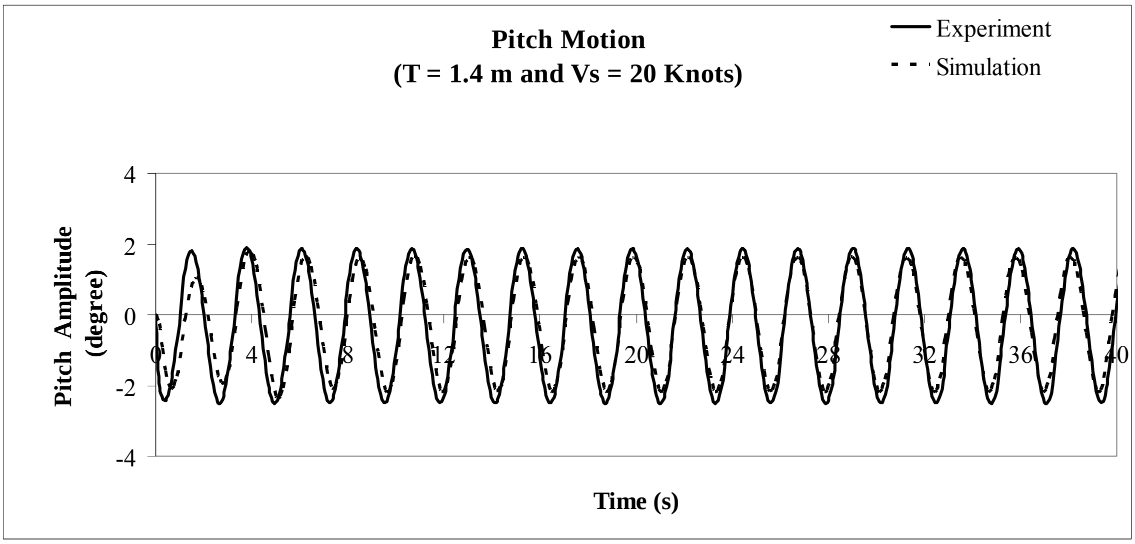


Figure 7.86 Pitch motion on the Semi-SWATH vessel at wave height = 0.762 m, $\lambda/L_s = 1.6$

Case 2.A5

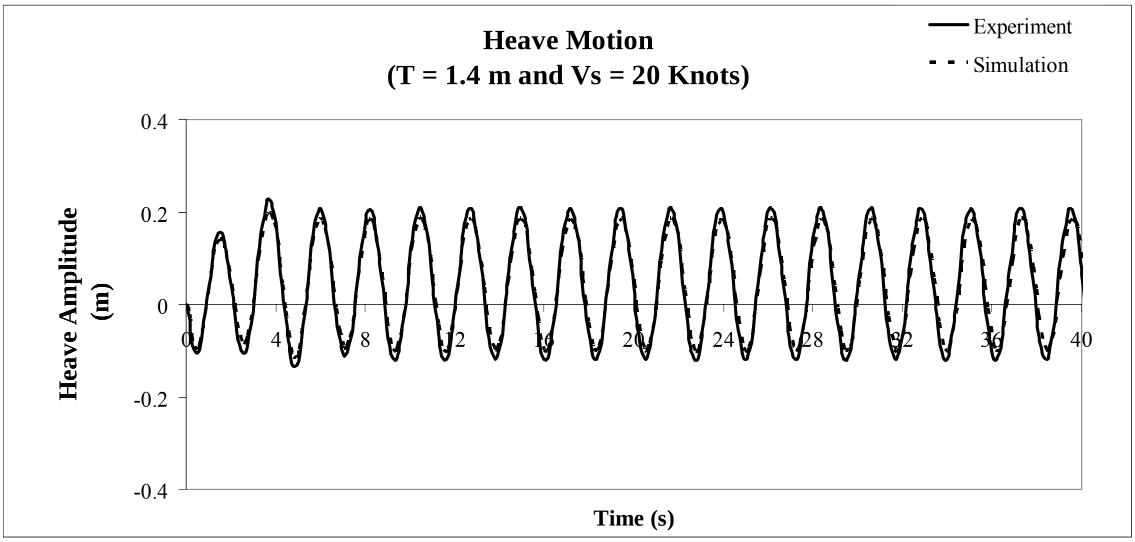


Figure 7.87 Heave motion on the Semi-SWATH vessel at wave height = 0.857 m, $\lambda/L_s = 1.8$

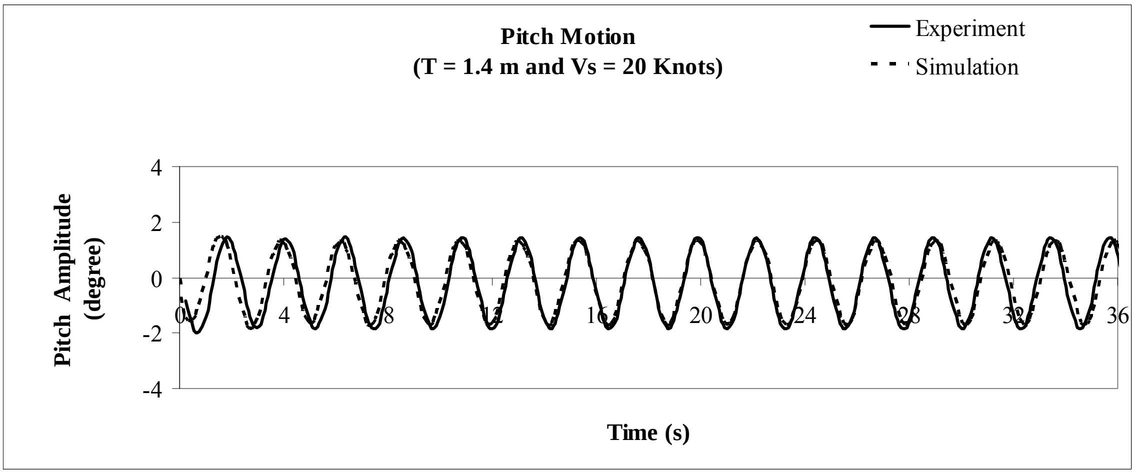


Figure 7.88 Pitch motion on the Semi-SWATH vessel at wave height = 0.857 m, $\lambda/L_s = 1.8$

Case 2B

Fin Angle: Aft $=10^0$ and Fore $= 15^0$

Case 2.B1

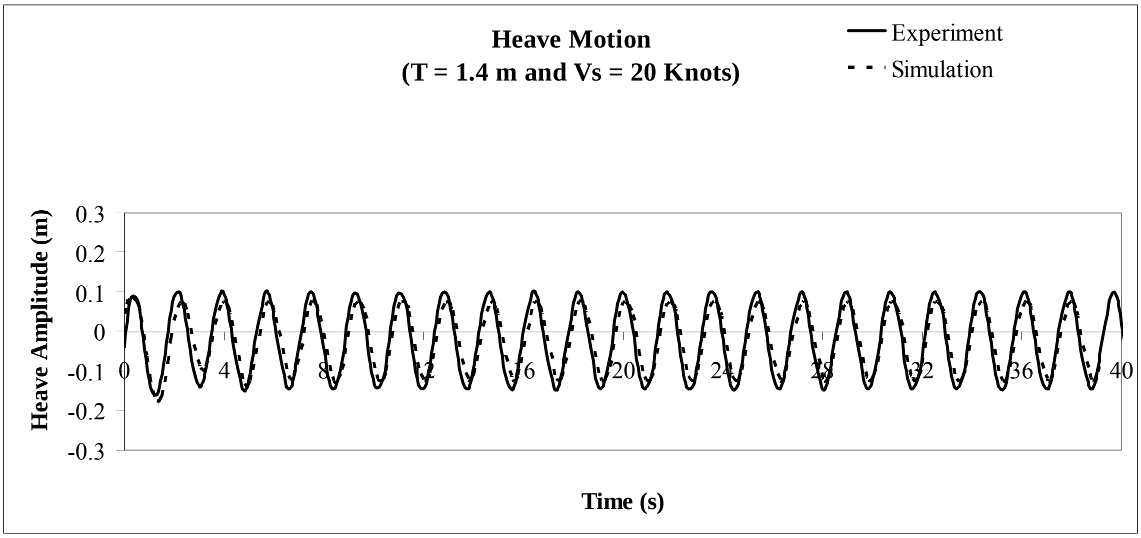


Figure 7.89 Heave motion on the Semi-SWATH vessel at wave height = 0.476 m, $\lambda/L_s = 1.0$

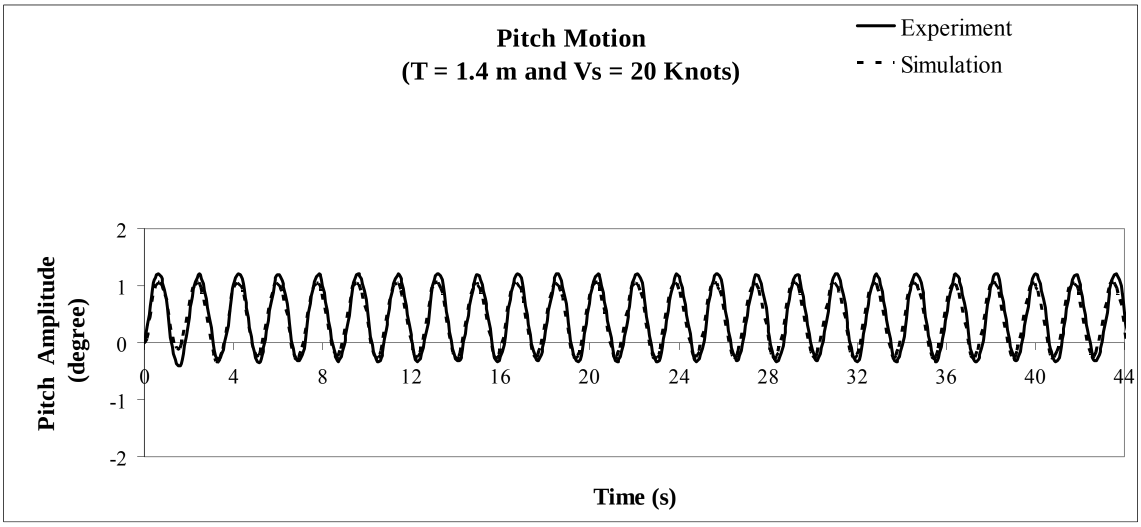


Figure 7.90 Pitch motion on the Semi-SWATH vessel at wave height = 0.476 m, $\lambda/L_s = 1.0$

Case 2.B2

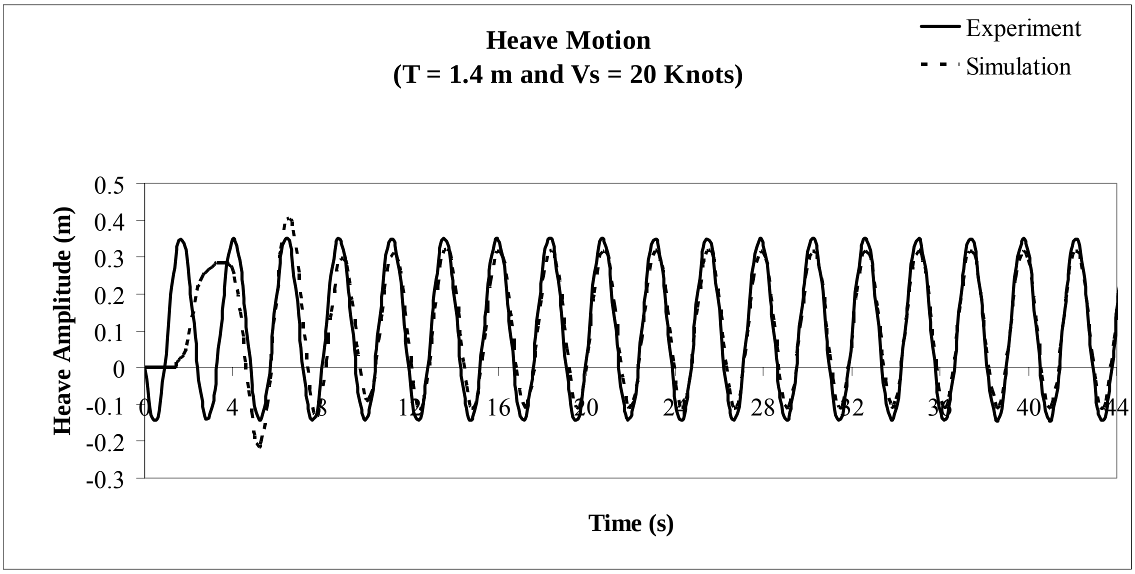


Figure 7.91 Heave motion on the Semi-SWATH vessel at wave height = 0.571 m, $\lambda/L_s = 1.2$

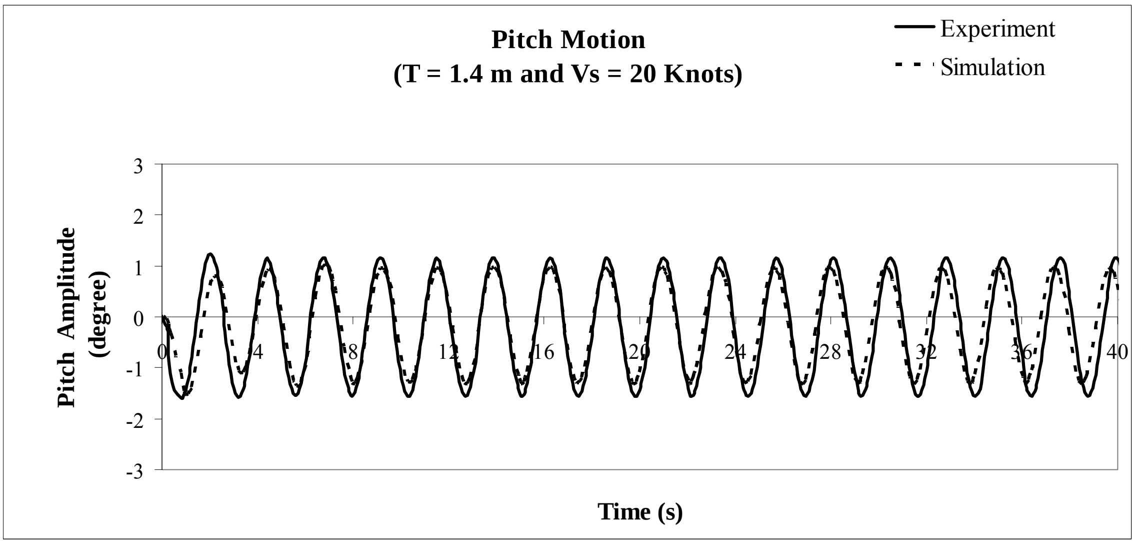


Figure 7.92 Pitch motion on the Semi-SWATH vessel at wave height = 0.571 m, $\lambda/L_s = 1.2$

Case 2.B3

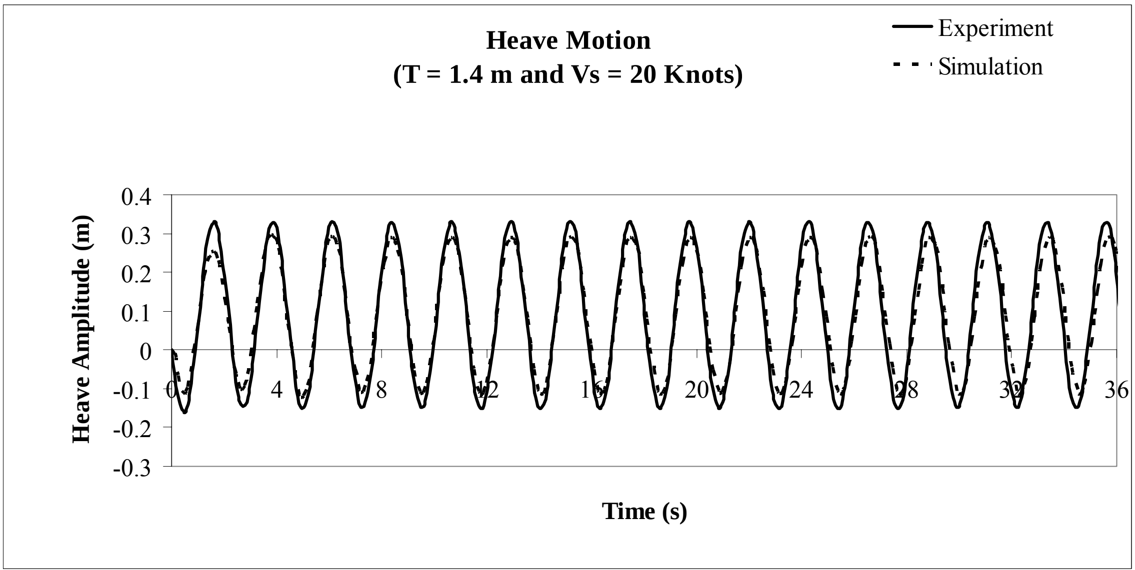


Figure 7.93 Heave motion on the Semi-SWATH vessel at wave height = 0.666 m, $\lambda/L_s = 1.4$

