

PERFORMANCE PREDICTION FOR HIGH SPEED CRAFT

ANUAR BIN BERO

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Dedicated with love to my wife and children

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ABSTRACT

Generally the performance of the high speed craft can be divided into six main components such as resistance and powering, propulsion, dynamic instability, seakeeping and manoeuvring. Performance prediction on high speed craft especially in planing hull is complicated due to complex combination of ship behaviour in rough sea condition. The performance of high speed craft is becoming more important due to their various functions and purposes to marine community which is unable to be predicted using conventional methods. The fundamental of this research is to study the two main components of the vessel i.e. resistance and seakeeping quality by incorporating stern foil at aft portion planing craft (M Hull) that gives significant effect to the performance of the vessel. Theoretically, stern foil has a similar function with transom flap, trim wedges and trim tab which is to reduce the resistance and also as a damping for motion reduction. In the scope of resistance performance for this vessel with stern foil, the Savitsky and two dimensional Methods are used for resistance prediction at different angle of attack. While Computational Methods i.e. SEAKEEPER program was applied to seakeeping prediction in regular wave (head sea). Both result of resistance and seakeeping performance prediction was validated by conducting model test for the model with and without stern foil. The performance of ship model with stern foil gives a positive performance in term of seakeeping quality at constructive resistance. By adapting with stern foil the heave and pitch Response Amplitude Operator (RAO) trim down by 4.0% and 18.91% respectively. Furthermore, the reduction of forward and aft acceleration RAO also occurs concurrently which the decreasing of both acceleration are 21.10% and 6.14%.

ABSTRAK

Secara amnya, prestasi kapal laju dibahagikan kepada enam komponen utama iaitu rintangan dan daya tujahan, dorongan, ketidakstabilan dinamik, *seakeeping* dan *manoeuvring*. Anggaran terhadap prestasi kapal laju terutamanya *planing hull* adalah sangat sukar disebabkan gandingan sifat kapal yang kompleks pada keadaan laut yang bergelora. Kajian ini lebih menumpukan kepada dua perkara iaitu rintangan dan kualiti *seakeeping* pada kapal laju berbentuk M Hull yang dipasang dengan foil buritan. Secara teori, foil buritan mempunyai fungsi yang sama dengan kepek buritan, baji buritan dan *trim tab* yang mana berpotensi bagi mengurangkan rintangan dan juga sebagai peredam untuk meminimumkan pergerakan kapal. Kaedah anggaran Savitsky dan dua dimensi telah diaplikasi bagi mengira prestasi rintangan kapal yang mempunyai sudut pesongan yang berbeza. Sementara program simulasi SEAKEEPER pula digunakan dalam anggaran sifat kapal seperti *heave*, *pitch*, pecutan haluan dan buritan pada keadaan ombak yang seragam. Hasil keputusan secara teori bagi pengiraan rintangan dan simulasi *seakeeping* dibandingkan dengan keputusan data ujian rintangan dan ujian *seakeeping* untuk mengesahkan prestasi kapal dengan foil buritan atau sebaliknya. Ini dibuktikan secara eksperimen, dengan memasang foil buritan prestasi kapal laju dapat ditingkatkan yang mana pengurangan *heave* RAO sebanyak 4% dan *pitch* RAO 18.91%. Malahan pecutan haluan dan buritan juga berkurang, masing-masing menunjukkan prestasi dapat ditingkatkan sehingga 21.10% dan 6.14%.

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NOMENCLATURE

L_{OA}	-	Length Overall (m)
L_{WL}	-	Length waterline (m)
B_{oa}	-	Breadth overall (m)
B_{WL}	-	Breadth at waterline (m)
T	-	Moulded draft (m)
Δ	-	Displacement (tone)
∇	-	Volume (m^3)
V	-	Ship speed (m/s)
LCB	-	Longitudinal Centre of Buoyancy (m)
LCG	-	Longitudinal Centre of Gravity, from transom (m)
B/T	-	Breadth draught ratio
L/B	-	Length breadth ratio
$L/\nabla^{1/3}$	-	Length-displacement ratio
g	-	Specific gravity ($9.81m/s^2$)
ρ	-	Mass density, ($1025 kg/m^3$)
ν	-	Kinematic velocity, m^2/s
R_n	-	Renault number $\frac{VL}{\nu}$
F_n	-	Froude number V/\sqrt{gL}
S	-	Wetted surface area
R_T	-	Total resistance (N)
R_F	-	Friction resistance according to the ITTC-1957 friction formula (N)
R_R	-	Residual resistance (N)
P_E	-	Effective power (kW)
$1 + k_1$	-	Form factor the viscous resistance of the hull form in relation to R_F
C_F	-	Coefficient of friction
C_R	-	Residuary resistance coefficient

C_A	-	Ship model-ship correlation
C_T	-	Total resistance coefficient
i_E	-	The angle measured in the plane of the water plane, between the hull and the centerline (deg)
β	-	Deadrise angle (deg)
C_B	-	Block coefficient
C_{WP}	-	Waterplane area coefficient
R_W	-	wave resistance
C_M	-	Midships coefficient
C_P	-	Prismatic coefficient
C_v	-	Speed Coefficient $\frac{V}{\sqrt{gB_{WL}}}$
τ	-	Trim angle (deg)
C_{L_o}	-	Flat plate lift coefficient
$C_{L\beta}$	-	The lift coefficient for finite deadrise
λ	-	Wetted length beam ratio
λ_k	-	Wetted length of the keel (m)
W	-	Vessel Displacement (N)
R_{nb}	-	Renault Number based on B_{WL}
ΔC_F	-	Correction factor which obtained from ATTC Standard Roughness
I_w	-	Linearized Integral
w_i	-	Vertical Inflow Velocity
ϕ	-	Velocity Potential
Γ	-	Circulation
D	-	Foil Drag
R_v	-	Viscous resistance Foil or Strut (N)
C_{Dv}	-	Viscous resistance coefficient Foil or Strut
L_v	-	Lift Force due to Viscous
C_{Lv}	-	Lift coefficient due to Viscous
C_F	-	Friction coefficient
R_{nc}	-	Reynolds number Chord Based
t/c	-	Foil thickness to chord ratio
V	-	Ship speed (m/s)
C	-	Chord length (m)

ν	-	Kinematic viscosity (m^2/s)
A	-	Planform area Foil or Strut (m)
R_i	-	Induced resistance Foil or Strut (N)
L_i	-	Lifting Force (induced) Foil or Strut
C_{Di}	-	Induced resistance coefficient
C_{Li}	-	Lift coefficient
$\frac{A}{c_0}$	-	Aspect ratio, $\left(\frac{4s}{\pi c_0}\right)$
α	-	Angle of attack (radian)
c_0	-	Chord length at midspan (m)
A	-	Planform area (projected area of the elliptical foil), $\left(\frac{s^2}{\lambda}\right)$ (m^2)
s	-	Span length (m)
$A(\theta)$	-	Complex wave amplitude function
B	-	Parabolic strut
C_{Lw}	-	Lift coefficient (wave)
R_w	-	Wave resistance Foil (N)
C_{Dw}	-	Wave resistance coefficient
C_{Lw}	-	Lift coefficient
f	-	Maximum camber (m)
R_s	-	Spray Resistance
ΣF	-	The sum of various fluid forces (vertical hydrodynamic forces as well as the wave excitation force)
ΣM	-	The sum of corresponding moments acting on the vessel because of relative motion of vessel and wave.
\ddot{z}, \dot{z}, z	-	Heave acceleration, velocity and displacement, respectively.
$\ddot{\theta}, \dot{\theta}, \theta$	-	Pitch angular acceleration, velocity and displacement respectively.
Δ/g	-	Mass of vessel.
I_{yy}	-	Pitch inertia moment of vessel.
$a_{zz}, b_{zz}, c_{zz}, a_{z\theta}, b_{z\theta}, c_{z\theta}, a_{\theta\theta}, b_{\theta\theta}, c_{\theta\theta}, a_{\theta z}, b_{\theta z}, c_{\theta z}$	-	Stability derivatives
ΔF	-	Foil excited force
ΔM	-	Foil excited moment
ζ	-	Amplitude of heaving (m)
α	-	Maximum slope of the surface wave
ω	-	Frequency of the surface

ε	-	Phase angle between exciting moment and wave elevation
ω_0	-	Wave frequency
Λ	-	Tuning factor = ω / ω_0
M	-	Magnification factor = $1 / \{ (1 - \Lambda^2)^2 + 4k^2 \Lambda^2 \}^{1/2}$
$\delta\Delta$	-	Added mass moment of Inertia
k_{xx}^2	-	Radius of gyration
ε_1	-	phase angle between exciting moment and wave elevation

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CHAPTER 1

INTRODUCTION

1.1 Background of Study

Generally, the performance of high speed craft is difficult to obtain due to several factors that shall be considered by designer such as resistance and powering, propulsion, dynamic instability, seakeeping and manoeuvring criteria. Normally, all these considerations are not fully achieved due to low budget and the owner has to cut cost. Another factor that contributes to the failure of performance of high speed craft is many of the assumptions used either with numerical or experimental techniques. The formulation of conventional vessel is not suitable for predicting the performance of high speed craft especially after several modifications has been conducted on their hullform.

High speed crafts are known to have rough water problem is essentially one of compromise between speed and seakeeping performance. As the speed of vessels increases, the resistance also increase and required more power to move. At high speed regime, the seakeeping becomes more important especially for passengers vessel and vessel fit in with high technology equipment. However, speed is the main factor and followed by comfort condition (seakeeping quality) to be considered during preliminary design of this vessel and that factor must go well with rough sea condition in order to achieve the mission or task within time frame.

In this study will discuss in detail the performance prediction of high speed craft in term of resistance and seakeeping quality for the high speed craft (planing

craft M-hull) before and after incorporating with stern foil. The reason of this adapting of a stern flap foil is to combine the seakeeping qualities of the vessel with the dynamic effect and higher speed attainable at favourable ship resistance.

1.2 Objective

1. To investigate the effect of stern foil on resistance and seakeeping of M-Hull Planing Craft.

1.3 Scope of Work

1. Literature review on stern foil analysis of M-Hull Planing Craft.
2. To develop a computer program for resistance prediction of M-Hull Planing Craft by using Savitsky and two dimensional methods with effect of stern foil.
3. To perform seakeeping analysis by using an existing computational software Maxsurf SEAKEEPER.
4. To conduct resistance and seakeeping tests with and without stern foil.

1.4 Schedule of the Project

1.4.1 Project I

1. Literature review on resistance and seakeeping behaviour of high speed craft. The study shall begin by determining the characteristic of the parameters of high speed craft in high speed region. The study also expands on the effect of tool for

controlling motion in waves which gives a significant effect to the speed of the vessel.

2. The work will be continued with collecting all data and ships particulars including hydrostatic data, drawing and materials for appropriate vessel which is related to research objectives.

3. Perform the theoretical calculation and introduce the Savitsky equation and develop the foil and strut formulation in FORTRAN programming to predict the resistance of effect of stern flap foil on research vessel.

4. Conduct seakeeping simulation by using SEAKEEPER programming in order to predict the motions by effect of stern flap foil.

1.4.2 Project II

1. A model will be constructed at Marine Technology Laboratory, Universiti Teknologi Malaysia UTM.

2. Model test shall be conducted in order to assess the theory of performance of high speed craft against the results from model test. Basically the purposes of this experiment are:

a. To determine the resistance of the vessel with and without stern foil at speed of 25 knots (12.86m/s).

b. To determine the significant effect of motions (heave and pitch) in head sea at design speed with and without stern foil.

c. To confirm that by adapting stern flap foil at transom stern to the motion of the vessel will decrease at vertical acceleration.

3. To perform the performance comparison for research ship between the results of model test and theoretical estimates.

4. The details of methodology being simplified is illustrated in the figure 1.1 (project flow chart) while the detail planning chart for Project I and Project II is in Appendix A.

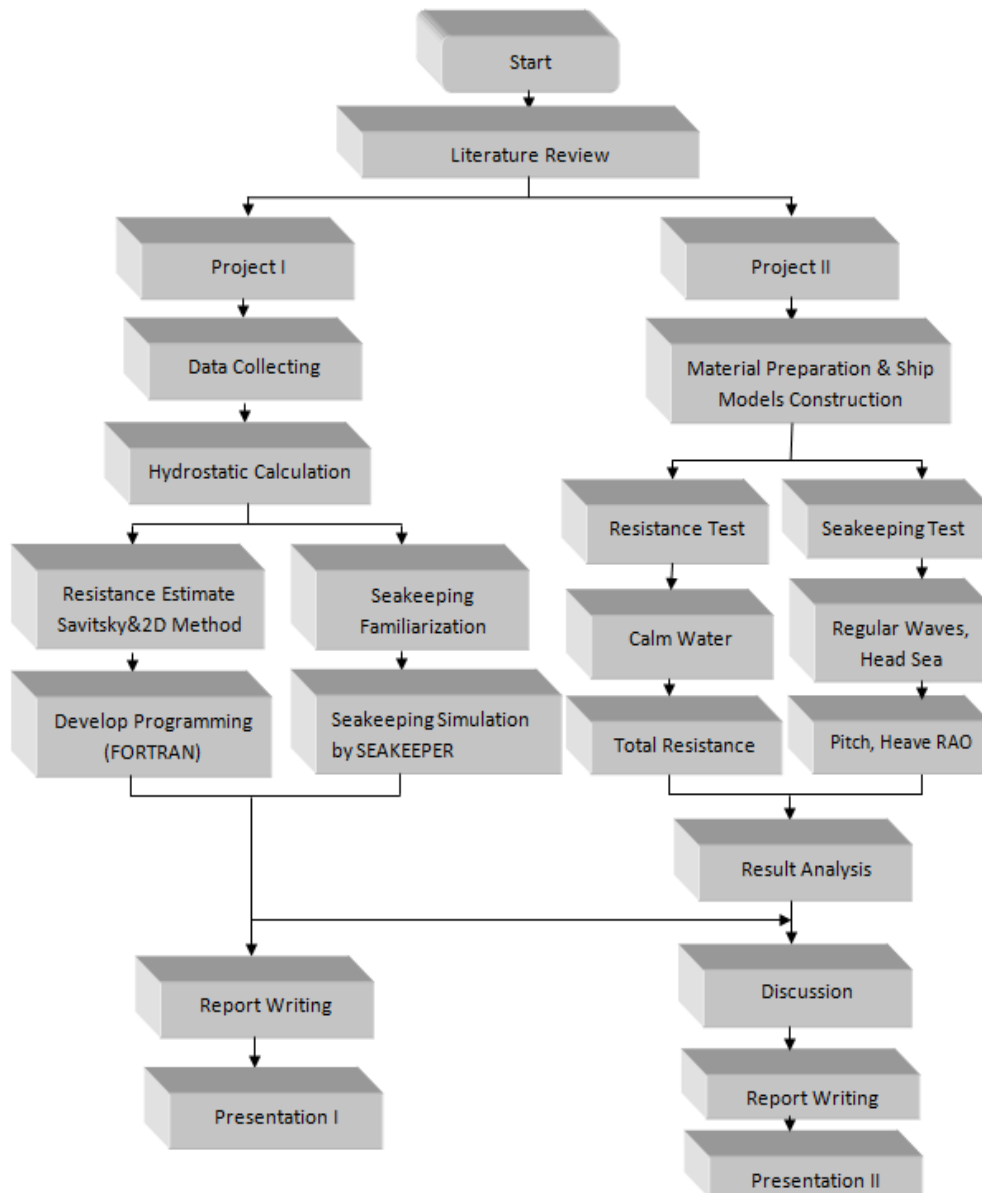


Figure 1.1 : Project Flowchart

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

A high-speed craft (HSC) is a high speed vessel for civilian use, also called a fast craft or fast ferry and is called patrol craft for military purposes. A vast increase in high speed crafts due to existing needs in the field of fast transport of light and expensive cargo, passengers at high speed craft for marine transportation has drawn considerable interest for both shipowners and naval architectures. The function of high speed also gives advantages to ships which are design to be used for a surveillance and patrol in maritime area at open sea. Advanced concept was applied in many types of high speed craft in order to obtain a great performance in seaway. The design and safety of high-speed craft is regulated by the High Speed Craft Codes of 1994 and 2000, adopted by the Maritime Safety Committee of the International Maritime Organisation (IMO).