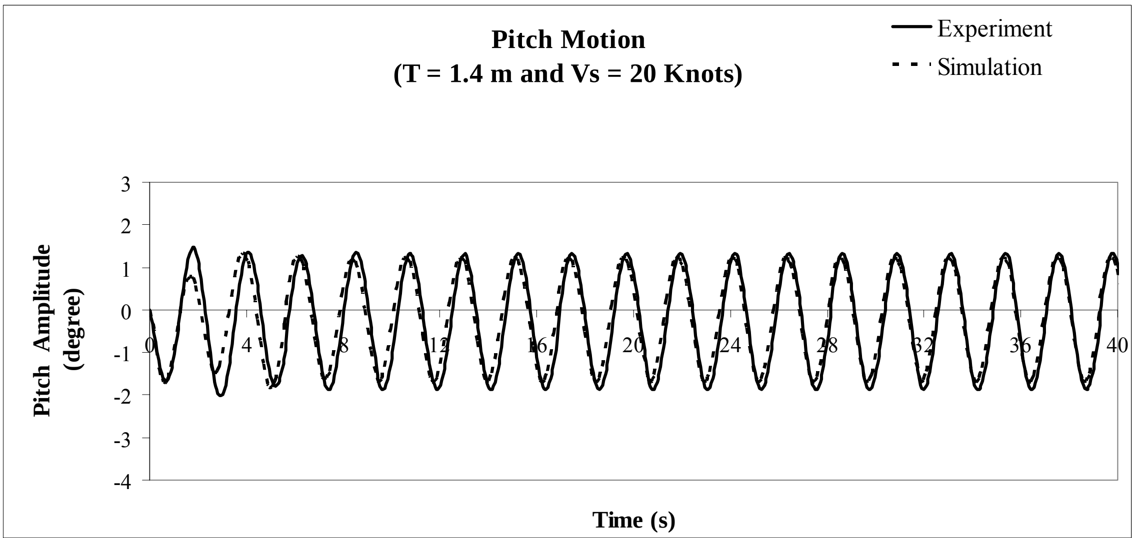


Figure 7.94 Pitch motion on the Semi-SWATH vessel at wave height = 0.666 m, $\lambda/L_s = 1.4$

Case 2.B4

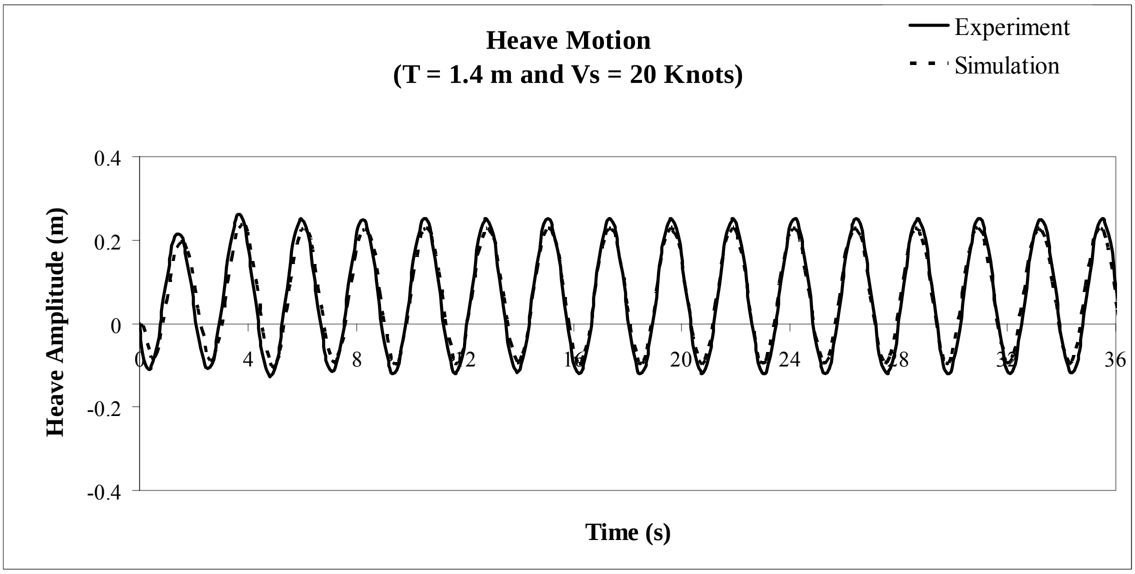


Figure 7.95 Heave motion on the Semi-SWATH vessel at wave height = 0.762 m, $\lambda/L_s = 1.6$

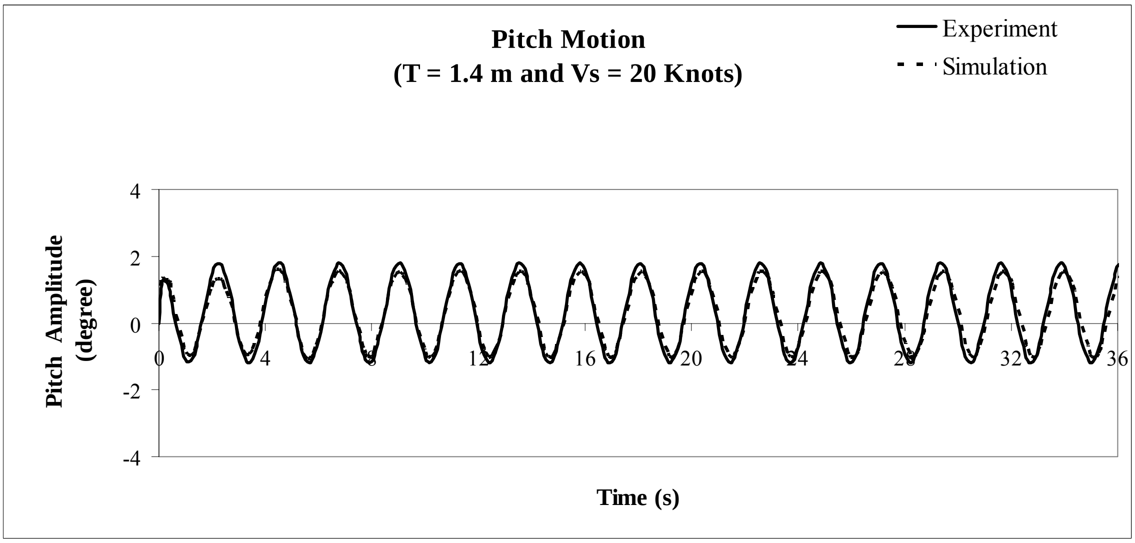


Figure 7.96 Pitch motion on the Semi-SWATH vessel at wave height = 0.762 m, $\lambda/L_s = 1.6$

Case 2.B5

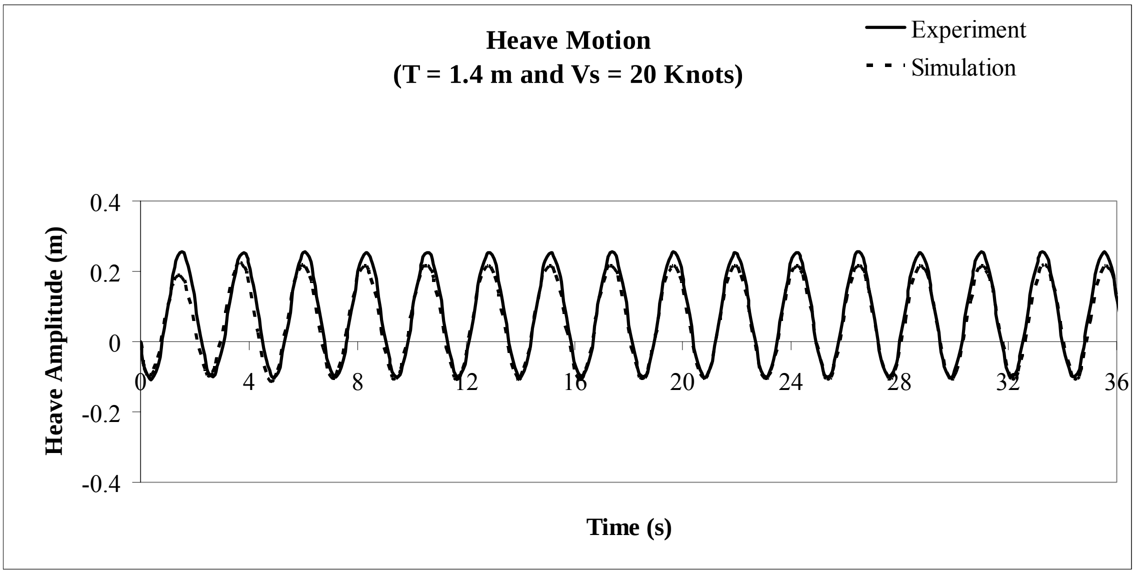


Figure 7.97 Heave motion on the Semi-SWATH vessel at wave height = 0.857 m, $\lambda/L_s = 1.8$

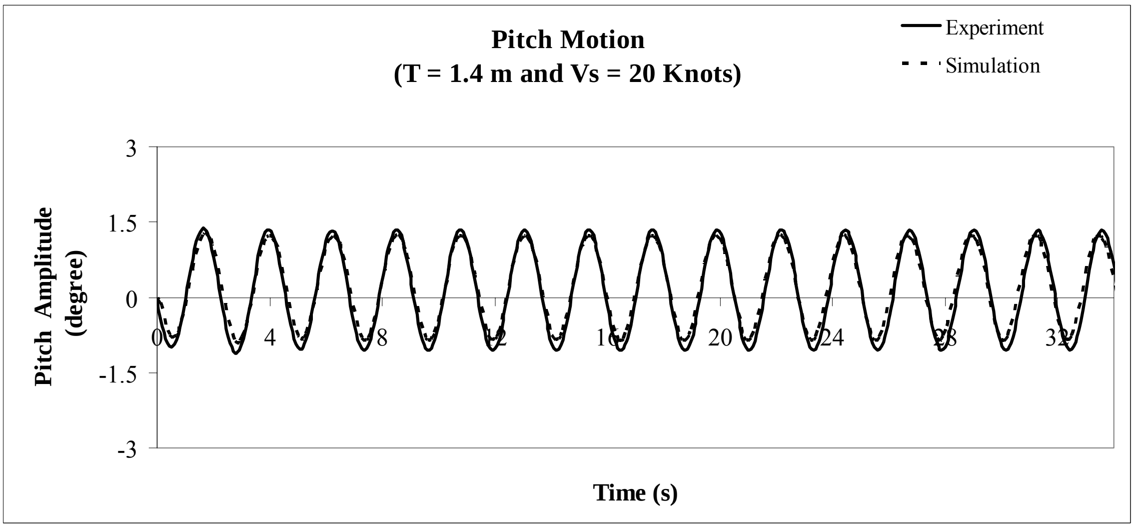


Figure 7.98 Pitch motion on the Semi-SWATH vessel at wave height = 0.857 m, $\lambda/L_s = 1.8$

Case 2C

Fin Angle: Aft =15^0 and Fore = 15^0

Case 2.C1

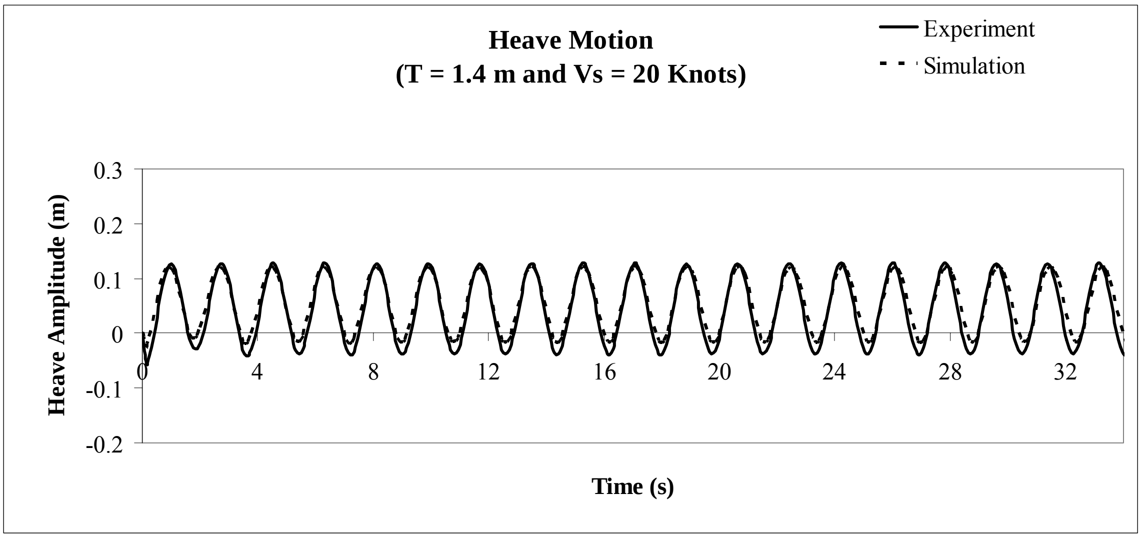


Figure 7.99 Heave motion on the Semi-SWATH vessel at wave height = 0.476 m, $\lambda/L_s = 1.0$

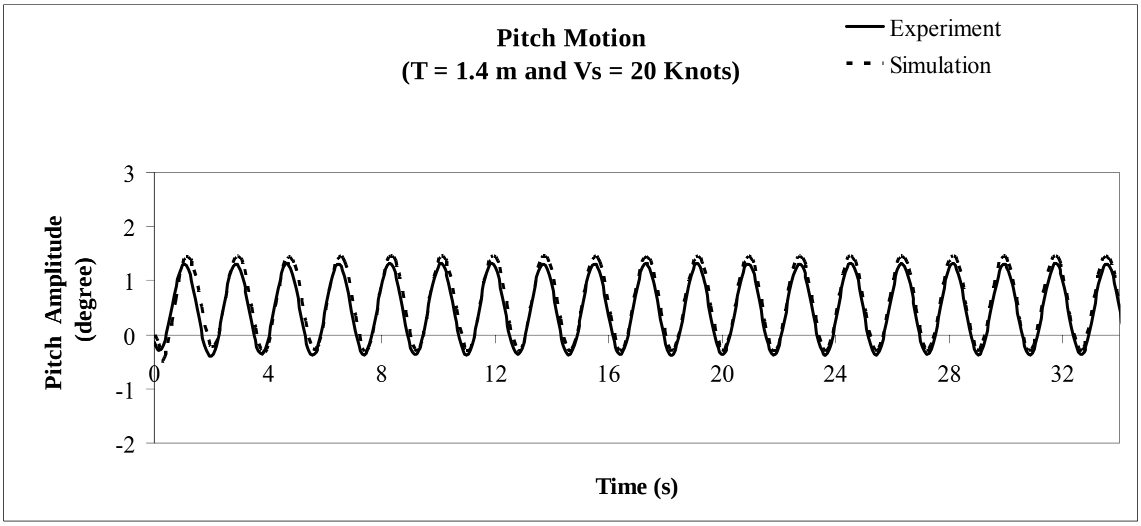


Figure 7.100 Pitch motion on the Semi-SWATH vessel at wave height = 0.476 m, $\lambda/L_s = 1.0$

Case 2.C2

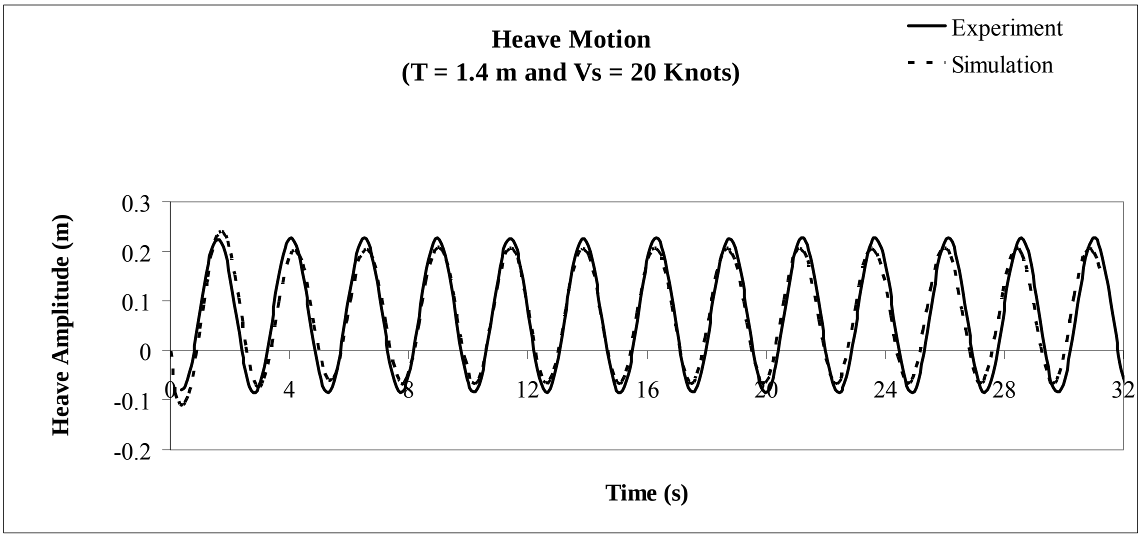


Figure 7.101 Heave motion on the Semi-SWATH vessel at wave height = 0.571 m, $\lambda/L_s = 1.2$