Clayton et al [1] defined the high speed craft when the craft speeds are reached for which hydrostatic force less or equal to pressure force, $F_h \ll F_p$, but some part must be submerged. Therefore wetted length, $l_k \neq 0$ and also $F_h \neq 0$. However this definition is hard to improve which Newton [2] stated that it is generally recognized that the definition of high speed when the R_T/W curve level off and stern trim has reached maximum. Another definition was identified by Mandel [3] that the craft to be high speed when it ceases to be buoyantly supported and becomes completely dynamically supported. Clement and Pope [4] found this occurred when volume Froude number, $F_r \nabla \geq 3.5$.

At different perspective, Savitsky et al [5] defined that high speed craft are considered to be vessels that can travel at a sustained speed equal to or greater than 35 knots with bursts of high speeds of 40-60 knots. Froude number allows for another way to hydrodynamically classify ships. Naval architects use the Froude number when vessels deal with the interaction of the water's free surface and the hull. High speed vessels are typically defined with $F_n \nabla > 0.4$ which at this speed range the crafts weight is almost entirely supported by dynamic forces and the trim tend to be much lower than hump as shown in figure 2.1.

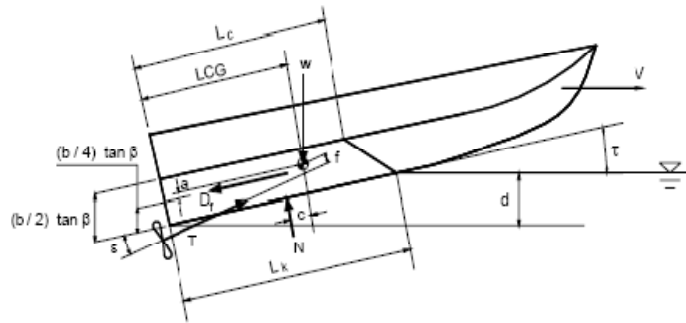


Figure 2.1 : Forces acting on a Planing Hull

International Maritime Organization (IMO) [6] defined the high speed craft is a craft capable of maximum speed in metres per second (m/s) equal or exceeding:

$$V \geq 3.7 \nabla^{0.1667} \quad (2.1)$$

Symbol of ∇ is a volume of displacement corresponding to design waterline (m^3) excluding craft the hull of which is supported completely clear above the water surface in non-displacement made by aerodynamic forces generated by ground effect such as Wing in Ground.

High speed craft can be classed in two broad categories, Air-Supported and Displacement type. Air supported craft include Air Cushion Vehicles (ACV), Surface-Effect Ships (SES) and Foil Supported craft such as hydrofoils and jetfoils. Displacement type vessels include conventional monohull, catamaran, trimaran, small waterplane-area twin-hull (SWATH), and air lubricated hulls. While each type

of vessel has its own unique characteristics, they all suffer from the common problem of limited payload and a sensitivity to wind and sea state.

2.2 Planing Craft

Planing craft are used as patrol boats, sport vessels, service craft and for sport competitions. When a vessel is planing, it is mainly supported by hydrodynamic forces. A length Froude number of 1-1.2 is often used as a lower limit for planing conditions. There are many important dynamic stability problems associated with planing vessels such as porpoising stability. Strongly nonlinear phenomena will appear during planing including spray jet, breaking waves etc. Therefore it is hard to apply conventional linear theories for displacement vessels to study planing hulls. In order to accurately predict the hydrodynamic behavior of a planing vessel, non-linear effects must be included in the analysis. The characteristic of this craft was defined by Froude and operates Froude number (F_n) larger than about 0.4. Generally, the buoyancy force dominates relative to the hydrodynamic force effect when F_n is less than approximately 0.4. When $F_n > 1.0$, the hydrodynamic force mainly carries the weight, then the vessel turn into a planing craft.

A planing hulls are hullforms characterized by relatively flat bottoms and shallow V-sections (especially forward of amidships) that produce partial to nearly full dynamic support for light displacement vessel and small craft at higher speeds. These types of hull form lift and skim the surface of the water causing the stern wake to break clean from the transom. The crafts are generally restricted in size and displacement because of the required power-to weight ratio and the structural stresses associated with traveling at high speed in wave. Most planing hull crafts are also restricted to operate in reasonably calm water, although some “deep-V” hull forms are capable operated in rough water. As mentioned in [7], in general there are three types of planing hulls with different characteristics and advantages. They are namely;

1. Deep Vee Bottom.
2. Inverted Vee Bottom.
3. Round Bottom.

In Deep Vee Bottom type, single hard chine is most frequently used. This form has the advantages of being a substantially good planing surface form, simple and economical to produce and having excellent accommodation space for machinery, armament, and crew. It has a disadvantage of having a greater wetted surface with consequent greater resistance. Its characteristics in a seaway compared to other planing hull forms are only fair. On the contrary, planing hull forms are poor in rough water [8].

Numerous researches in planing hull seakeeping technology have quantified the relations between hull form, loading, speed/length ratio, sea state and the expected added resistance, motions and wave impact accelerations. The process of optimization of hull form for achieving the 'best' hull shape is shown in reference [9]. The shapes with the best performance at sea, are in terms of hydrodynamic behaviors, structural robustness, transport efficiency, operational and economic advantage.

2.2.1 Geometry of Planing Craft

The simplest geometry of planing surface is a flat one which illustrated in figure 2.2. The principal feature of the flow is the rise in the water level ahead of the line of intersection between the undisturbed water surface and the plane. Consequently, the dynamic wetted-surface length (L) is greater than the submerged length (L_i). Both of these lengths are different from the stationary length (L_o). The leading edge of the wetted surface is nominally defined to coincide with the location of the spray-root line. The slight curvature that this line possesses (when projected onto the plane) is usually ignored in any calculations.

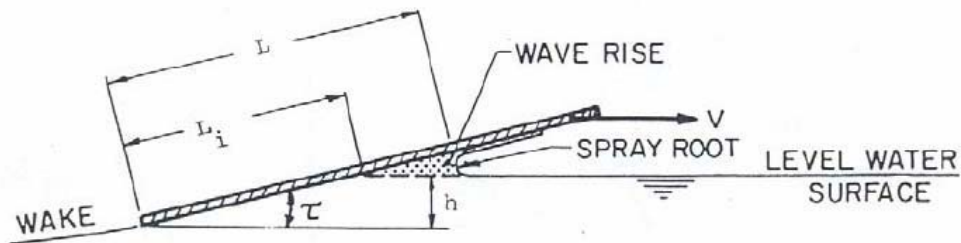


Figure 2.2 : The Simplest Geometry of Planing Surface [10]

The plane is wetted ahead of the spray-root line. This is more clearly indicated in figure 2.2. However, it can be shown that the thickness of the spray is closely proportional to the square of the trim angle τ , for the usual range of trims of interest. Since this is generally quite small, one can ignore the lift of the pressure distribution on the plane ahead of the spray-root region. The pressure distribution is seen to have its stagnation value near the spray root. It falls off to zero at the trailing edge of the plate.

Figure 2.3 [10], shows the main features of a prismatic planing surface. This difference from the flat surface in that a deadrise angle is included in the description of its shape. The use of deadrise results in a decrease in the lifting capacity of the surface. However, a deadrise surface has two main advantages; less motion in waves and better directional stability. Experiments show that there is almost no build-up or water rise under the keel. Thus, contrary to the case of the flat planing surface, the spray root starts at the intersection of the undisturbed water surface and the keel. This is referred to as point 0 in figure 2.2.

However, a rise in water level away from the keel. As noted in figure 2.3, the dynamic half-wetted $(\pi b_1)/2$, where b_1 is the half-wetted beam computed on the basis of the intersection of the undisturbed water surface with the hull. This factor of $\pi/2$ is an outcome of a simple two-dimensional theory which assumes that the body is slender. It has been experimentally verified for prismatic planing surfaces with various aspect ratios and deadrise angles and operating at different Froude numbers.

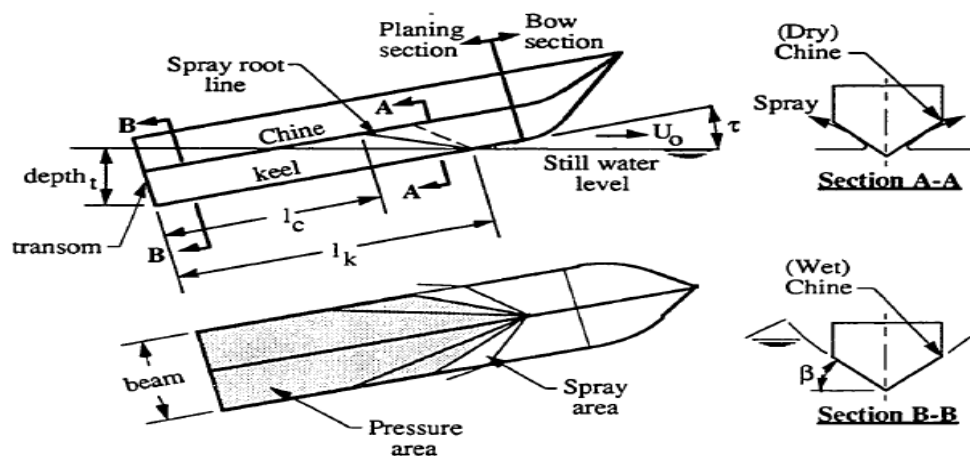


Figure 2.3 : The Main Features of a Prismatic Planing Surface [5]

2.3 Behaviour of Planing Craft

2.3.1 Calm Water

The steady behaviour of planing craft on straight course in calm water is a function of trim moment, vertical force and longitudinal force on the hull depend on the trim angle, draft and speed. The pressure can be divided into hydrodynamic and hydrostatic pressures. The hydrodynamic can be described by potential flow and by neglecting gravity.

The lift of planing craft can be obtained by buoyant forces which the craft as a displacement hull at low speed. When speed increase to speed coefficient $Cv = \frac{v}{\sqrt{gB_T}} \cong 0.50$ [10], there appears the first visual evidence of the dynamic effects upon the flow patterns. Complete ventilation of the transom occurs and appears to be independent of deadrise, trim or hulls length for typical values of these parameters.

The dynamic effects produce a positive contribution lift at speed coefficient ranges between 0.5 and 1.5 although in many cases was not sufficient to result in a significant rise of the centre of gravity or emergence of the bow. The flow has only slightly separated from the forward length of the chine and there has significant side

wetting. The planing craft should develop dynamic lift forces when the speed coefficient larger than 1.5. The lift forces give a significant rise of the centre of the gravity, positive trim and emergence of the bow and separation of the flow from the hard chines. The hydrodynamic resistance is due to the horizontal component of the bottom pressure force and the friction component of the flow over the bottom which is there is no bow contribution to drag.

The lift force is approximately proportional to the trim angle. If the craft has hard chines, the separation lines along the hull are well defined along the chines. Calculation can be made by neglecting the effect of the viscous boundary layer on the pressure distribution. But this assumption not applicable to the vessel with round bilges, whereas the separation lines may then be dependent on laminar or turbulent flow conditions in the boundary layer.

According to Savitsky [10], the lift on the planing surface is attributed to two separated effects. One is the positive dynamic reaction of the fluid against the moving planing bottom, and the second one is the so-called buoyant contribution which is associated with the static pressures corresponding to a given draft and hull trim. At very low speeds, the buoyant lift predominates, while at high speed, the dynamic contribution predominates. The lift coefficient, C_L is a function of mean wetted length beam ratio, λ .

For a flat planing surface, $\beta = 0^\circ$, the lift coefficient is:

$$C_{L0} = \tau^{1.1} \left(0.0120\sqrt{\lambda} + \frac{0.0055\lambda^{\frac{5}{2}}}{C_v^2} \right) \quad (2.2)$$

Where C_{L0} can be obtain by solving equation (2.3) and (2.4)

$$C_{L\beta} = C_{L0} - 0.0065\beta C_{L0}^{0.6} \quad (2.3)$$

$$C_{L\beta} = \frac{\rho g \nabla}{1/2 \rho V^2 b^2} \quad (2.4)$$

Then, any lifting surface or flat plate in a free stream, as well as planing surface or planing hull, is subjected to lift force:

$$L = \frac{1}{2} C_L \rho V^2 S \quad (2.5)$$

While other parameters being constant, the hydrodynamic lift varies as the square of the beam. The planing lift is predominately due to dynamic bottom pressures when the speed coefficient and Froude number defined above is greater than 10. The effect of deadrise angle is to reduce the lift coefficient, while all other factor being equal.

2.3.2 Rough Water

The occurrence of waves has considerable importance on the design of planing craft and this factor must be included in the prediction of seakeeping at initial stage of design. The hull form is dependant on the expected wave encounter spectra, or perhaps the worst wave spectra envisaged, since the new problem must include a measure of seakeeping ability, so that hull accelerations and response amplitudes can be determined. It is well known that for planing craft the flat-water/rough water problem is essentially one of a compromise between speed and seakeeping.

A planing craft running into waves can be considered to be a coupled 3 degree of freedom (heave, pitch and surge) dynamic system wherein the hull is acted upon by a forcing function generated by the waves it encounters. The dynamic properties of the hull are represented by the acceleration forces due to its physical and added mass and moments of inertia; the force and moment acting on the hull due to a unit displacement in heave or pitch (the so-called spring constants of the hull); and the force and moment acting on the hull due to its heave or pitch velocity (the so-called damping forces). The forcing function acting on the hull at a given speed is due to the interaction of the encountered wave properties (height, orbital velocities, accelerations, etc.) and the physical geometry of the hull. As expected, this is a

complex non-linear mathematical system even without the force and moment inputs due to an active control system.

The behaviour of a ship in rough water is fundamentally similarly to the oscillatory response of the classical damped spring mass system illustrated in figure 2.4. The classical spring mass system consists of a mass a (tonnes) which is connected to a fixed rigid base through a dashpot and a spring. The dashpot exerts a damping force b (kN) in response to a velocity of 1 metre/second and the spring exerts a restoring force c (kN) if the displacement is 1 metre. If the system is not disturbed it will adopt an equilibrium position which we shall define as a datum displacement, $x=0$ (m).