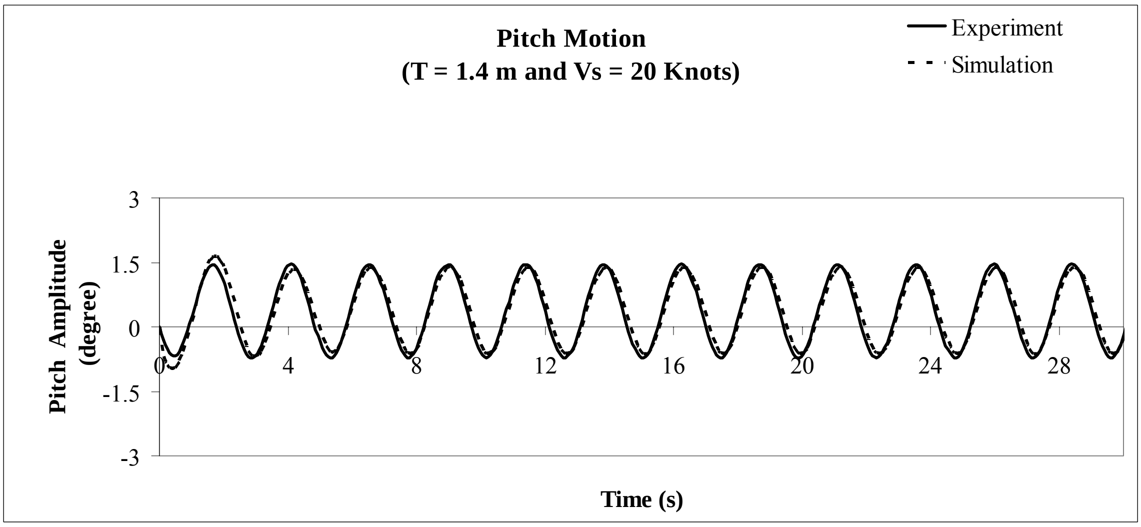


Figure 7.102 Pitch motion on the Semi-SWATH vessel at wave height = 0.571 m, $\lambda/L_s = 1.2$

Case 2.C3

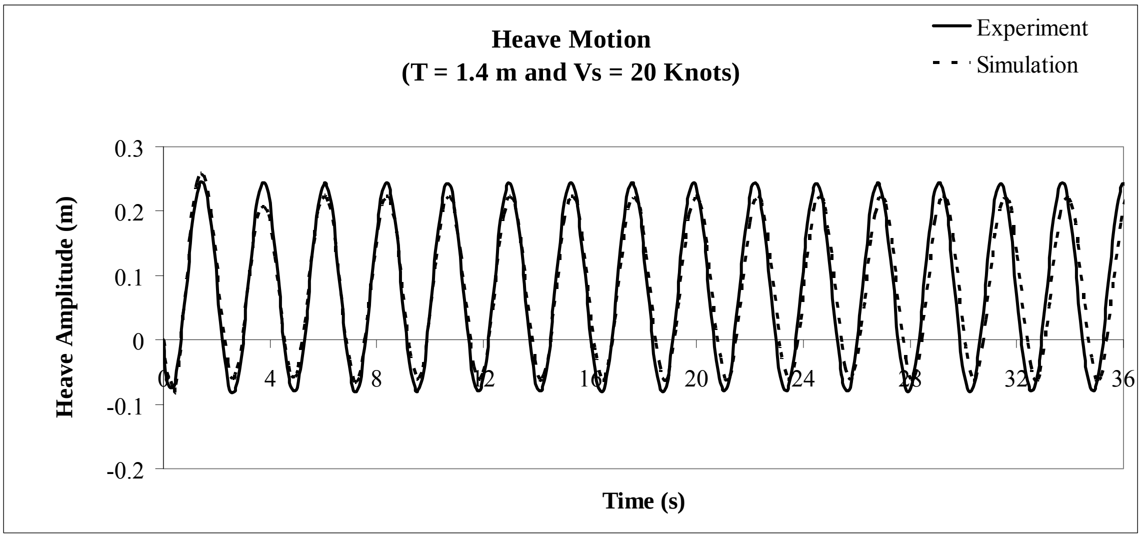


Figure 7.103 Heave motion on the Semi-SWATH vessel at wave height = 0.666 m, $\lambda/L_s = 1.4$

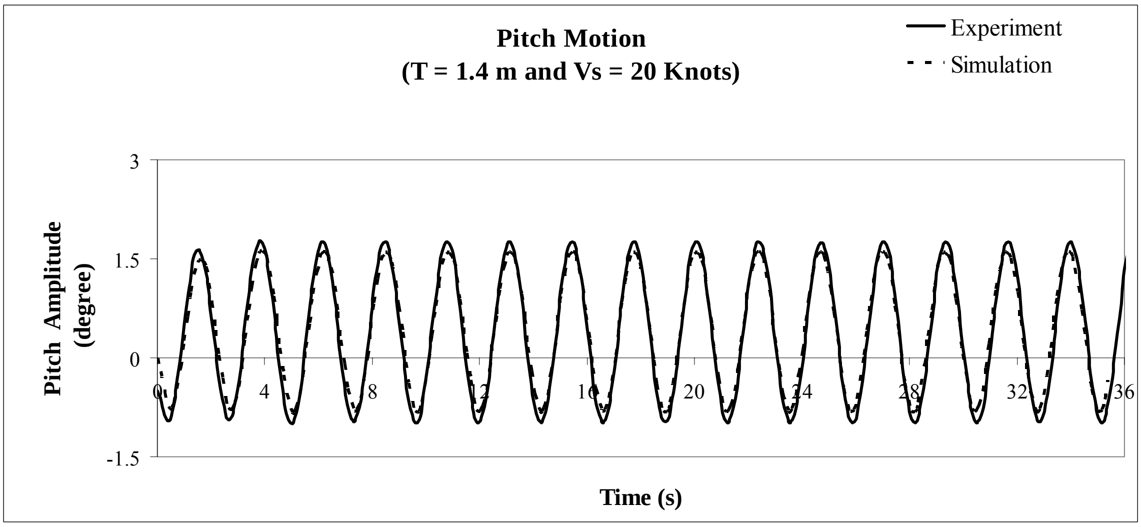


Figure 7.104 Pitch motion on the Semi-SWATH vessel at wave height = 0.666 m, $\lambda/L_s = 1.4$

Case 2.C4

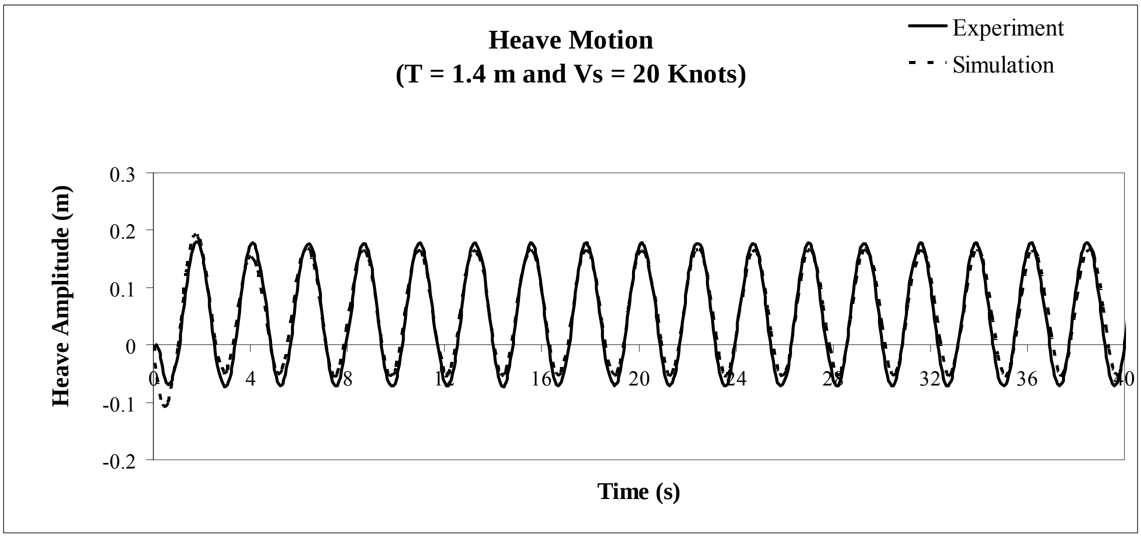


Figure 7.105 Heave motion on the Semi-SWATH vessel at wave height = 0.762 m, $\lambda/L_s = 1.6$

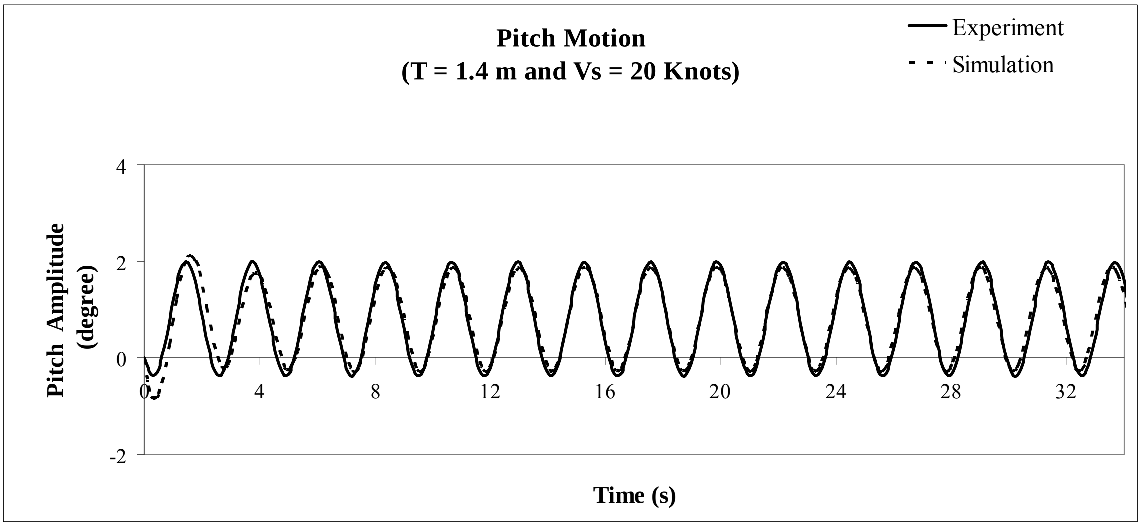


Figure 7.106 Pitch motion on the Semi-SWATH vessel at wave height = 0.762 m, $\lambda/L_s = 1.6$

Case 2.C5

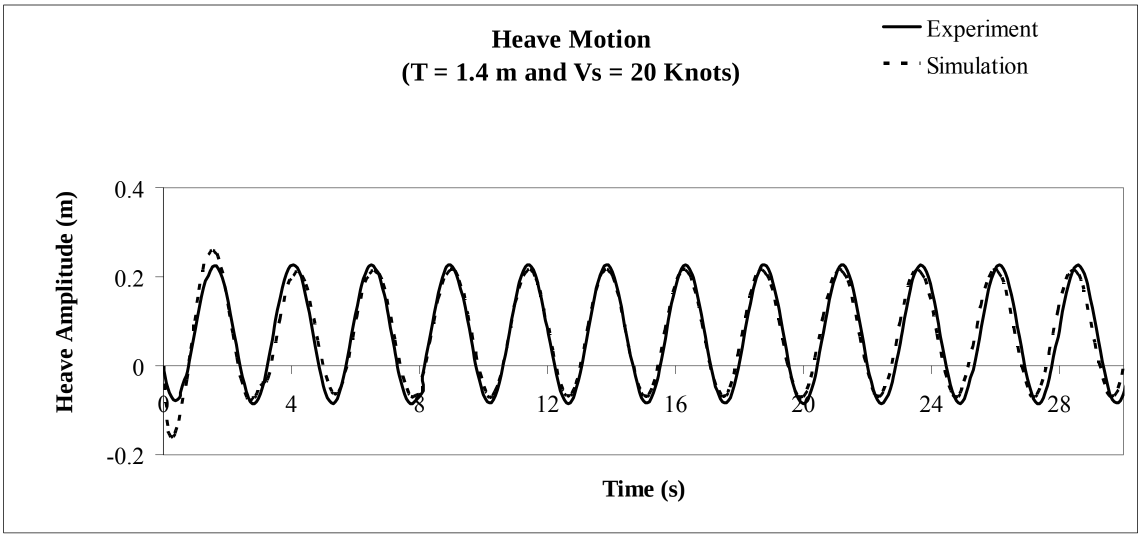


Figure 7.107 Heave motion on the Semi-SWATH vessel at wave height = 0.857 m, $\lambda/L_s = 1.8$

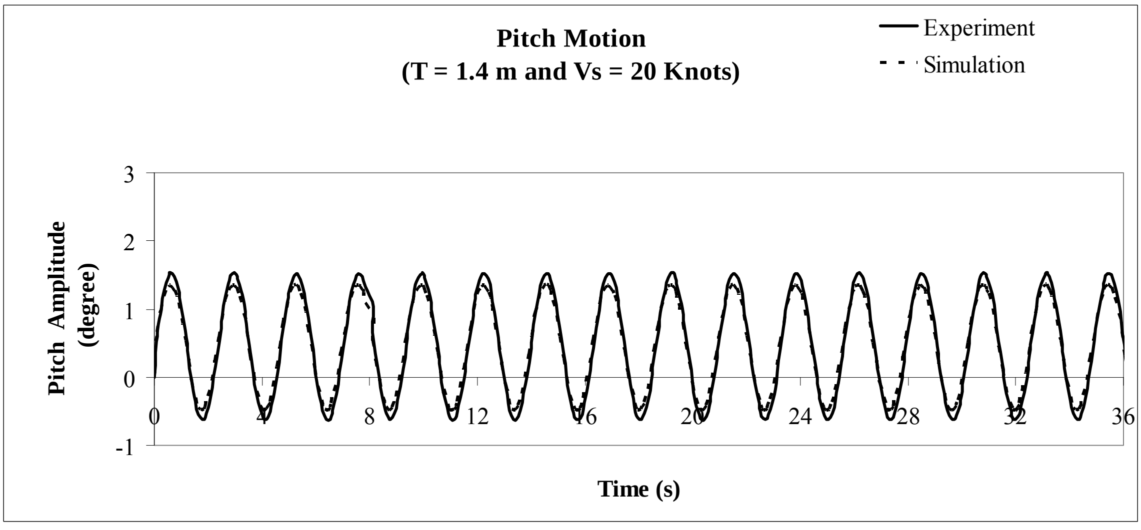


Figure 7.108 Pitch motion on the Semi-SWATH vessel at wave height = 0.857 m, $\lambda/L_s = 1.8$

Case 2D

Fin Angle: Aft $=20^0$ and Fore $= 15^0$

Case 2.D1

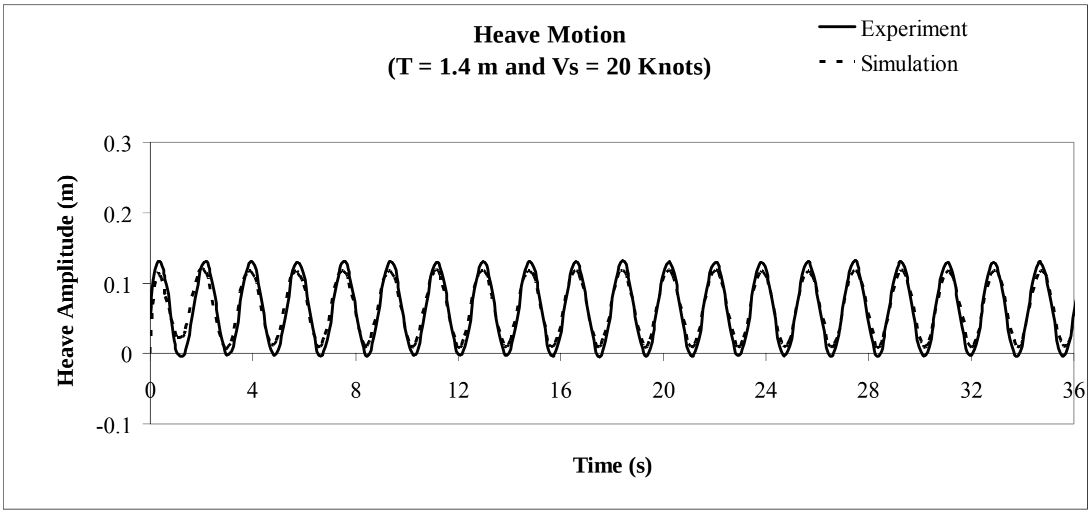


Figure 7.109 Heave motion on the Semi-SWATH vessel at wave height = 0.476 m, $\lambda/L_s = 1.0$

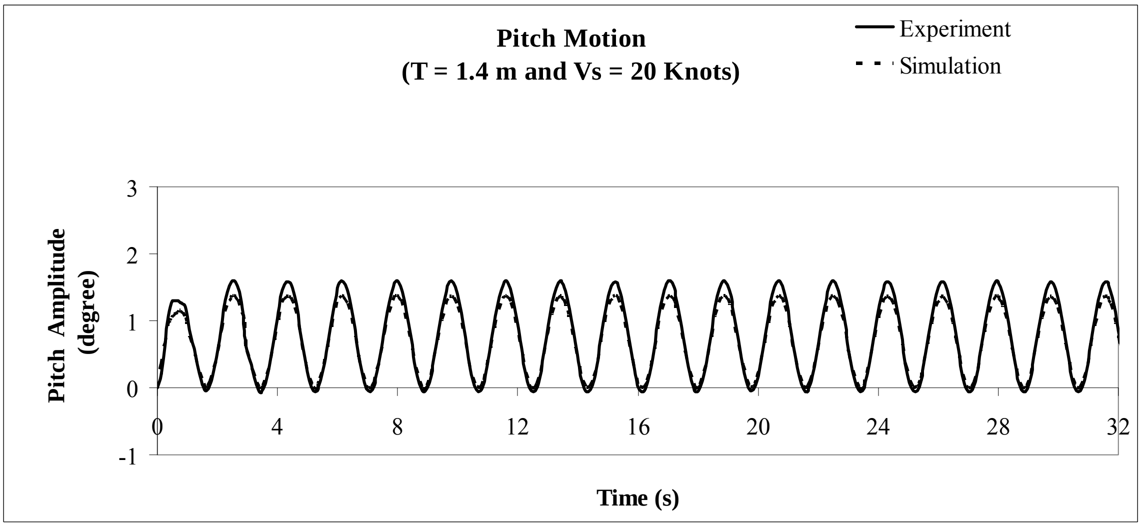


Figure 7.110 Pitch motion on the Semi-SWATH vessel at wave height = 0.476 m, $\lambda/L_s = 1$