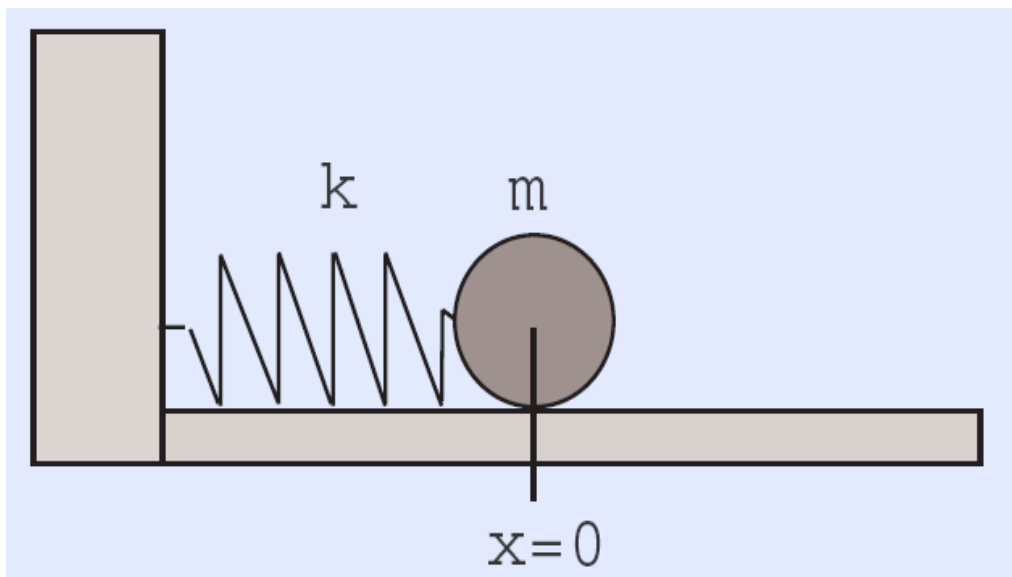


Figure 2.4 : Spring Mass System [11]

The total force F was applied to the mass at any instant of time which related to the motion by the equation:

$$a \frac{d^2x}{dt^2} + b \frac{dx}{dt} + cx = F \quad (2.6)$$

The first term is the force acting on the system due to acceleration of the physical mass, a and the added mass; the second term is the force due to damping of

the system taken to be linearly dependent upon the normal velocity of the hull and the damping coefficient, b , the third term is the force due to displacement where c is the effective spring constant of the hull. The term on the right hand side is the force F assumed to be acting on the system. The natural frequency ω_n of the system can be simplified by neglecting the damping which gives a small effect to the system. The natural frequency can be expressed as:

$$\omega_n = \sqrt{\frac{c}{a}} \quad (2.7)$$

For the steady condition the amplitude response can be expressed in non dimensional form as:

$$x_0 = x_{st}\mu \quad (2.8)$$

Where:

μ = Magnification factor

$$x_{st} = \frac{F_0}{c}$$

F_0 = Force acting on body

$$\mu = \frac{x_0}{x_{st}}$$

$$= \frac{1}{\sqrt{(1-A^2)^2 + 4K^2A^2}} \quad (2.9)$$

A = Tuning factor, $\frac{\omega}{\omega_n}$

K = Non dimensional damping factor, $\frac{b}{2\sqrt{ca}}$

Thus the phase response can be expressed in non dimensional form as:

$$\varepsilon = \tan^{-1} \frac{2KA}{1-A^2} \quad (2.10)$$

The amplitude of the response of a linear spring mass system is show in the form of a classic plot that readily quantifies the response of a dynamic system to a

sinusoidal forcing function. The plot of amplification factor versus ratio of applied frequency of forcing function /natural frequency of the dynamic system as a function of the damping ratio of the system is shown in figure 2.5.

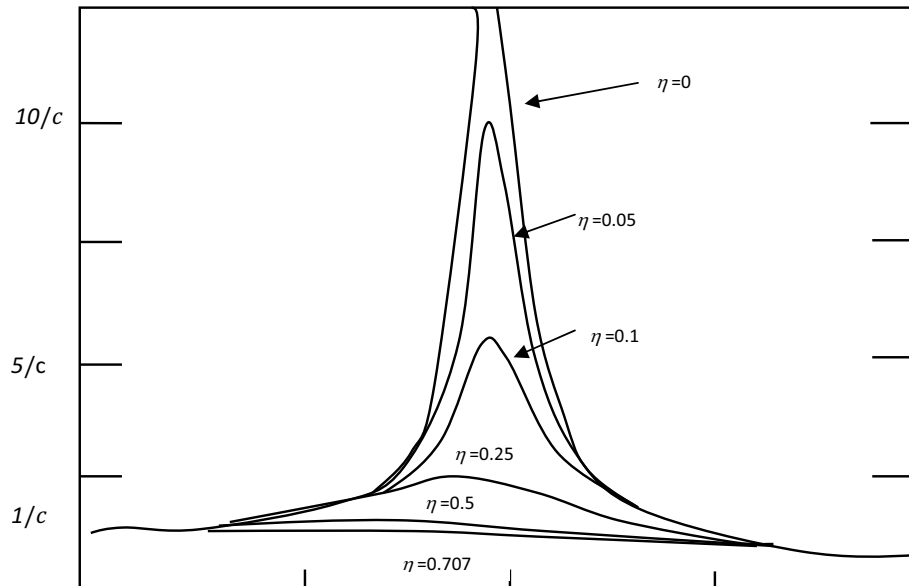


Figure 2.5 : Response of a Linear Spring Mass System [11]

In terms of ship motion terminology, the magnification factor can be likened to the response amplitude operator (RAO), i.e. considering heave motion; the RAO is the ratio of the magnitude of heave of the craft to the height of the wave that caused this heave motion. Likewise this RAO can also represent the ratio of craft pitch amplitude to the slope of the wave that caused the craft to pitch. There is a separate curve for different values of damping ratio. It is noticed that the RAO decreases rapidly with increasing damping ratio particularly at resonance ($\omega=\omega_n$). At super-critical damping ratios $K < 0.70$, the RAO is < 1.0 so that the heave and pitch motions will actually be less than the disturbing wave height or wave slope. The maximum RAO occur at all damping ratios when the frequency of wave encounter is equal to the natural frequency of the craft (resonant condition).

2.4 Hydrodynamic forces on Planing Craft

The first step in describing the hydrodynamic behaviour of high speed craft (planing) is made by an analysis of the influence of the main design parameters, such as length to beam ratio, deadrise angle, displacement and LCG, on the calm water resistance of the hull. Because the running trim is also important for the vertical accelerations in waves both the sinkage and the running trim of the craft at speed are analyzed too. This research was carried out in a number of steps by Gerritsma and Keuning [12,13]. They extended the original Series 62 experiments as, carried out by Clement and Blount [14], with a large number of similar models but now with 19, 25 and 30 degrees of deadrise of the planing bottom respectively.

A characteristic result from this method is presented in figures 2.6 and 2.7 [15]. In this figure the calm water resistance of two models with 25 and 12.5 degrees of deadrise are compared for two different displacements which is represent by Length/ Beam ratio and the Loading factor ($A_p/\Delta^{2/3}$) and two different L/B ratios. From these figures it is obvious that the influence of the L/B ratio is strong and for the low L/B ratio hulls the influence of the deadrise angle and the displacement on the resistance is very pronounced, while for the high L/B ratio hulls this influence is very small. Also the high L/B ratio hulls show a substantially suppressed “hump” behavior, which is beneficial for patrol boats, which mission profiles ask for operations in various speed regimes.