Case 2.D2

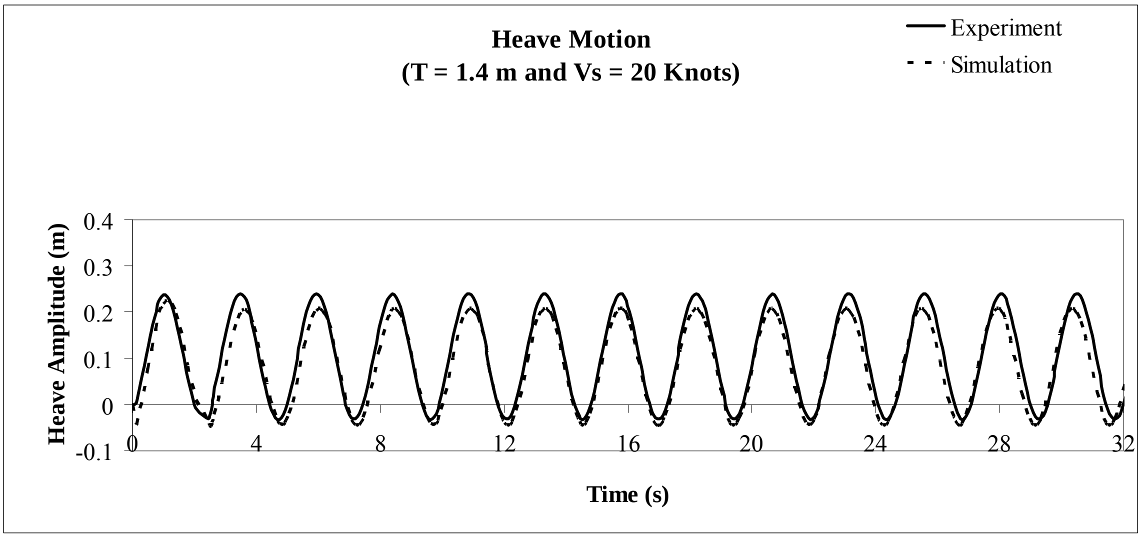


Figure 7.111 Heave motion on the Semi-SWATH vessel at wave height = 0.571 m, $\lambda/L_s = 1.2$

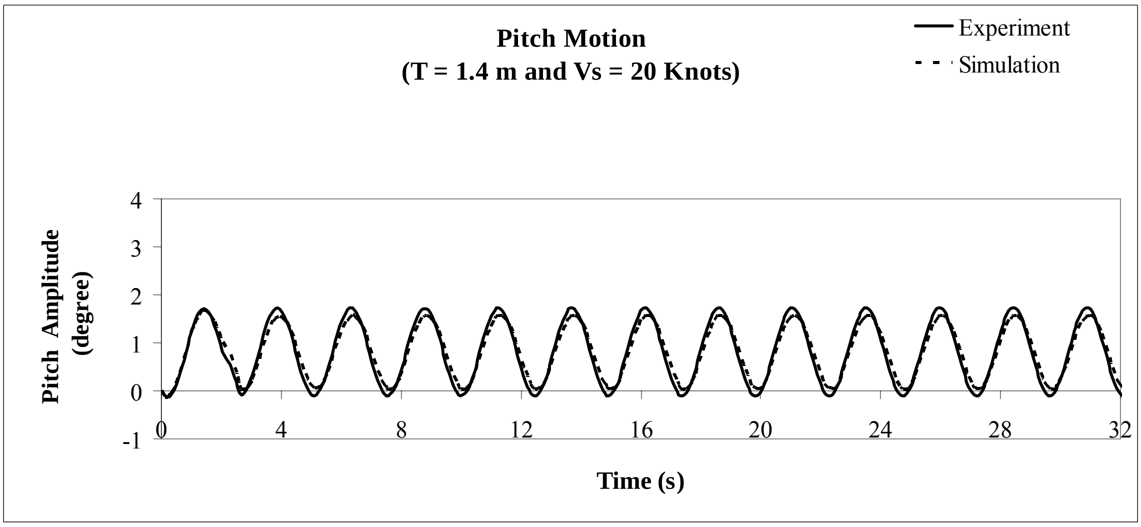


Figure 7.112 Pitch motion on the Semi-SWATH vessel at wave height = 0.571 m, $\lambda/L_s = 1.2$

Case 2.D3

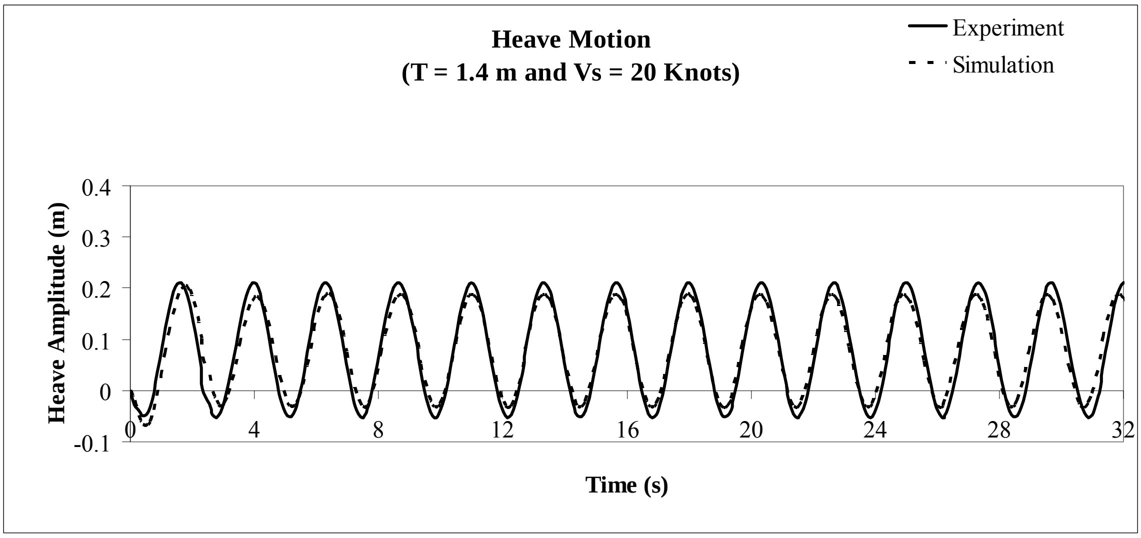


Figure 7.113 Heave motion on the Semi-SWATH vessel at wave height = 0.666 m, $\lambda/L_s = 1.4$

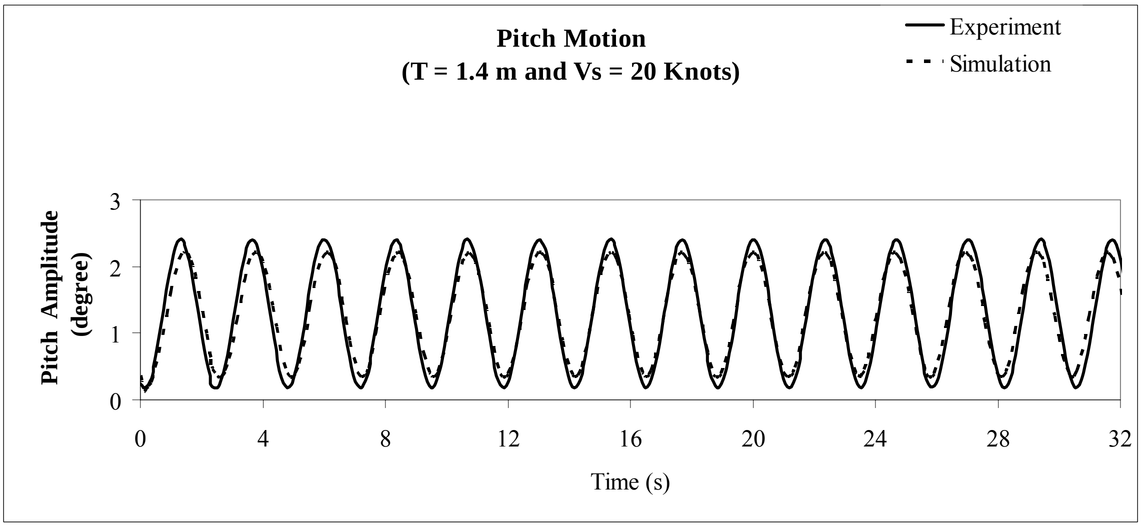


Figure 7.114 Pitch motion on the Semi-SWATH vessel at wave height = 0.666 m, $\lambda/L_s = 1.4$

Case 2.D4

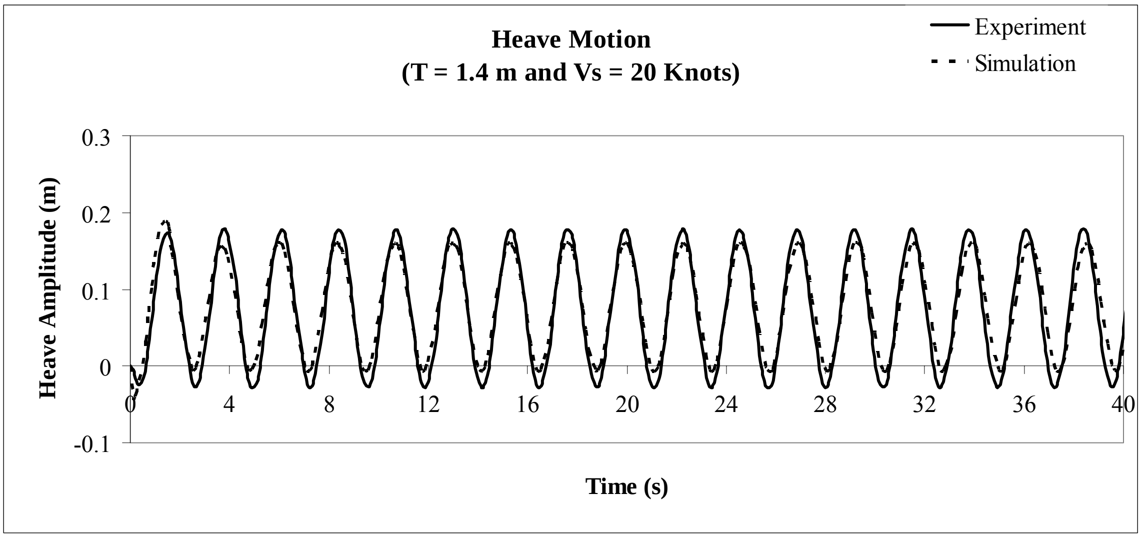


Figure 7.115 Heave motion on the Semi-SWATH vessel at wave height = 0.762 m,

$\lambda/L_s = 1.6$

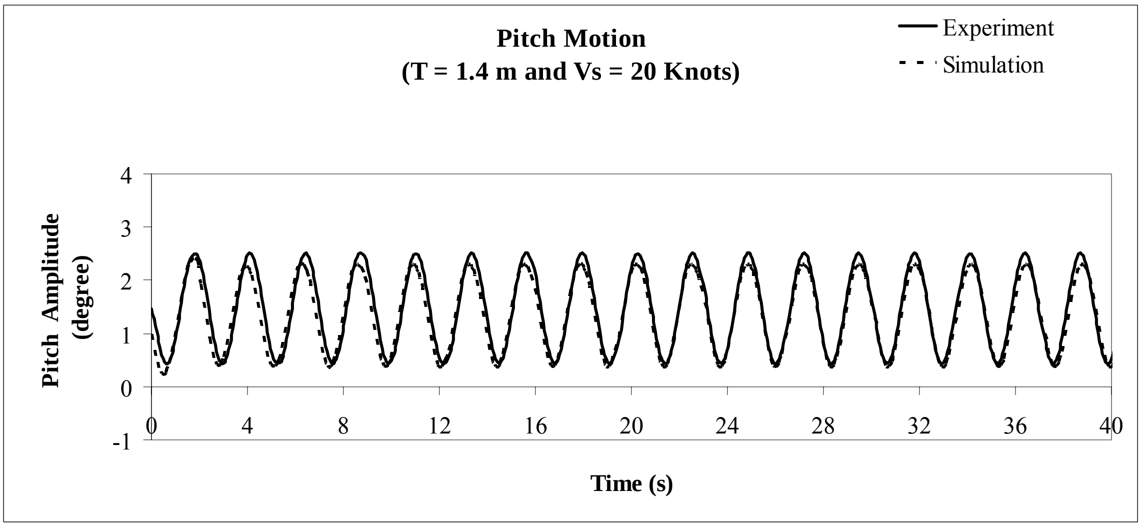


Figure 7.116 Pitch motion on the Semi-SWATH vessel at wave height = 0.762 m,

$\lambda/L_s = 1.6$

Case 2.D5

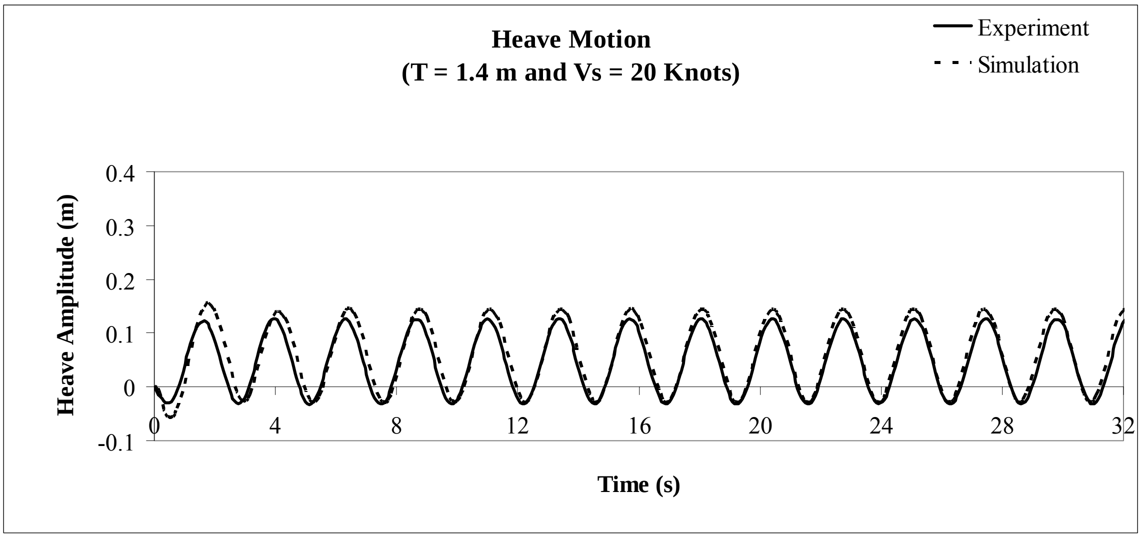


Figure 7.117 Heave motion on the Semi-SWATH vessel at wave height = 0.857 m, $\lambda/L_s = 1.8$

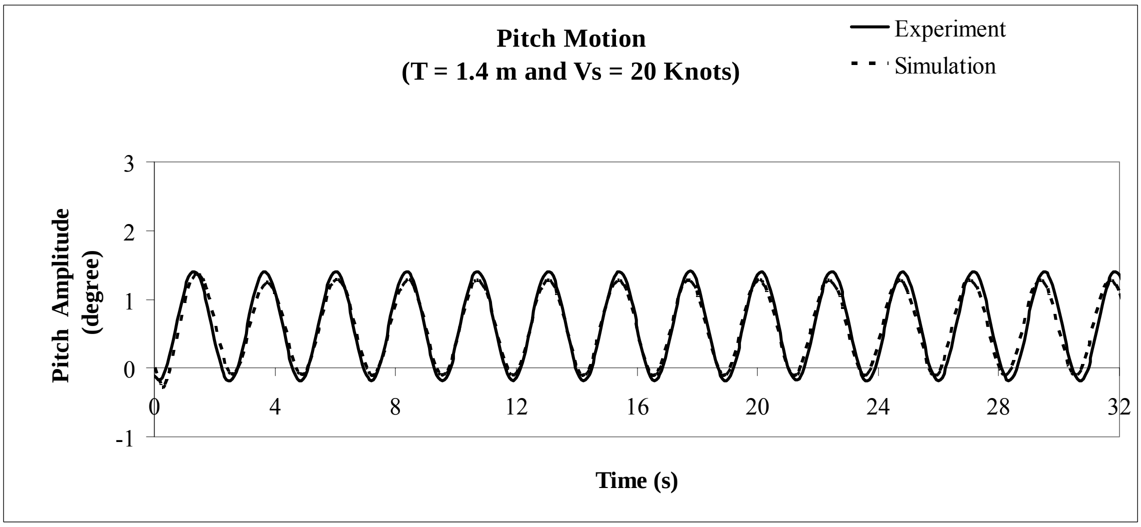


Figure 7.118 Pitch motion on the Semi-SWATH vessel at wave height = 0.857 m, $\lambda/L_s = 1.8$

Case 2E

Fin Angle: Aft =25^0 and Fore = 15^0

Case 2.E1

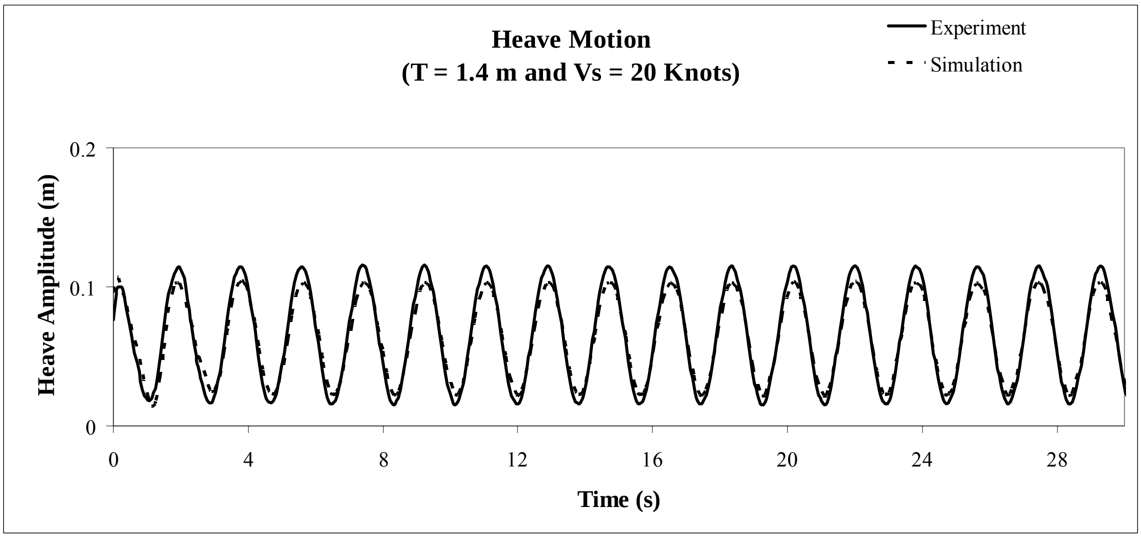


Figure 7.119 Heave motion on the Semi-SWATH vessel at wave height = 0.476 m, $\lambda/L_s = 1$

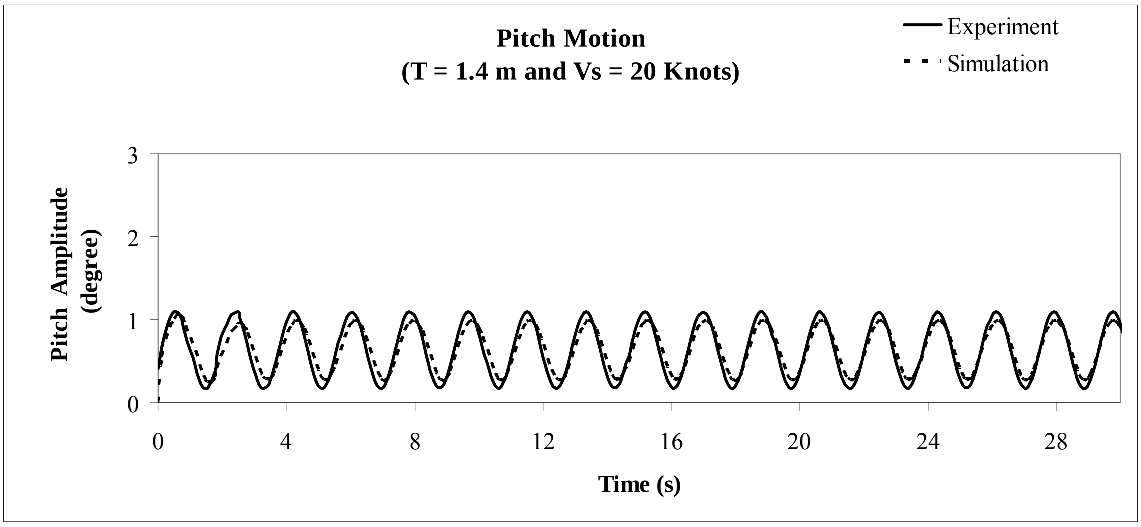


Figure 7.120 Pitch motion on the Semi-SWATH vessel at wave height = 0.476 m, $\lambda/L_s = 1$

Case 2.E2

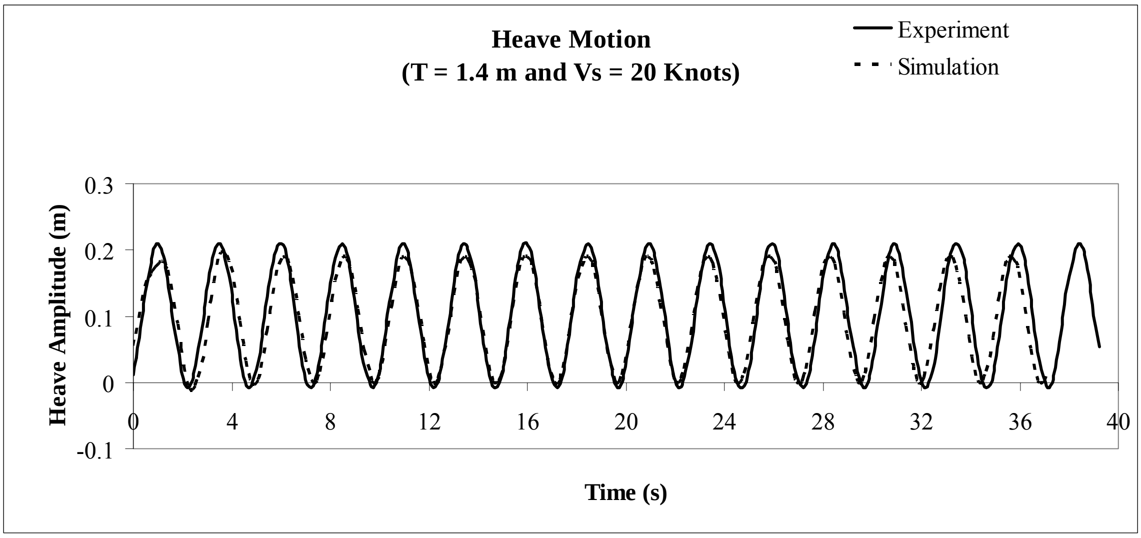


Figure 7.121 Heave motion on the Semi-SWATH vessel at wave height = 0.571 m, $\lambda/L_s = 1.2$

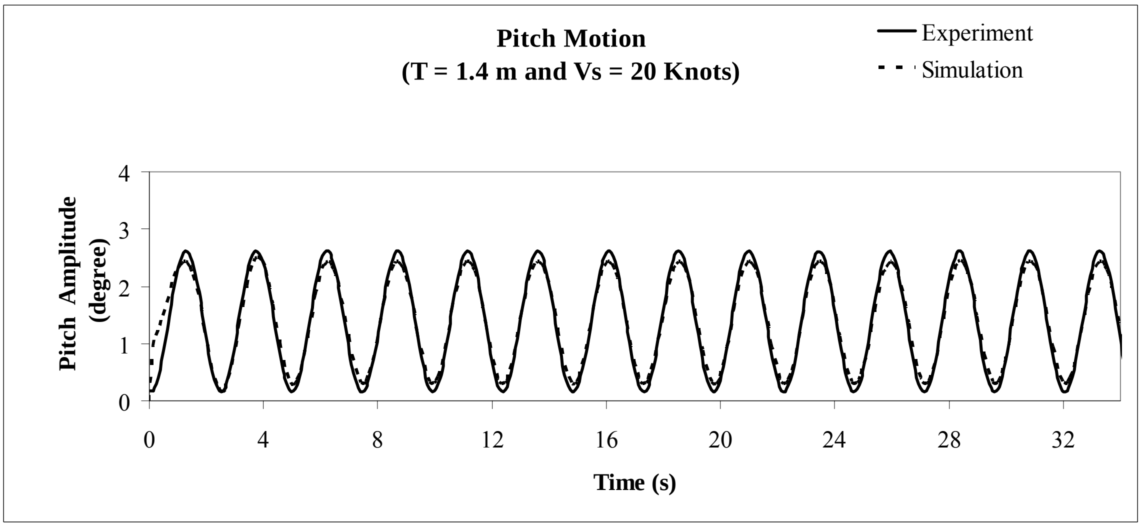


Figure 7.122 Pitch motion on the Semi-SWATH vessel at wave height = 0.571 m, $\lambda/L_s = 1.2$

Case 2.E3

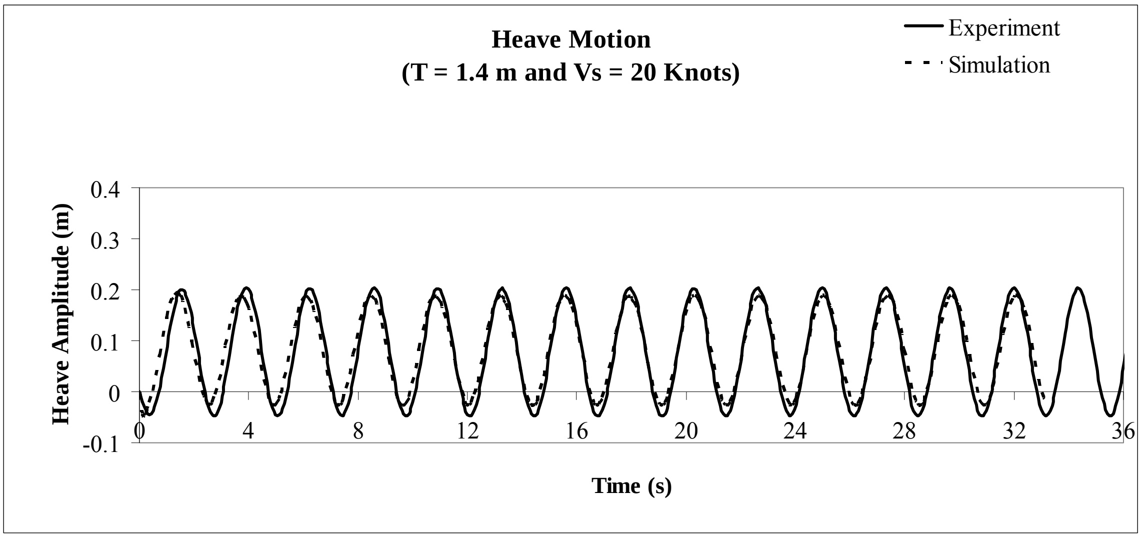


Figure 7.123 Heave motion on the Semi-SWATH vessel at wave height = 0.666 m, $\lambda/L_s = 1.4$

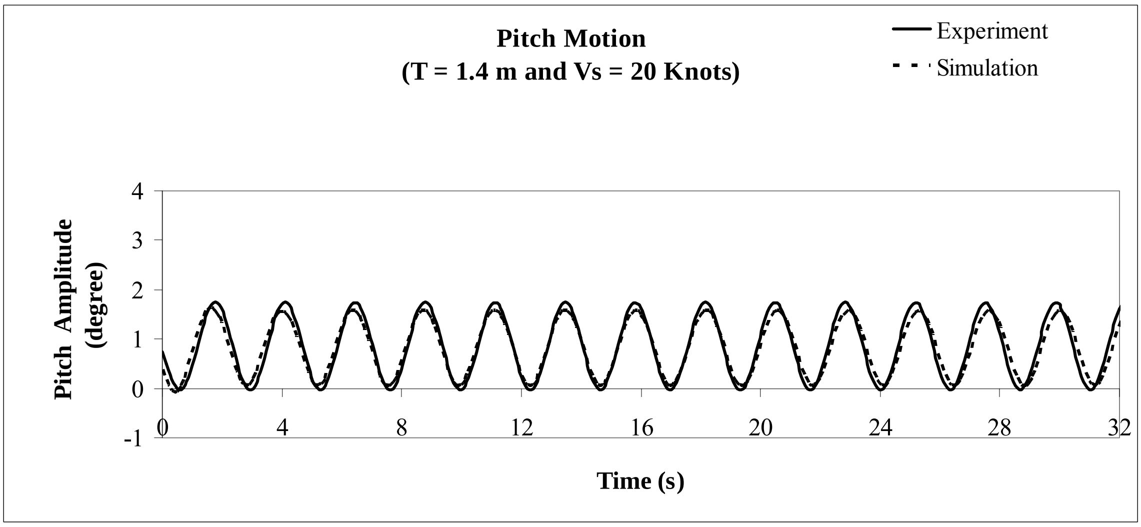


Figure 7.124 Pitch motion on the Semi-SWATH vessel at wave height = 0.666 m, $\lambda/L_s = 1.4$

Case 2.E4

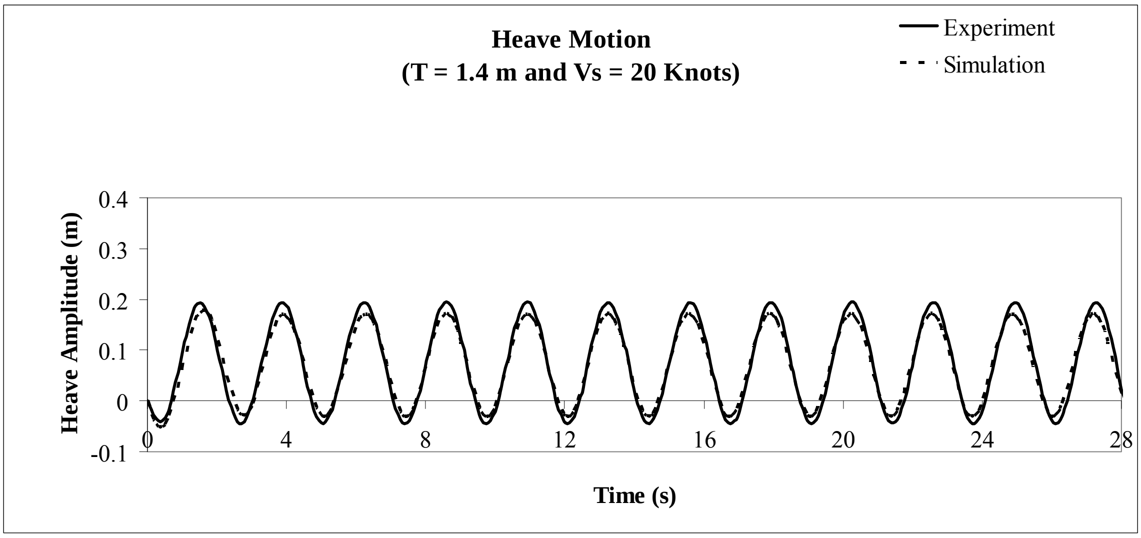


Figure 7.125 Heave motion on the Semi-SWATH vessel at wave height = 0.762 m, $\lambda/L_s = 1.6$

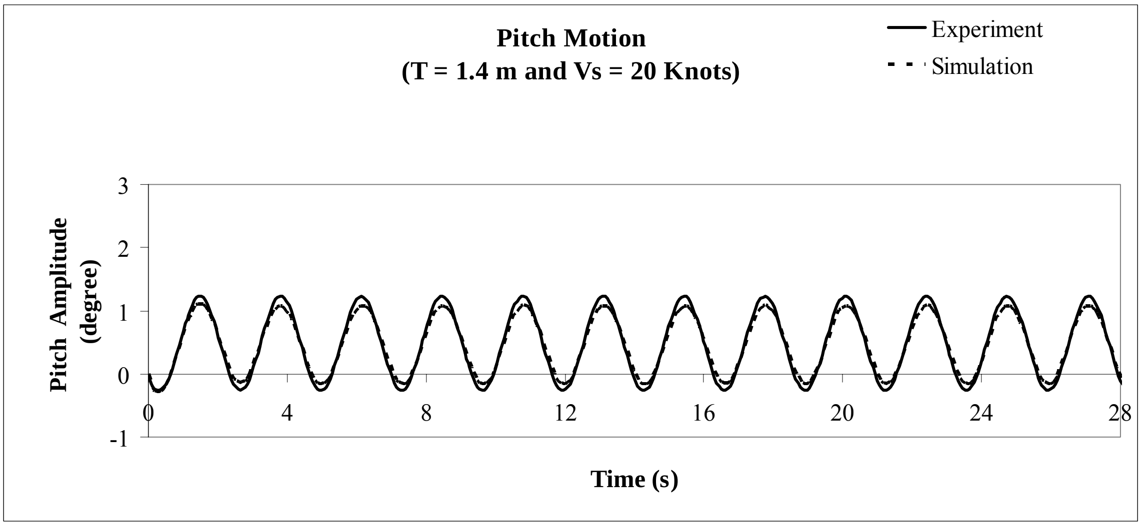


Figure 7.126 Pitch motion on the Semi-SWATH vessel at wave height = 0.762 m, $\lambda/L_s = 1.6$

Case 2.E5

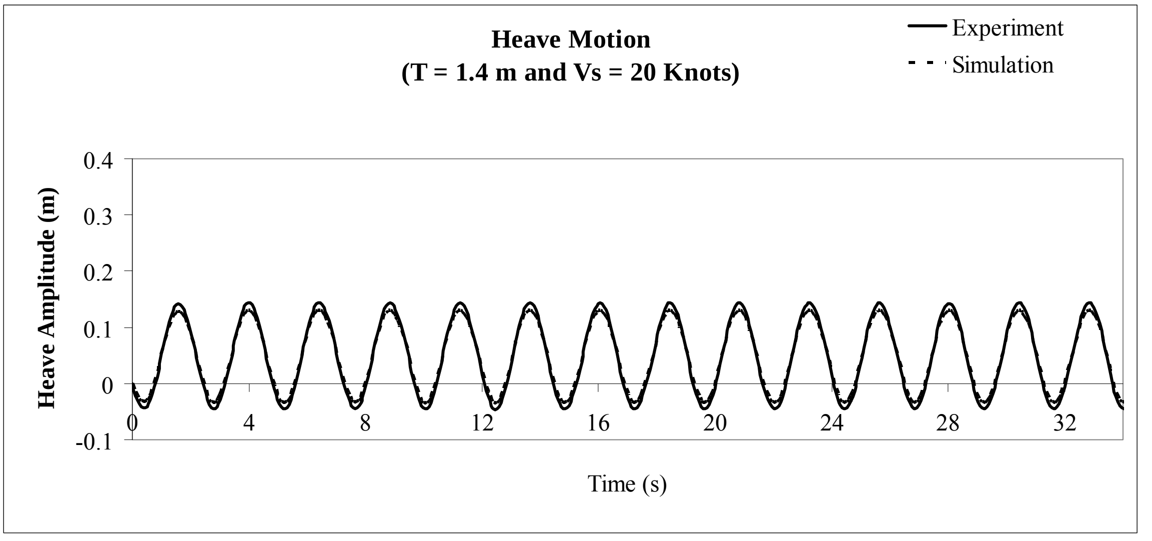


Figure 7.127 Heave motion on the Semi-SWATH vessel at wave height = 0.857 m, $\lambda/L_s = 1.8$