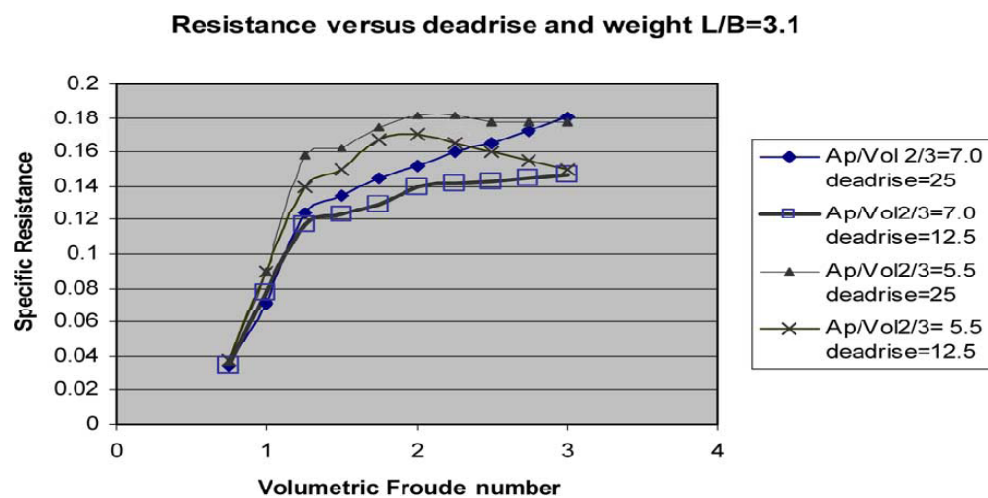


Figure 2.6 : Resistance Curves of Planing Hulls with L/B = 3.1 [15]

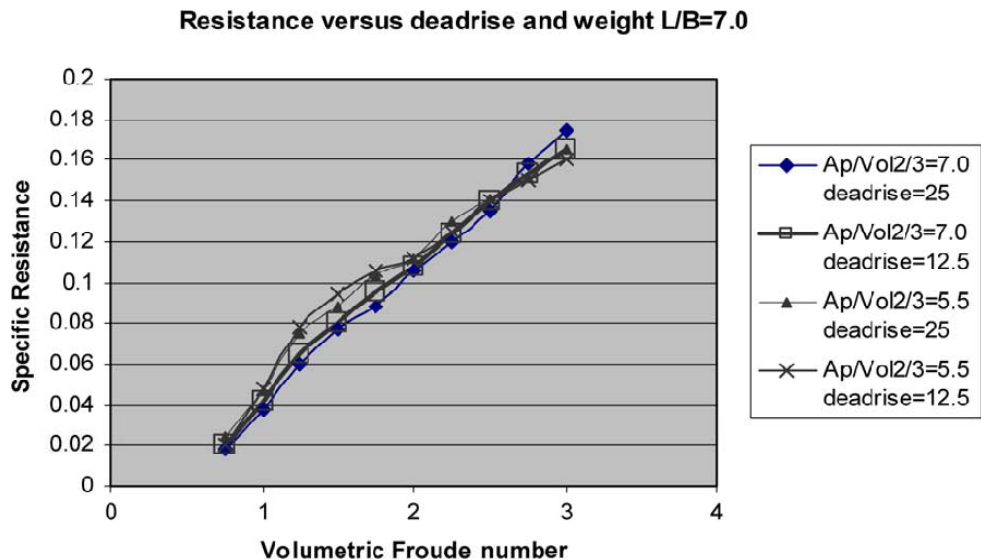


Figure 2.7 : Resistance Curves of Planing Hulls with L/B = 7.0 [15]

From the hump behaviour of the L/B = 2 ship and its lower resistance at the highest speeds it may also be concluded that the hydrodynamic lift plays a much bigger role with these ships when compared to the higher L/B ratio hulls. This is a known phenomenon with planing hulls where the L/B ratio may be considered as inversely related to the aspect ratio of a wing analogue.

The formulations of the hydrodynamic forces acting on high speed craft sailing in waves used in the present report are largely based on the mathematical model as first presented by Zarnick [16] and later further extended by amongst others by Keuning [17,18]. In the present report the formulations will be restricted to a short summary. The complete set of formulations of all forces involved may be found in these References. The development of both the hydrostatic and hydrodynamic lift forces on a fast moving ship is described by making use of a strip theory type approach. Dividing the ship in an arbitrary number of segments along its length (strips) the force on each of the segments may be considered to be constituted of a hydrostatic component related to the displaced water, a dynamic component related to the change of momentum of the incoming fluid and a viscous part, i.e.:

$$\partial F = \frac{D}{Dt}(m_a v) + C_{dc} \rho b v^2 - a_{bf} \rho g A \cos \varphi \quad (2.11)$$

Which:

m_a = added mass of the strip,

v = vertical velocity of the strip,

C_{dc} = cross flow drag coefficient,

b = instantaneous half beam of the section,

a_{bf} = buoyancy correction coefficient,

A = instantaneous submerged sectional area,

ρ = specific density,

g = acceleration due to gravity,

and u and v are the velocity components along the length of the hull resulting from the combination of the forward speed, the heave and pitch motion and the wave orbital velocities, and may be expressed as:

$$u = \dot{x}cg \cos \theta - (\dot{z}cg - wz) \sin \theta, \quad (2.12)$$

$$v = \dot{x}cg \sin \theta - (\dot{z}cg - wz) \cos \theta - \dot{\theta}\xi \quad (2.13)$$

Which:

$\dot{x}cg$ = forward speed,

$\dot{z}cg$ = vertical velocity,

$\dot{\theta}$ = pitch velocity,

wz = vertical orbital velocity component.

Explanation of these expressions yields for the vertical force on each of the sections:

$$\partial F = \left\{ -m_a \frac{dv}{dt} - v \frac{dm_a}{dt} + um_a \frac{dv}{d\xi} + uv \frac{dm_a}{d\xi} + C_{dc}bv^2 \right\} \cos \theta - a_{bf}\rho gA \quad (2.14)$$

From these expressions it may be seen that for the hydrodynamic lift component the added mass and its distribution over the length as well as the change in time play an important role. For the determination of the hydrodynamic lift the change of the added mass of the cross section in time is of prime importance. In the present approach the determination of the sectional added mass is carried out

considering it to be directly related to the instantaneous maximum submerged maximum beam of the section under consideration, which is additionally corrected for the “pile-up” of the water in the dry chine sections. The expression for added mass following Wagner’s approach then reads:

$$m_a = \frac{1}{2} \rho \pi b^2 k_a \quad (2.15)$$

Whereas k_a is a coefficient, which may be determined for each section and which is dependent on the beam to draft ratio and the deadrise angle. The magnitude of a_{bf} and k_a are determined separately in the steady state equilibrium condition and considered constant also for the motions in waves. Due to the fact that in a non-linear approach the relative motion of the ship with respect to the disturbed water surface is no longer considered to be small, the change of shape of the actual submerged part of the cross section is taken into account. By doing so the change in added mass in time, needed in the formulation of the hydrodynamic lift, is taken into account. In the present approach the added mass of each section of the fast ship is considered to be frequency independent. On the other hand the added mass is taken to be dependent on the actual momentaneous submerged geometry of the section and so very much time dependent. The validity and the importance of such an approach for the assessment of the hydrodynamic lift forces on the planing hull, is demonstrated by Keuning [18]. The cross flow drag term is determined using the instantaneous value of the normal velocity component on each of the sections. The cross flow drag coefficient C_{dc} is determined using the work of Shuford for V shaped sections and is:

$$C_{dc} = 1.30 \cos \beta \text{ in which: } \beta = \text{the deadrise angle of the sections.}$$

In general it is found that this cross flow drag is of minor importance when compared with the other forces involved. Due to the dynamic lift and the flow separation over at least a part of the chine’s and the entire transom the buoyancy force, which is determined supposing hydrostatic pressure distribution, needs a correction. The buoyancy related lift therefore is corrected by a correction coefficient. This correction coefficient is a_{bf} and in the present approach this a_{bf} is

assumed to be the same for all sections along the length of the ship were displacement of water is present. In the computer code FASTSHIP developed by the Delft Shiphydrodynamics Department for the calculation of the heave and pitch motions of fast ships in irregular waves, see [18], the values of k_a and a_{bf} are determined from the equilibrium condition of the craft at speed in calm water (no waves). For this particular condition the running trim (pitch) and the sinkage (heave) of the ship are determined from the results of the Delft Systematic Deadrise Series (DSDS), see [13]. Combining these results with the forces calculated in the equations of motions the unknowns can be solved.