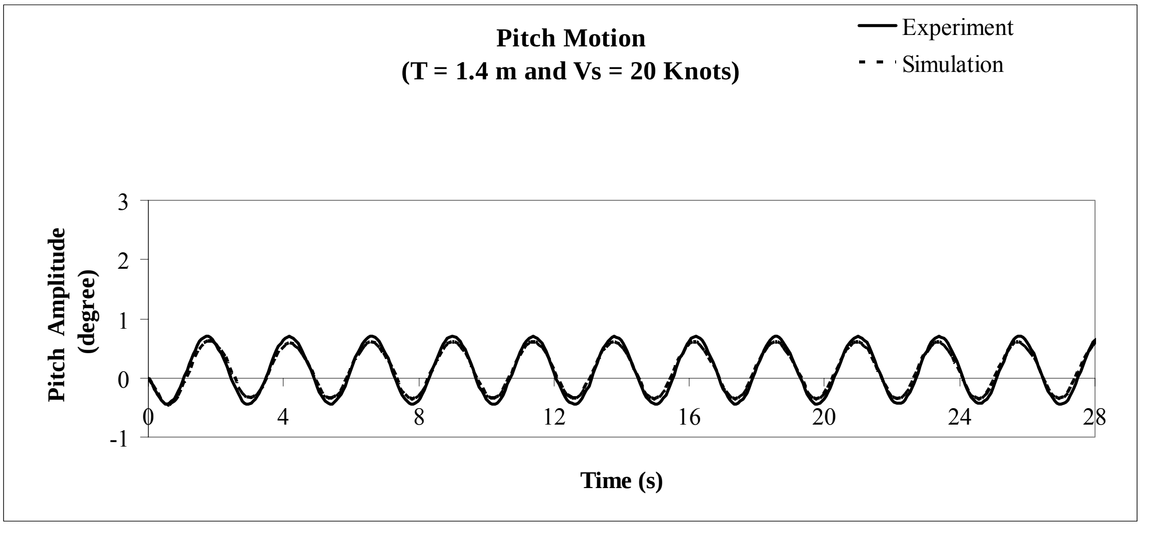


Figure 7.128 Pitch motion on the Semi-SWATH vessel at wave height = 0.857 m, $\lambda/L_s = 1.8$

7.5 Concluding Remarks

Generally, the comparison between simulation program and experimental results, gives confidence to develop seakeeping prediction by utilising the simulation program. Qualitatively, the present mathematical model successfully simulate the motion response prediction of heave and pitch for a Semi-SWATH vessel both with and without fin stabilizers at medium (15 knots) and high-speed (20 knots) conditions at range ratio of wavelengths to the model length ($\frac{\lambda}{L_{Ship}}$) is about 0.5-2.5 with wave height to wavelength ratio at the experiments was 1/25.

CHAPTER 8

DISCUSSION

8.1 General

The twin-hull vessels are known to have more stable platform and provides a better seakeeping qualities as compared to mono-hull. However, the significant drawback of the twin-hull vessel especially Semi-SWATH vessel is when encountering head-sea at high forward speed. Due to its low stiffness, it has a tendency for large pitch motions. In this condition, deck wetness, slamming, and structural damage are the most dangerous situation that should be considered in vessel seakeeping performances. For this reason, two modes of coupled motions are considered in this study, there are heave and pitch motions as the main parametric that have quite contributions affected to seakeeping qulity of Semi-SWATH vessel.

Since the non-linear mode both of heave and pitching responses are considered, Time Domain Simulation approach is utilized to solve the equations of motion. One of the important aspects of the present simulation program is the calculation and evaluation of seakeeping performance in step of time inherent to the bare hull vessel and after present an additional damping due to two pairs of fin stabilizers on the Semi-SWATH hull in waves.

A simulation procedure had been followed to conduct parameter study in systematic way. The main purpose of the parametric study is to compare motion characteristics of different conditions i.e. bare hull condition and after installed fixed and controllable fin stabilizers at two operational draughts (shallow water and deep water) with different service speeds i.e. 15 knots (medium speed) and 20 knots (high-speed) by utilizing the Time Domain Simulation approach.

In case on the simulation of controllable aft fin stabilizers, the conventional PID controller will be applied to refine the plant system. For that reason, the vessel will experience a better ride performance with more improvement of reduction of heave and pitch motion compared to uncontrolled fin. In addition, the simulation and experiments results obtained through the parametric study become a guideline prediction of vessels' comfortability. Simply, the study of the seakeeping evaluation of Semi-SWATH vessel was divided into four sections, as follow;

1. The first involved the development of the mathematical model for simulation purposes. The fin angles with fixed and controllable fin inclined relative to the main hull body have to be adequate to have a significant effect on the seakeeping performance of Semi-SWATH vessel during sailing.
2. The second investigate of the controller scheme of the controllable aft fin stabilizer on the vessel performances.
3. The third involved the development of PID controller that applied on the controllable aft fins to obtain a better ride performance of Semi-SWATH vessel passengers comfort, which give a more significant reduction of heave and pitch motions than using fixed fins.
4. The fourth, the experimental work of seakeeping study will be conducted to validate the derived mathematical model that has been conducted in PC base simulation program. In this case, the validation is only to justify the simulation condition on bare hull vessel and using fixed fin relative to the main hull body only.

8.2 Mathematical model

In the present study, the main interest is given on the ship motion in head seas condition. Basically, the modular mathematical model has been chosen for this study. The motion system of the vessel consists two-degrees-of-freedom i.e. coupled heave and pitch motions. The mathematical model comprising of heave and pitch motions, which incorporated with fixed and controllable fins stabilizers on the simulation was presented in a simple block diagram using Matlab-SIMULINK.

A computation procedure to calculate the forces up to free surface has been applied to obtain the hydrostatic forces including restoring forces and Froude-Krylov forces and lift forces due to fin stabilizers. This computation procedure successfully simulates heave and pitch motion responses both of bare vessel and after attached fixed and controllable fins.

For the hydrodynamic forces, it is determined through the empirical formulations. However, the mathematical model requires coefficients such as added mass and damping and restoring moment. These coefficients are obtained directly through the published literature or experiment. Frank-Close Fit method (Atlar, 1982) and Maimun (1993) experimental results are the two important references to estimate the coefficients. This study showed that ship motion is sensitive to the change of pitch restoring. Therefore, pitch restoring force should be accurately determined in thesimulation run.

8.3 Investigation of Controller Scheme of The Fin Stabilizers

In this investigation, the controller scheme of fin stabilizer was applied into two rules, namely;

a) Fin with constant inclined angle relative the main hull body

   This case means that the stabilizer fin is adjusted to have an inclined angle relative to the ship body. The total lift on the ship is then due to the resultant of angle of attack of incident flow to the body and the inclined angle of fin. This technique can be applied to adjust the trim condition of Semi-SWATH vessel advancing in waves with initial trim by bow or stern.

In this work, the aft and fore fin with fixed angle inclined relative to the main hull body. This case is achieved by adjusting manually the stabilizer fin to be with a constant angle of fins relative to the main hull body while the Semi-SWATH vessel running in waves. The gradual adjustable angle of the aft fin will result the heave and pitch motion decreases equals to their positive inclined angles of fixed fore fin (15^0) and fixed aft fin of 5^0, 10^0, 15^0, 20^0, and 25^0 relative to the main hull. The maximum reduction of heave and pitch motion are about 40.14% and 37.95% (Figure 7.35 and Figure 7.44).

b) Time-Varying Controllable Fin

The purpose of this technique is to obtain more improvement of reducing heave and pitch motions of the Semi-SWATH vessel advancing in waves. The principle is to adjust the fin lift force to balance the force due wave excitation on the Semi-SWATH vessel.

In this work, by controlling the aft fins properly, the sailing style of Semi-SWATH vessel may be more effective than uncontrolled one. The aft fins control (δ_{aft}) were set by 0^0 up to 25^0 and fixed fore fins (δ_{fore}) is set by 15^0. The aft fins are controlled which the all fixed operated fins angle speeds of 22.5^0/sec. We find that the quick operated speed affects the pitch motion faster reduced. This control scheme is quite reasonable because the system response is more stable. In this case, the controllable aft fins can be operated more secure and smoothly and a gradual response is achieved too. Practically, the aft fins are controlled simultaneously to achieve a better operation condition.

8.4 Development of PID Controller

The present study concentrates on the solution of ship motion reduction especially for heave and pitch motions in head-seas condition due to controllable aft fin stabilizers. A conventional PID Controller (Proportional, Integral and Derivative) has been applied control plant systems by actuating the aft fin stabilizer automatically. This is mainly due to its simple design, straightforward parameters' tuning, and robust performance. In the present work the PID Controller will control the actuators (servomotors) as a position control. Jingzhuo, et al (2000), Habib and Maki (2001), Yang, et al (2002), and Yusuf, et al (2003) discussed he position controls utilizing PID Controller. To design an effective PID controller, three gain parameters need to be specified properly namely;

1. Proportional gain (K_p),
2. Integral gain (K_i) and,
3. Derivative gain (K_d).

The conventional approach to determine the PID parameters is to study the mathematical model of the process and try to come up with a simple tuning law that provides a fixed set of gain parameters. One famous example of such approach is the Ziegler-Nichols method.

8.5 Experimental Result

From the Figure 7.65 to 7.78, Figure 7.15 to 7.64 and Figure 7.79 to 7.128, it is found that the model of fin stabilizers have successfully developed experimentally with obtaining satisfactory results. It was shown by some reductions of heave and pitch motions. This indicated that the fixed fore and fin stabilizers give the dynamic effects as an additional damping and restoring forces to the vessels. In addition, the fin stabilizer plays an important role in improvement of heave and pitch motion responses. As consequence, the phenomena of some serious inconveniences in term passenger comfortability and safety can be alleviated.

8.6 Concluding Remarks

As mentioned earlier, the drawback of twin-hull vessel performances at high-speed in waves is closely related to large amplitude of vertical motions (heave and pitch motions). For that reason, the installation of fixed fin and controllable fins are the most appropriate way to alleviate the vertical motions and enhance the safety and the passenger comfortability. In the analysis has shown that additional damping and pitch restoring forces have has been known to have significant effect on reducing heaving and pitching responses.

Additionally, the series of studies of fixed and controllable fins on the Semi-SWATH vessel advancing in waves have been developed. The procedure applied in this study successfully solves the problem of twin-hull vessel travelling in head waves theoretically. In this simulation, the Semi-SWATH vessel's performance has involved a non-linearity of vertical motions. Hence, a Time-Domain Simulation approach is the most appropriate method used in this situation.

In case of controllable aft fins, the simulation has been solved by applying the PID Controller to the servo motor as the actuator to obtain a better ride performance of Semi-SWATH vessel in waves. According to the present study, it was found that by using PID control procedure, the heave and pitch motion are indeed reduced effectively and the Semi-SWATH vessel has better ride performance as compared to use fixed fins.

Finally, the seakeeping test experimentally will be conducted to validate the simulation program in case of bare hull vessel and using fixed fore fins both adjustable aft fins manually. The validation results showed that the simulation program utilized by Time-Domain Simulation has been developed successfully.

CHAPTER 9

CONCLUSSION

The series of seakeeping evaluation using Time-Domain Simulation and experimental model tests for Semi-SWATH vessel without and with attached fixed fore fins and controllable aft fins advancing in regular waves have been developed successfully. Although only parts of the results are shown here, the following conclusion can be drawn:

1. The results of the model tests at condition T = 2.0 m and Vs = 15 knots with fixed fin angle (Aft = 20^0 and fore = 15^0) are obtained a maximum reduction of heave by about; 39.18% and with fixed fin angle (Aft = 15^0 and fore = 15^0), the reduction of pitch motions by about 37.95% as compared to bare hull vessel.

2. According to the present control study at condition; T = 2.0 m and Vs = 15 knots, the application of the PID controller on the controllable aft fins (saturation of fin angle is 0^0 – 25^0), the significant reductions of heave and pitch motions are achieved using an automatic controllable aft fins by about; 42.57% and 48.98% as compared to the uncontrolled fins.

3. Furthermore, at condition; T = 1.4 m, Vs = 20 knots, the application of the PID controller on the controllable aft fins (saturation of fin angle is 0^0 – 25^0), the maximum percentages of heave and pitch motion reductions are by about 26.775% and 25.771% as compared to the uncontrolled fins.

CHAPTER 10

FUTURE RESEARCH

1. It will good basis for naval architects to design a suitable controllable fin for a Semi-SWATH vessel in irregular seas.

2. Although the present control technique here is conventional PID controller theory shows a good results regarded as useful for basic controller reference results. In fact, the present control theory can be incorporated with a modern control theories, e.g. fuzzy control, ANN control, LQG control or adaptive control, etc., to make the fin control even more effective and robust. In other words, the analytical tool developed here may offer very useful information to the Semi-SWATH vessel designers in case automatic controllable fins.

APPENDIX A

THEORETICAL ANALYSIS OF ADDED MASS AND DAMPING

A. 1. Single Degree-of-Freedom

$$\left(m_j + a_{jj}\right)\ddot{\eta}_{j(t)} + b_{jj}\dot{\eta}_{j(t)} + c_{jj}\eta_{j(t)} = f_{j(t)} \quad \text{(A.1)}$$

Displacement, $\eta_{j(t)} = \eta_{j0}.\cos\left(\omega t - \varepsilon_j\right)$ (A. 2)

Velocity, $\dot{\eta}_{j(t)} = -\eta_{j0}.\omega.\sin\left(\omega t - \varepsilon_j\right)$ (A. 3)

Acceleration, $\ddot{\eta}_{j(t)} = -\eta_{j0}.\omega^2.\cos\left(\omega t - \varepsilon_j\right)$ (A. 4)

Exciting force, $f_{j(t)} = f_{j0}.\cos\left(\omega t\right)$ (A. 5)

Where:

m_j = Mass of the system

a_{ij} = Hydrodynamic reaction in phase with acceleration (added mass)

b_{ij} = Hydrodynamic reaction in phase with velocity (damping)

c_{ij} = Stiffness (restoring)

f_{j0} = Amplitude of motion

ε_j = Phase angle

Substituting $\eta_{j(t)}$, $\dot{\eta}_{j(t)}$ and $\ddot{\eta}_{j(t)}$ in (a.1), yields,

$$\left(m_j+a_{ij}\right)\left(\eta_{j0}.\omega^2.\cos\left(\omega t-\varepsilon_j\right)\right) - b_{jj}\left(\eta_{j0}.\omega.\sin\left(\omega t-\varepsilon_j\right)\right) +$$

$$c_{jj}\left(\eta_{j0}.\cos\left(\omega t-\varepsilon_j\right)\right) = f_{j0}.\cos\omega t \qquad \text{(A. 6)}$$

Expanding (a. 2)

$$\begin{aligned}&\left(m_j+a_{ij}\right)\left(\eta_{j0}.\omega^2\right)\left(\cos\omega t\ .\ \cos\varepsilon_j+\sin\omega t\ .\ \sin\varepsilon_j\right)\\&-b_{jj}\left(\eta_{j0}.\omega\right)\left(\sin\omega t\ .\ \cos\varepsilon_j - \cos\omega t\ .\ \sin\varepsilon_j\right)\\&+c_{jj}\left(\eta_{j0}\right)\left(\cos\omega t\ .\ \cos\varepsilon_j + \sin\omega t\ .\ \sin\varepsilon_j\right) = f_{j0}.\cos\omega t\end{aligned} \qquad \text{(A. 7)}$$

Equating the $\left(\cos\omega t\right)$ terms in (a. 7);

$$-\left(m_j+a_{ij}\right).\eta_{j0}.\omega^2.\cos\varepsilon_j + b_{jj}.\eta_{j0}.\omega.\sin\varepsilon_j + c_{jj}.\eta_{j0}.\cos\varepsilon_j = f_{j0} \qquad \text{(a. 8)}$$

Equating the $\left(\sin\omega t\right)$ terms in (a. 7);

$$-\left(m_j+a_{ij}\right).\eta_{j0}.\omega^2.\sin\varepsilon_j + b_{jj}.\eta_{j0}.\omega.\cos\varepsilon_j + c_{jj}.\eta_{j0}.\sin\varepsilon_j = 0 \qquad \text{(A. 9)}$$

From equations (a. 8) and (a. 9), the two unknowns a_{ij} and b_{jj} can be found. Therefore,

$$a_{jj} = \frac{1}{\omega^2}\left(c_{jj} - \left(\frac{f_{j0}.\cos\varepsilon_j}{\eta_{j0}}\right)\right) - m_j \qquad \text{(A. 10)}$$

$$b_{jj} = \left(\frac{f_{j0}}{\eta_{j0}.\omega}\right).\sin\varepsilon_j \qquad \text{(A. 11)}$$

A. 2. Two Degrees-of-Freedom

For a two degrees-of-freedom system with forcing in the j^{th} direction, the resulting coupled motions in the j^{th} and k^{th} directions can be described by the following equations of motion:

$$\left(m_j+a_{jj}\right)\ddot{\eta}_{j(t)}+b_{jj}\dot{\eta}_{j(t)}+c_{jj}\eta_{j(t)}+a_{jk}\ddot{\eta}_{k(t)}+b_{jk}\dot{\eta}_{k(t)}$$

$$+c_{jk}\eta_{k(t)}=f_{jj(t)} \qquad \text{(A. 12)}$$

$$\left(m_k+a_{kk}\right)\ddot{\eta}_{k(t)}+b_{kk}\dot{\eta}_{k(t)}+c_{kk}\eta_{k(t)}+a_{kj}\ddot{\eta}_{j(t)}+b_{kj}\dot{\eta}_{j(t)}$$

$$+c_{kj}\eta_{j(t)}=0 \qquad \text{(A. 13)}$$

Displacement, $\eta_{j(t)}=\eta_{j0}.\cos\left(\omega t-\varepsilon_j\right)$ (A. 14)

Velocity, $\dot{\eta}_{j(t)}=-\eta_{j0}.\omega.\sin\left(\omega t-\varepsilon_j\right)$ (A. 15)

Acceleration, $\ddot{\eta}_{j(t)}=-\eta_{j0}.\omega^2.\cos\left(\omega t-\varepsilon_j\right)$ (A. 16)

Exciting force, $f_{jj(t)}=f_{j0}.\cos\left(\omega t\right)$ (A. 17)

Displacement, $\eta_{k(t)}=\eta_{k0}.\cos\left(\omega t-\varepsilon_k\right)$ (A. 18)

Velocity, $\dot{\eta}_{k(t)}=-\eta_{k0}.\omega.\sin\left(\omega t-\varepsilon_k\right)$ (A. 19)

Acceleration, $\ddot{\eta}_{k(t)}=-\eta_{k0}.\omega^2.\cos\left(\omega t-\varepsilon_k\right)$ (A. 20)

Where:

m_k, m_j = Mass of body in the k^{th} and j^{th} directions.

a_{kk}, a_{jj} = Hydrodynamic reaction in phase with acceleration (added mass) in the k^{th} and j^{th} directions.

b_{kk}, b_{jj} = Hydrodynamic reaction in phase with velocity (damping) in the k^{th} and j^{th} directions.

c_{kk}, c_{jj} = Hydrostatic stiffness of body in the k^{th} and j^{th} directions.

a_{kj}, a_{jk} = Hydrodynamic reaction in the kth and jth directions but in phase with acceleration in the jth and kth direction (coupled added mass).

b_{kj}, b_{jk} = Hydrodynamic reaction in the kth and jth directions but in phase with velocity in the jth and kth direction (coupled damping)..

c_{kj}, c_{jk} = Hydrodynamic stiffness in the direction due to displacement in the kth and jth directions due to displacement in the jth and kth direction (coupled stiffness).

$\varepsilon_k, \varepsilon_j$ = Phase angle of motion in in the kth and jth direction.

Substituting $\eta_{j(t)}$, $\dot{\eta}_{j(t)}$, $\ddot{\eta}_{j(t)}$ and $\eta_{k(t)}$, $\dot{\eta}_{k(t)}$, $\ddot{\eta}_{k(t)}$ in (A.13), yields,

$$\begin{aligned}&\left(m_k+a_{kk}\right)\left(-\eta_{k0}.\omega^2.\cos\left(\omega t-\varepsilon_k\right)\right)+b_{kk}\left(-\eta_{k0}.\omega.\sin\left(\omega t-\varepsilon_k\right)\right)+\\&c_{kk}\left(\eta_{k0}.\cos\left(\omega t-\varepsilon_k\right)\right)+a_{kj}\left(-\eta_{j0}.\omega^2.\cos\left(\omega t-\varepsilon_j\right)\right)+\\&b_{kj}\left(-\eta_{j0}.\omega.\sin\left(\omega t-\varepsilon_j\right)\right)+c_{kj}\left(\eta_{j0}.\cos\left(\omega t-\varepsilon_j\right)\right)=0\end{aligned} \qquad \text{(A. 21)}$$

Expanding (A. 21)

$$\begin{aligned}&\left(m_k+a_{kk}\right)\left(\eta_{k0}.\omega^2\right)\left(\cos\omega t\,.\,\cos\varepsilon_k+\sin\omega t\,.\,\sin\varepsilon_k\right)\\&-b_{kk}\left(\eta_{k0}.\omega\right)\left(\sin\omega t\,.\,\cos\varepsilon_k-\cos\omega t\,.\,\sin\varepsilon_k\right)\\&+c_{kk}\left(\eta_{k0}\right)\left(\cos\omega t\,.\,\cos\varepsilon_k+\sin\omega t\,.\,\sin\varepsilon_k\right)\\&-a_{kj}\left(\eta_{j0}.\omega^2\right)\left(\cos\omega t\,.\,\cos\varepsilon_j+\sin\omega t\,.\,\sin\varepsilon_j\right)\\&-b_{kj}\left(\eta_{j0}.\omega\right)\left(\sin\omega t\,.\,\cos\varepsilon_j-\cos\omega t\,.\,\sin\varepsilon_j\right)\\&+c_{kj}\left(\eta_{j0}\right)\left(\cos\omega t\,.\,\cos\varepsilon_j+\sin\omega t\,.\,\sin\varepsilon_j\right)=0\end{aligned} \qquad \text{(A. 22)}$$

Equating the $(\cos\omega t)$ terms in (A. 22);

$$\begin{aligned}&\left(m_k+a_{kk}\right)\left(\eta_{k0}.\omega^2\right)\cos\varepsilon_k+b_{kk}\left(\eta_{k0}.\omega\right)\sin\varepsilon_k\\&+c_{kk}\left(\eta_{k0}\right)\cos\varepsilon_k-a_{kj}\left(\eta_{j0}.\omega^2\right)\cos\varepsilon_j+b_{kj}\left(\eta_{j0}.\omega\right)\sin\varepsilon_j\\&+c_{kj}\left(\eta_{j0}\right)\cos\varepsilon_j=0\end{aligned} \qquad \text{(A. 23)}$$

Equating the $(\sin\omega t)$ terms in (A. 22);

$$\left(m_k + a_{kk}\right)\left(\eta_{k0}.\omega^2\right)\sin\varepsilon_k - b_{kk}\left(\eta_{k0}.\omega\right)\sin\varepsilon_k$$
$$+ c_{kk}\left(\eta_{k0}\right)\sin\varepsilon_k - a_{kj}\left(\eta_{j0}.\omega^2\right)\sin\varepsilon_j - b_{kj}\left(\eta_{j0}.\omega\right)\cos\varepsilon_j \qquad (A.\ 24)$$
$$+ c_{kj}\left(\eta_{j0}\right)\sin\varepsilon_j = 0$$

The coefficients of a_{kk} and b_{kk} can be obtained from single-degree-of-freedom experiment. Coupled coefficient c_{kj} and phase angle ε_j and ε_k can be obtained from the two-degrees-of-freedom experiment.

APPENDIX B

MATLAB ODE

1. ode45

Ode45 is based on an explicit Runge-Kutta (4,5) formula, the Dormand-prince pair. It is, a one-step solver; that is, in computing y(tn), it needs only the solution at the immediately preceding time point, y(tn-1). In general, ode45 is the best solver to apply as a first try for most problems. For this reason, ode45 is the default solver used by Simulink for models with continuous states.

2. ode23

ode23 is also based on an explicit Runge-Kutta (2,3) pair of Bogacki and Shampine. It can be more efficient than ode45 at crude tolerances and in the presence of mild stiffness. Ode23 is a one-step solver.

3. ode113

ode113 is a variable order Adams-Bashforth-Moulton PECE solver. It may be more efficient than ode45 at stringent tolerances and when the ODE file function is particularly expensive to evaluated, ode113 is a multistep solver – it normally needs the solutions at several preceding time points to compute the current solution.

4. ode15s

ode15s is a variable order solver based on the numerical differentiation formulas (NDFs). These are related to but are more efficient than the backward, BDFs (also known as a Gear's method). Like ode113, ode15s is a multistep method solver. If you suspect that the problem is stiff, or if ode45 failed or was very inefficient, try ode15s.

5. ode23s

ode23s is based on modified Rosenbrock formula of order 2. Because it is one-step solver, it can be more efficient than ode15s at crude tolerances. It can solve some kinds of stiff problems for which ode 15s are not effective.

6. ode23t

ode23t is an implementation of the trapezoidal rule using a"free" interpolant. Use this solver if the problem is only moderately stiff and you need a solution without numerical damping.

7. ode23tb

ode23tb is an implementation of the TR-BDF2, an implicit Runge-Kutta Merson formula with a first stage that is trapezoidal rule step and a second stage that is a backward differentiation formula of order two. By construction, the same iteration matrix is used in evaluating both stages. Like ode23s, this solver can be more efficient than ode15s at crude tolerances.

APPENDIX C

SEMI-SWATH VESSEL PARTICULARS

L_{OA}	23.11	m
B_{OA}	8.0	m
B_{Hull}	1.6	m
Displacement	77.429	Tones (Full load)
Displacement	53.461	Tones (Light load)
Block Coefficient	0.666	
Midship Area Coefficient	0.886	
Waterplane Area Coefficient	0.652	

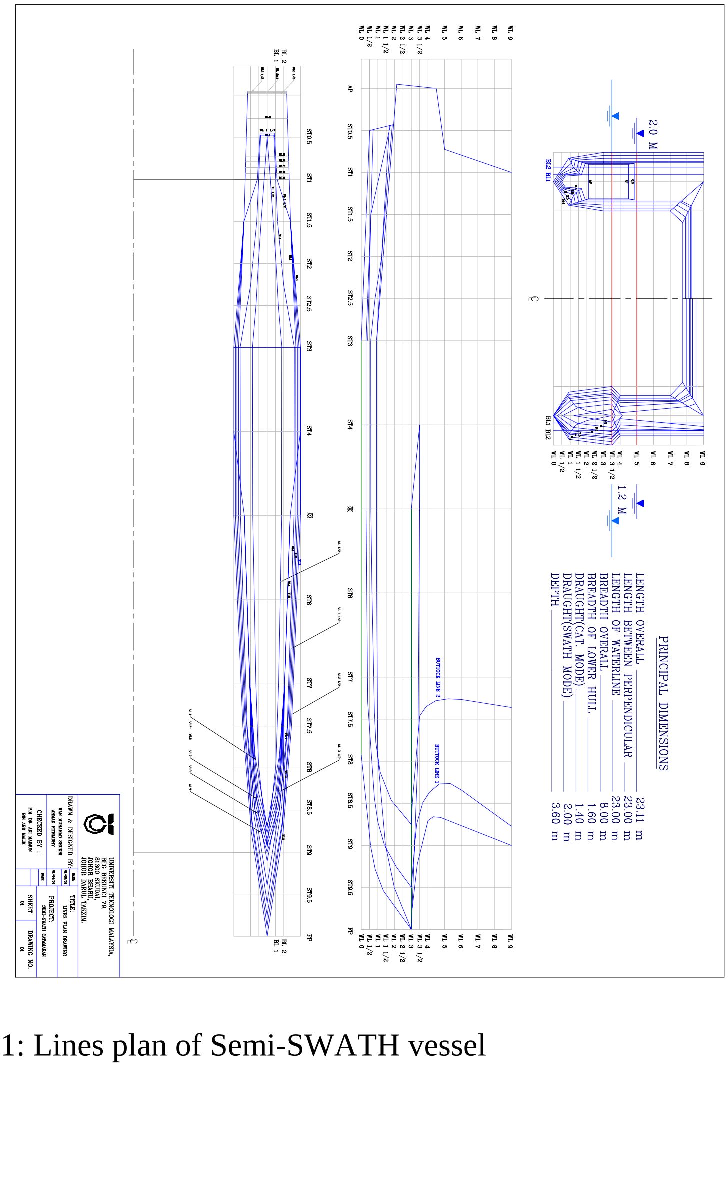


Figure 1: Lines plan of Semi-SWATH vessel

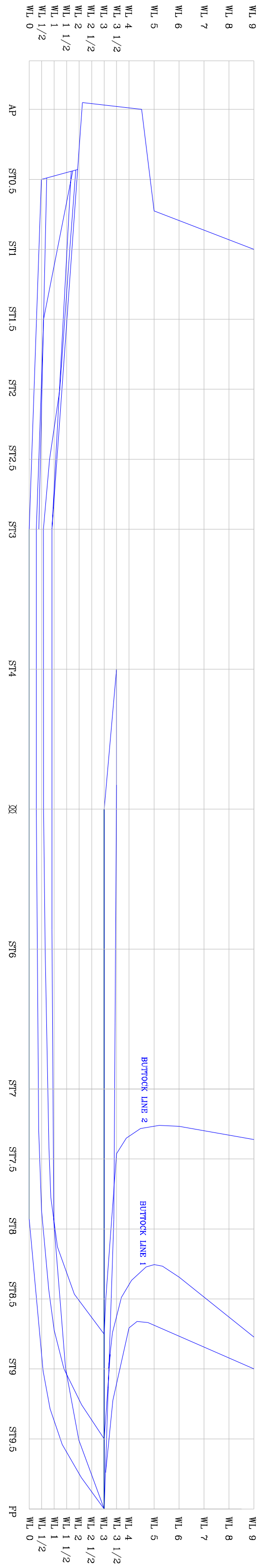


Figure 2: Side view of Semi-SWATH vessel

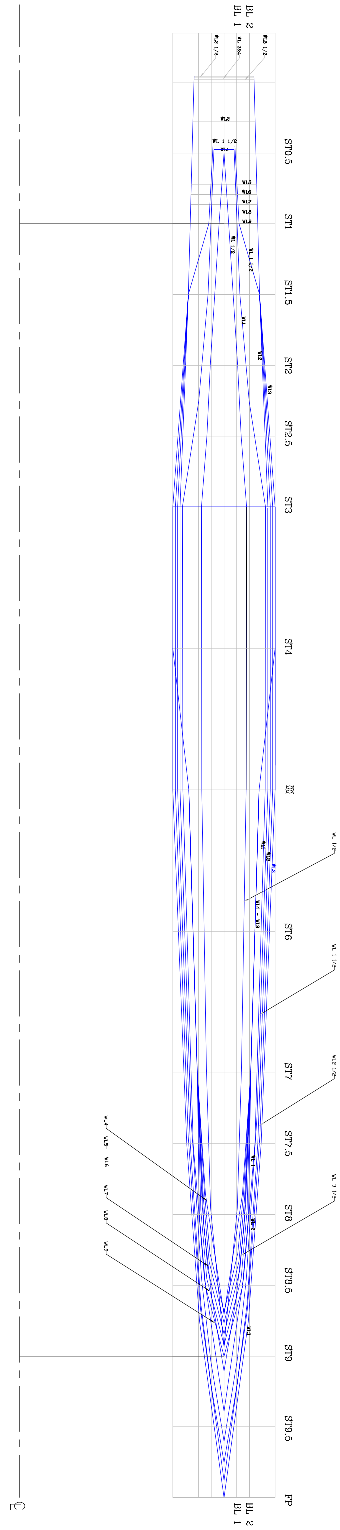


Figure 3: Top view of Semi-SWATH vessel

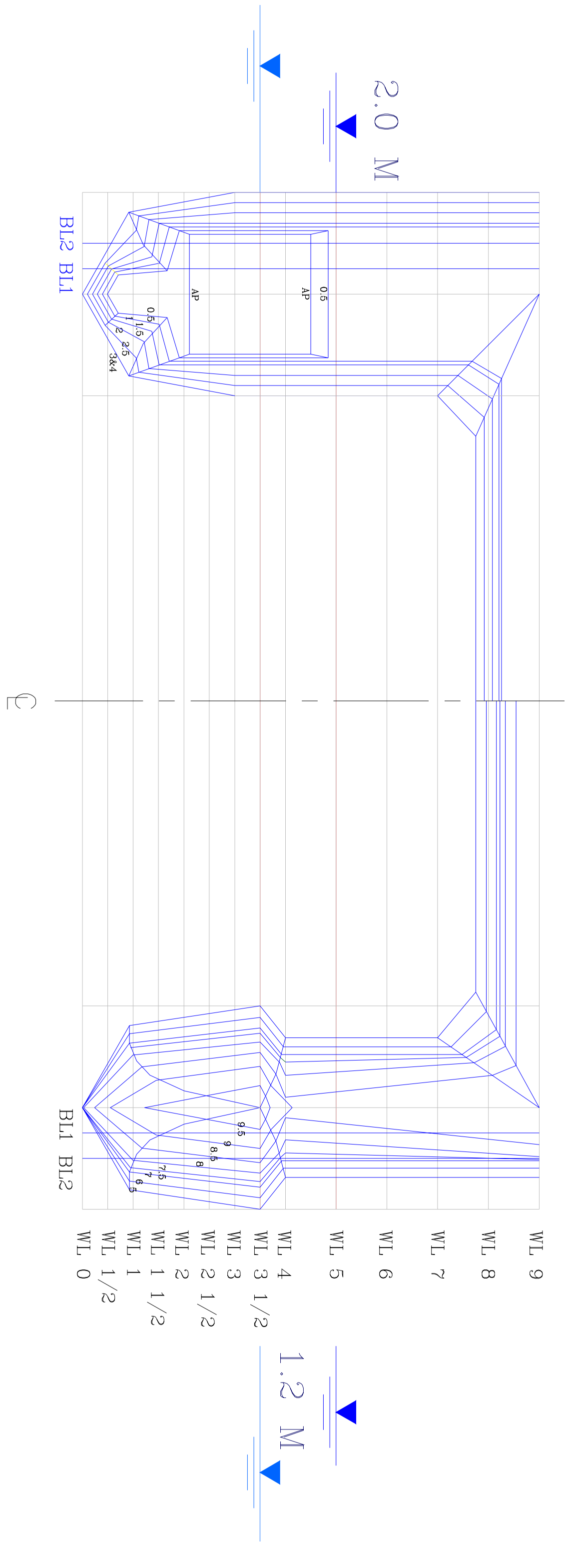


Figure 4: Front view of Semi-SWATH vessel

APPENDIX D

FIN STABILIZER DATA

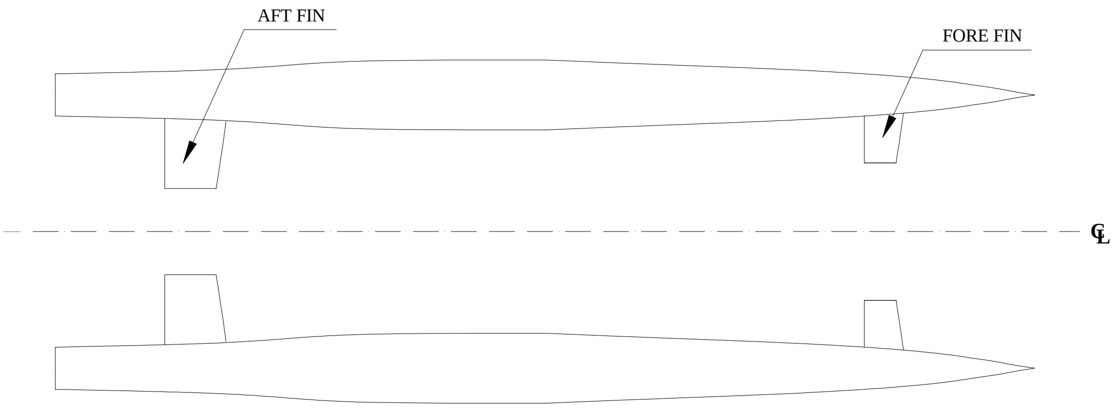


Figure 1: Plane view of aft and fore fin stabilizer

Type of Fin Stabilizer : NACA0015

The principle dimension of stabilizer fins

Description	Fore Fin	Aft Fin
c, Chord (m)	0.96	1.45
s, Span (m)	1.20	1.86
Location* (m)	3.50	19.5
Depth** for deep draught (m)	1.51	1.51
Depth** for shallow draught (m)	0.92	0.92
Maximum Thickness (m)	0.15	0.23