One other source for the non-linear behaviour of the planing hull in waves may be found in the wave exciting forces. These originate from the large relative motions that these craft perform with respect to the incoming waves. From the research as reported by Keuning [17] it was revealed that the wave exciting forces on a fast monohull sailing in waves are dominated by the non-linear Froude Kriloff component. This is an important conclusion when these non-linear wave-exciting forces are to be calculated in a time domain solution, in which no frequency dependency can be accounted for. This non-linear Froude Kriloff force is calculated by integration of the dynamic pressure in the undisturbed incoming wave over the actual momentaneous submerged area of the hull whilst performing large amplitude relative motions with respect to the disturbed water surface. For this calculation the full geometry of the hull from keel to deck is being used. The expression used yields:

$$F_{\dot{k}} = 2\rho g y_w \zeta - \rho g k A \zeta \quad (2.16)$$

Which:

$F_{\dot{k}}$ = Froude Kriloff force on section,

ρ = specific density of water,

y_w = momentaneous beam section on the waterline (time dependent),

ζ = wave height (time dependent),

k = wave number,

A = momentaneous submerged area of section

2.5 Transom or Stern Flap Performance

Practically, transom or stern flaps have been used on many high-speed craft, such as survey vessel, surveillance and patrol craft, and pleasure craft [19]. A transom flap represents an extension of the hull aft of the transom in the form of a flat plate. The flap is incorporated to the transom at an angle relative to the centreline buttock of the ship [20], as in figure 2.8 and 2.9. Every transom flaps, independent of what vessels size or type they are used on, create a vertical lift force at the transom, and modify the pressure distribution on the after portion of the hull. The modification of the afterbody flow field causes the principal performance enhancement on a hull.

A transom flap create the flow to slow down under the hull at a location extending from its position to a point generally forward of the propellers. This decreased flow velocity will cause an increase in pressure under the hull, which in turn, causes reduced resistance due to the reduced afterbody suction force (reduce form drag). Wave heights in the near field stern wave system, and far field wave energy, are both reduced by these devices. Localized flow around the transom, which represents lost energy through eddy making, wave breaking, and turbulence, is significantly modified by the stern flap. The flow exit velocity from the trailing edge of the flap is increased in comparison with the baseline transom, leading to a lower speed for clean transom flow separation, and again, reduced resistance [21].

Secondary effects of the transom flap include the lengthening of the hull, improved propeller-hull interactions, and improved propeller performance due to reduced loading, and reduced cavitations tendencies.

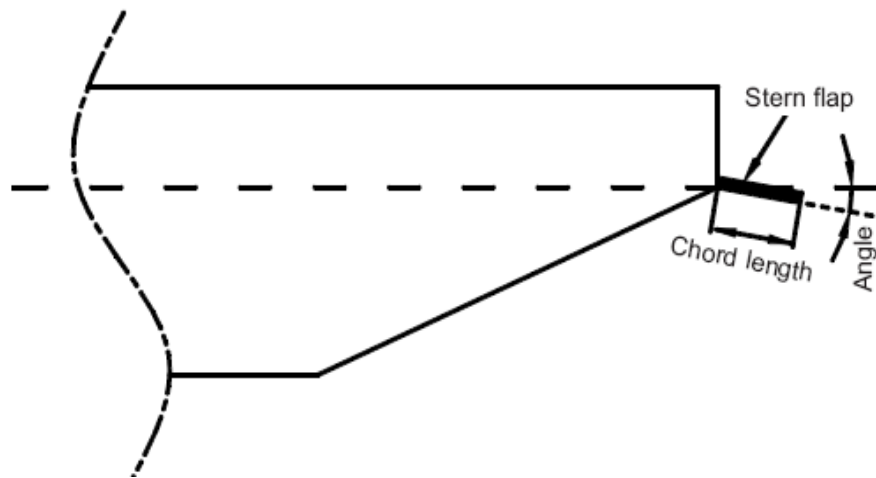


Figure 2.8 : The location of Stern Flap [22]

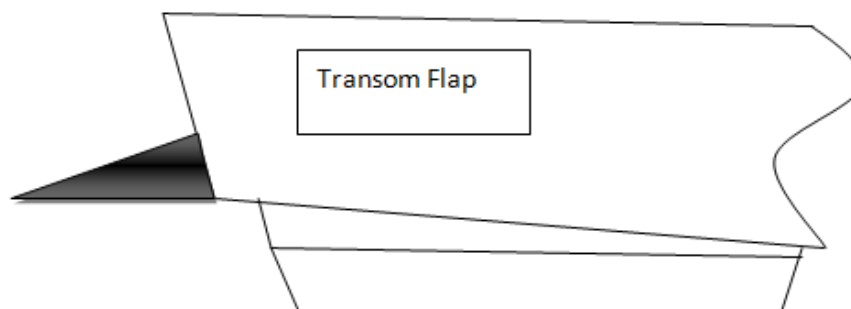


Figure 2.9 The Flap is mounted to the Transom at an Angle Relative to the Ship Centreline Buttock [22]

A transom flap concept was studied by Brown in 1971, which he used a simple expression for the increase of lift, drag and moment. Later, Wang Long Wen [23] used Martin's result in his research of the behaviour of planing craft in waves. He used a flap for reducing calm water resistance and controlling wave-induced motions of planing craft. In his study, Long Wen used Martin's equation for the hull dynamic and Brown's empirical equations for the flaps with a small modification to match his data. The flaps were treated as quasi-steady state force and moment input to the hull.

Long Wen performed five types of tests on a prismatic hull model at the Ship Hydromechanics Laboratory of the Delft University of Technology in the Netherlands. These tests studied as follows:

1. The effect of full width flaps on resistance, lift and trim for speed coefficient between 2.3 to 2.9 and flap angles between 0° to 3° .
2. The forces and moments on the flap for flap length to beam ratios between 8.3% to 16.7% and flap angles between 0° to 9° .
3. The responses of the model when excited by oscillating flaps.
4. The responses of the model without controllable flaps when excited by waves with wave length to vessel length ratios 1 to 6.
5. The responses of the model with controllable flaps when excited by waves with wave length to vessel length ratios 1 to 6.

The results from model tests were very positive. The calculations and experiment results had very good agreement for all of the tests performed. The flaps were useful for reducing the steady state resistance and for controlling the wave induced motions of a planing craft.

2.6 Theory of Foil

Motion control may be effective by reducing the heave and pitch of a high speed vessel especially in planing hull. In order to decrease these motions, the foil was effective technique rather than other methods. Basically the foil system was similar concept with trim tab, transom flap and interceptor which to reduce the heave and pitch motion and also to optimise the resistance. By adapting the foil, the damping of heave and pitch will be increased that resulting the reducing of the heave and pitch motions. Normally the foil was placed at the bow of the vessel because the vertical motions are largest at this area. But in this case, the foil was modifying by fit in below aft of the vessel in order to improve the seakeeping capability and maintained her cruising speed 25 knots at minimum sea state code 3.

The study will be focused on submerged foil which it gives an advantage in avoiding slamming, cavitation and ventilation. However, the cavitation and ventilation depends on the local flow around the flow which is affected by foils

design, the angle of attack of the incident flow to the foil and also foil motions. The higher the ship speed, the larger probability of the cavitation and ventilation cause of this foils add drag to the vessel. The effect of foil system will result reducing the trim angle which contributes the good seakeeping. The phasing flap angles can be controlled relative to trim angle (τ) which is cause by a pressure distribution on the hull and would result a trim moment on the vessel that reduces trim angle at high speed condition. It also shows that the dynamic lifting force become most significant and the seakeeping characteristics are considerably different which is given a great influence to the ship performance.

The trim angle of the vessel can be determined by considering the lift coefficient of the foils. Lift coefficient for foil system depend on many parameters, such as:

1. Angle of attack (α) of the incident flow.
2. Flap angles, (δ).
3. Camber (f).
4. Thickness to chord ratio.
5. Aspect ratio (Λ).
6. Ratio between foil submergence, h and maximum chord length, c .
7. Submergence Froude Number.
8. Interaction from upstream foils.
9. Cavitation number.
10. Reynolds number.

The steady lift force depends on α and δ . When α and δ are small, the linearly dependent on α and δ . If the foil has a camber, the lift is non-zero when α and δ are zero. However, α or δ are large then cavitation and ventilation will be materialized which it depending on speed and submerged of the foils. Refer to figure 2.10, the substantial decrease in lift as a consequence of ventilation. The magnitude of lift force with cavitation depends on the cavitation number. The suction side of the foil may be partially or fully cavitating, which partially cavitating may lead to unsteady lift forces.