* Distance from the main hull nose to the quarter-chord point

** Distance from the waterline to the chord line

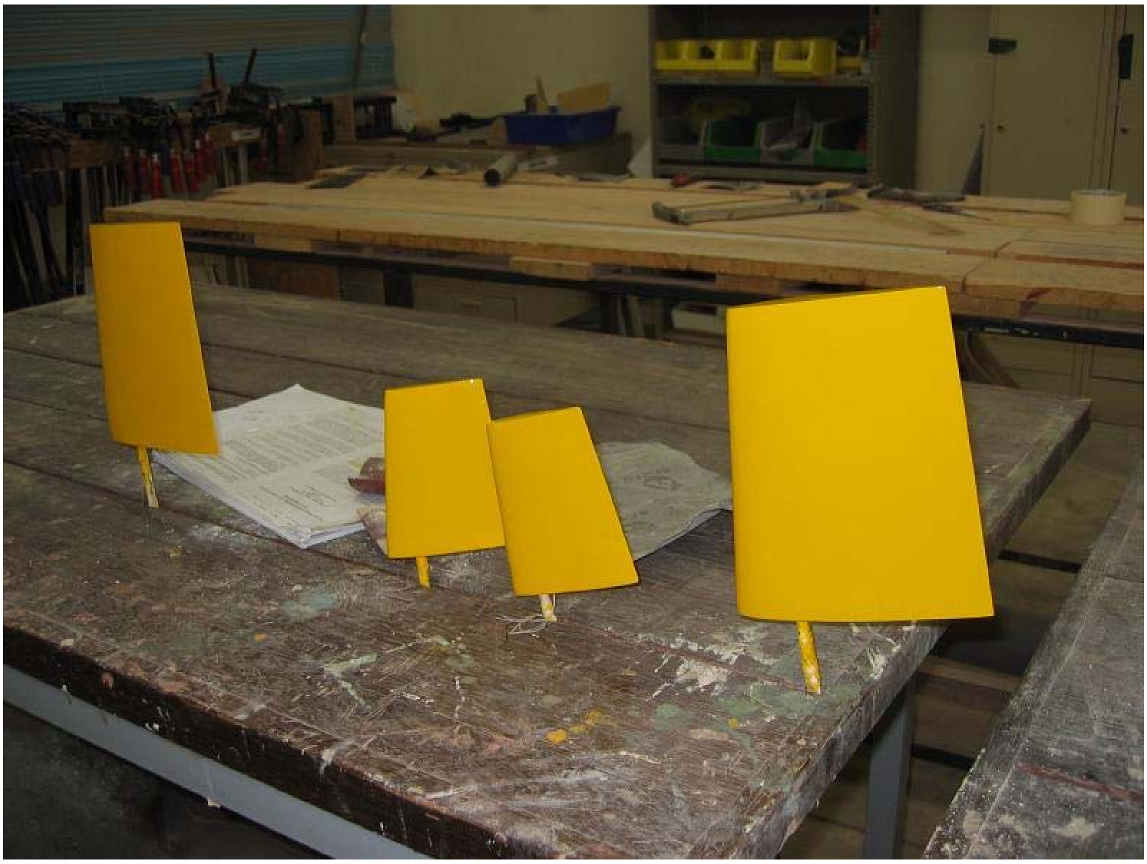
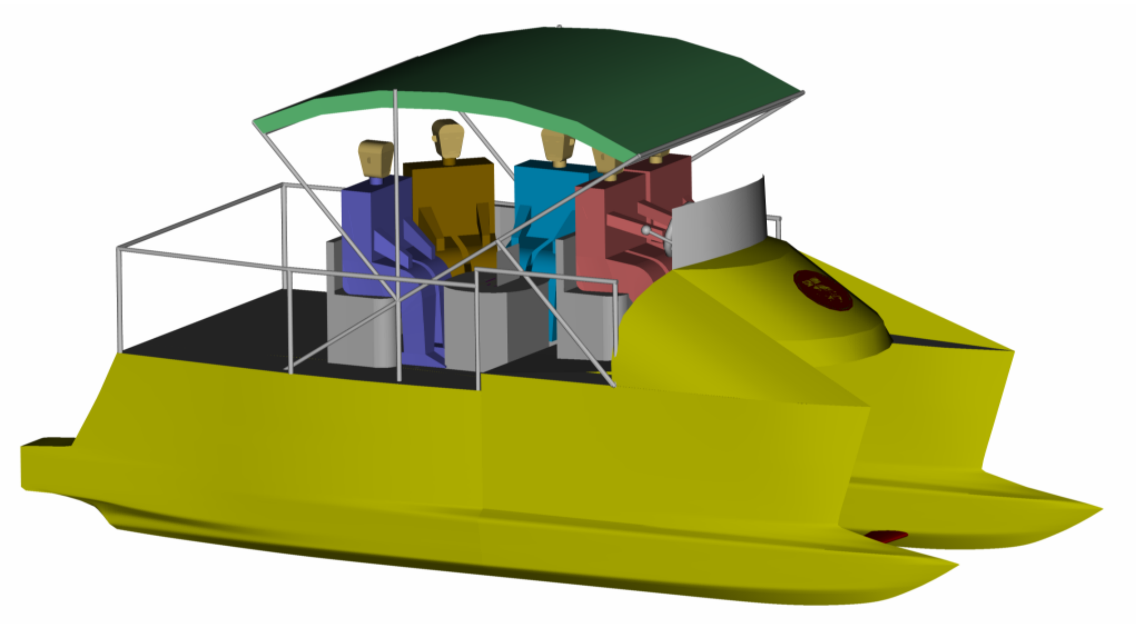


Figure 2: Photo of aft and fore fin stabilizer model

APPENDIX E

PROTOTYPE OF SEMI-SWATH VESSEL



LOA	6.0	m
BOA	2.2	m
B_{Hull}	0.42	m
Displacement	1.355	Tones (Full load)
Displacement	0.935	Tones (Light load)
Block Coefficient	0.666	
Waterplane Area Coefficient	0.652	

Figure 1: 6.0m length of Semi-SWATH prototype

APPENDIX F

EXPERIMENTAL PHOTOGRAPHS

Photo 1: Side view of the model during the bifilar test

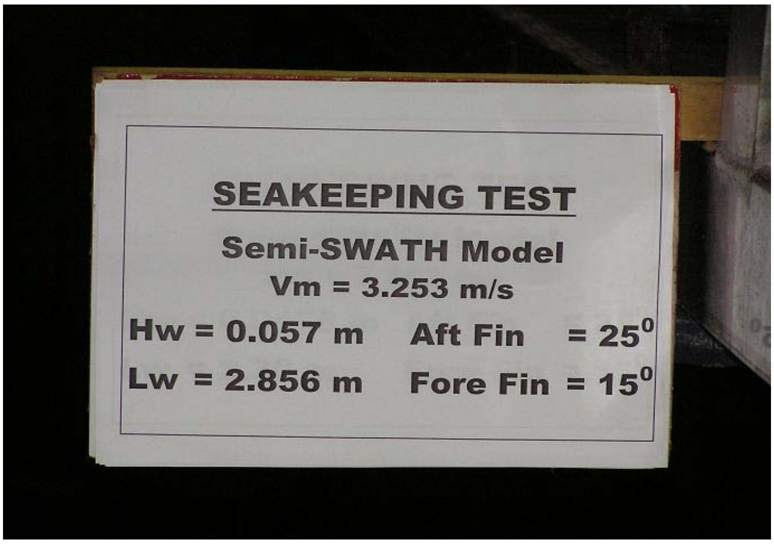


Photo 2: Sign board of seakeeping test for Semi-SWATH vessel

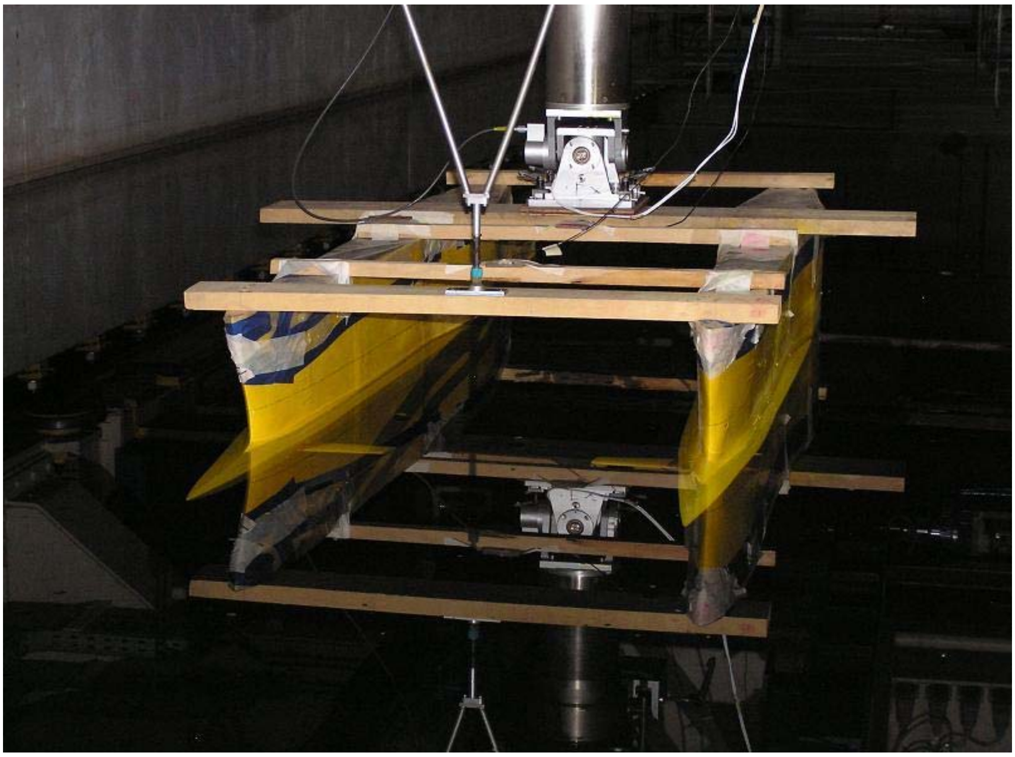


Photo 3: The model is locked to Gimbals

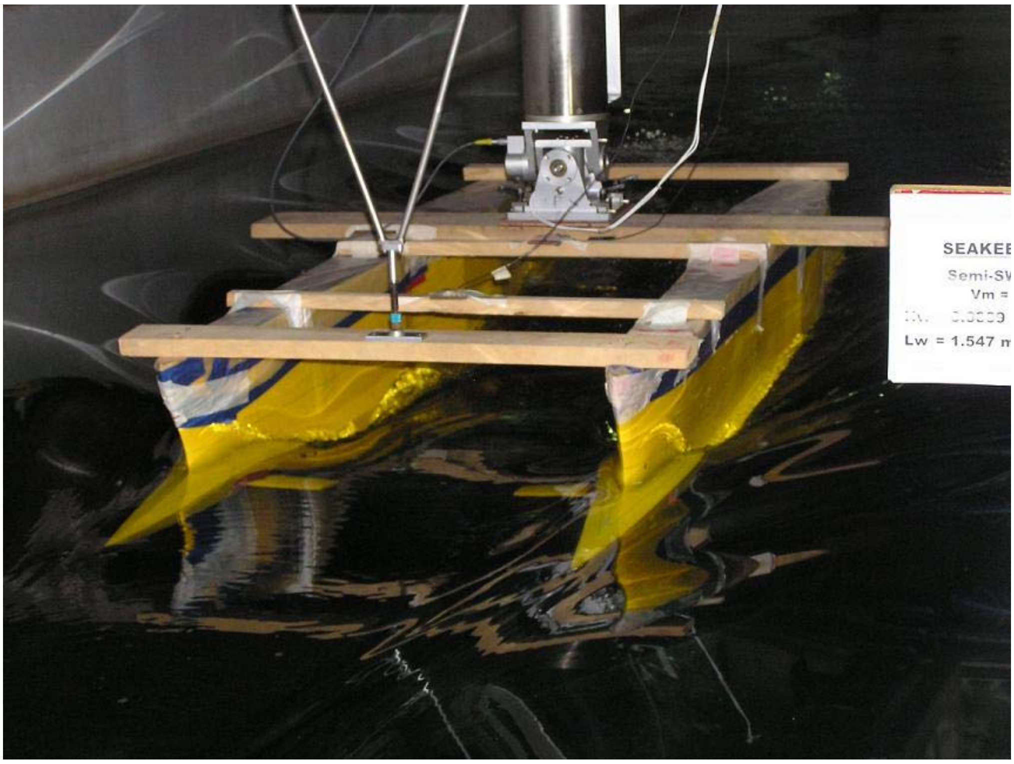


Photo 4: The model is towed

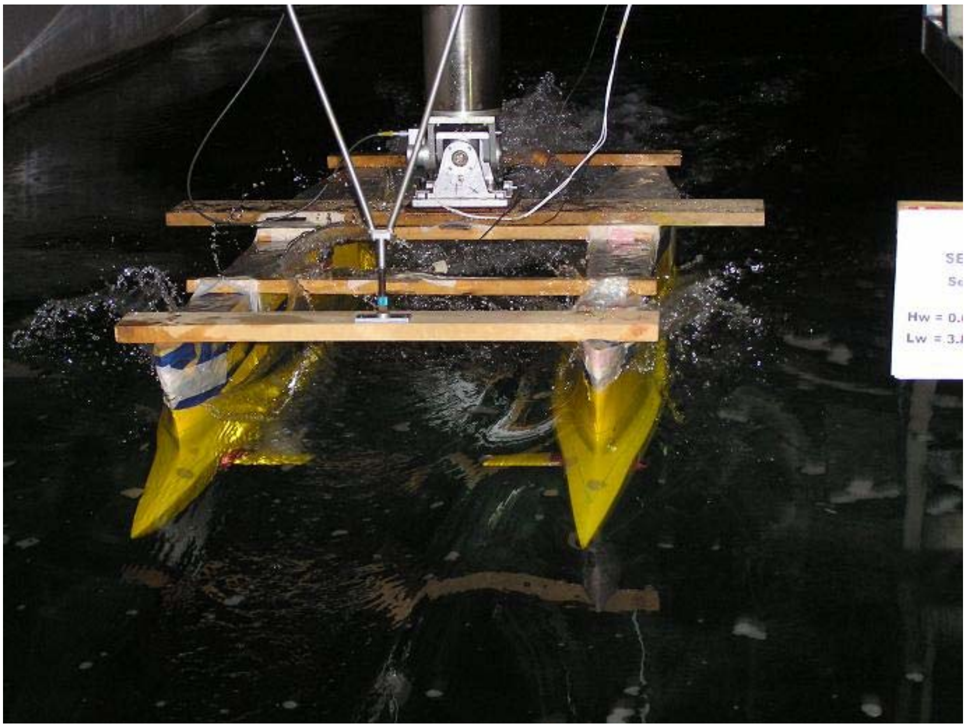


Photo 5: The water comes up to the main deck of model during seakeeping test

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BORANG PENGESAHAN
LAPORAN AKHIR PENYELIDIKAN

TAJUK PROJEK : **DEVELOPMENT OF A SEMI-SWATH CRAFT FOR MALAYSIAN WATERS**

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