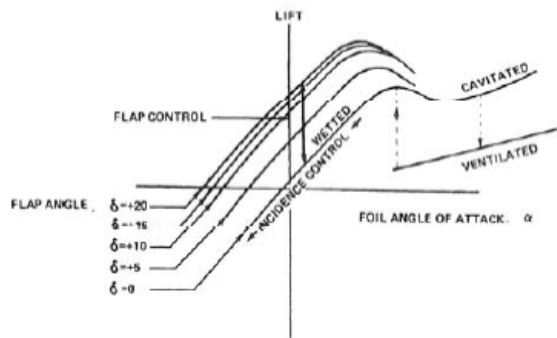


Figure 2.10 : Typical Foil Lift Curves [24]

Foil can give important heave, pitch and roll damping. The damping increases linearly with forward speed and it is important to high speed craft. The damping depends on the details of the foil and in a quasi-steady approximation, is proportional to the lift on the foil. Normally, the flow is assumed to be in 2D and there is no effect of boundaries such as the free surface effect, meaning that cavitation and ventilation are not used in this calculation.

But in this study, the parameters foils use are without chamber i.e. α and flap δ which gives no effect on cavitation and ventilation. NACA 0012 series foil was identified to be used in this study based on the performance research that foils which gives extra benefit than other series.

2.6.1 Physical Features of a Foil

The foils of the fully-submerged concept are designed to operate at all times under the water surface. The struts which connect the foils to hull and support it when the ship is foilborne generally do not contribute to the total hydrofoil system lifting force. In this configuration, the hydrofoil system is not self-stabilizing. Means must be provided to vary the effective angle of attack of the foils to change the lifting force in response to changing conditions of ship speed, weight and sea conditions. The principal and unique operational capability of hydrofoils with fully-submerged foils is the ability to uncouple the ship to a substantial degree from the effect of waves. This permits a relatively small ship to operate foilborne at high speed in sea

conditions normally encountered while maintaining a comfortable motion environment for the crew and passengers and permitting effective employment of equipment. Example of classification used for foil is presented in figure 2.11 which described the parameter of every section of foil.

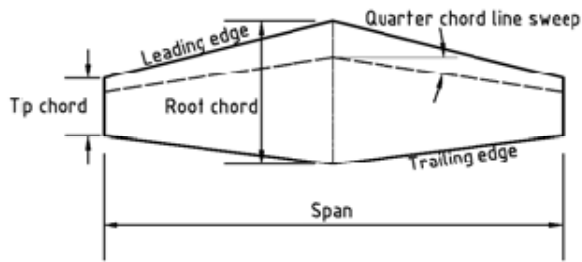


Figure 2.11a : Foil Parameters Plan View[24]

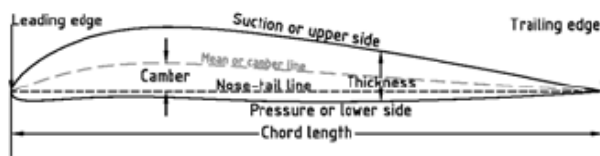


Figure 2.11b : Foil Parameters Sectional View[24]

2.6.2 Selection of Foil and Strut

In this study, the selection of foil and strut only put on the stern of the vessel in order to reduce the couple heave and pitch and also to optimize the resistance. By using the theory of hydrofoil vessel on fully submerged foil system, the planing craft with M-Hull can reduce the motion and resistance will be optimized in specific speed at regular waves. Figure 2.12 show the classical of foil configuration that has been used in design application. The foil are able to support the weight of the vessel at maximum speed either forward or aft foil. However, from time to time the foil system also has great improvement in term of size, geometry and function according to certain vessel.

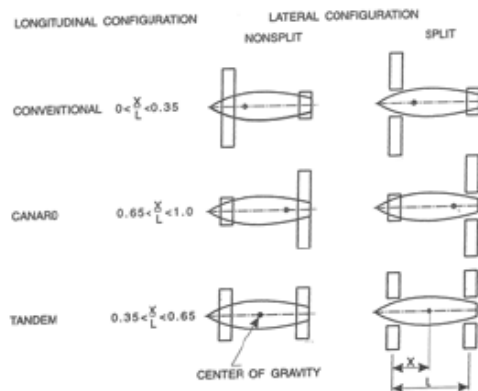


Figure 2.12 : Foil Configurations [24]

Basically, the aft foil and strut are located at $\frac{3}{4}$ from the bow of the vessel [25] But in this study, the aft foil and strut is placed exactly at transom of the vessel. The reason is to obtain the lower pitch angle (α) in order to reduce the couple heave and pitch motion and the same time to avoid interaction occurred between propulsion and foil. Figure 2.13 show that dimension of the foil and strut that has been used for selection of the foil system on planing craft (M-hull) using NACA series 0012.

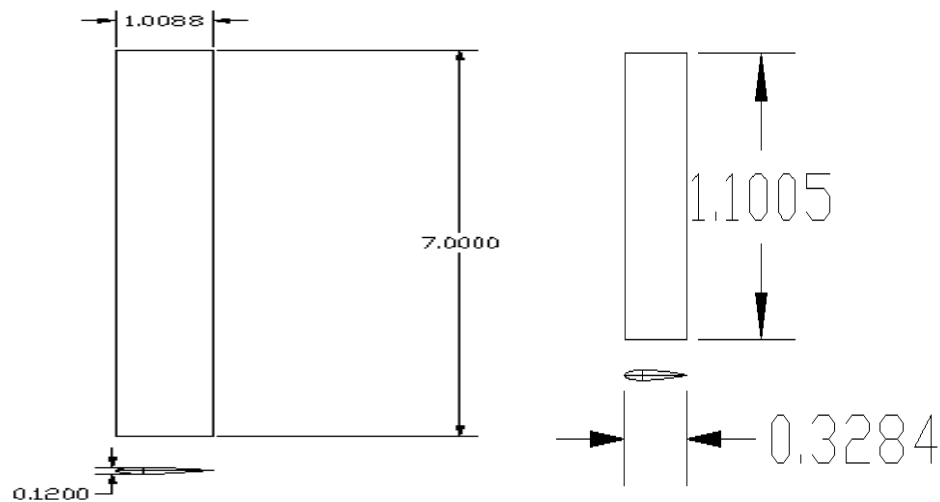


Figure 2.13 : Dimensions of the Research Foil and Strut

Based on the theoretical prediction of resistance and seakeeping, it shows that reducing the motion at best resistance optimisation at zero degree angle of attack which show in figure 2.14. The layout of foil should at flat condition in order to

avoid the cavitations occur which affected the performance of the vessel in term of resistance and motion.

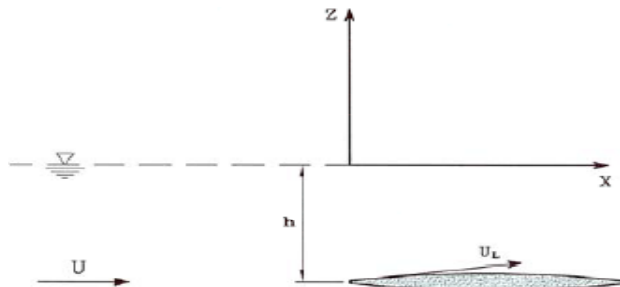


Figure 2.14 : Resistance and Motion Optimisation at 0° Angle of Attack [24]

The minimum drag condition occurs for angles of leeway or attack of less than 2 degrees. Beyond 2 degrees, friction drag rises rapidly. The ambient flow velocity is assumed small relative to the speed of sound that is the fluid may be considered incompressible. Figure 2.15 shows drag plotted in curves equivalent to the lift-coefficient curve. As can be seen, drag can also be expressed as a coefficient.

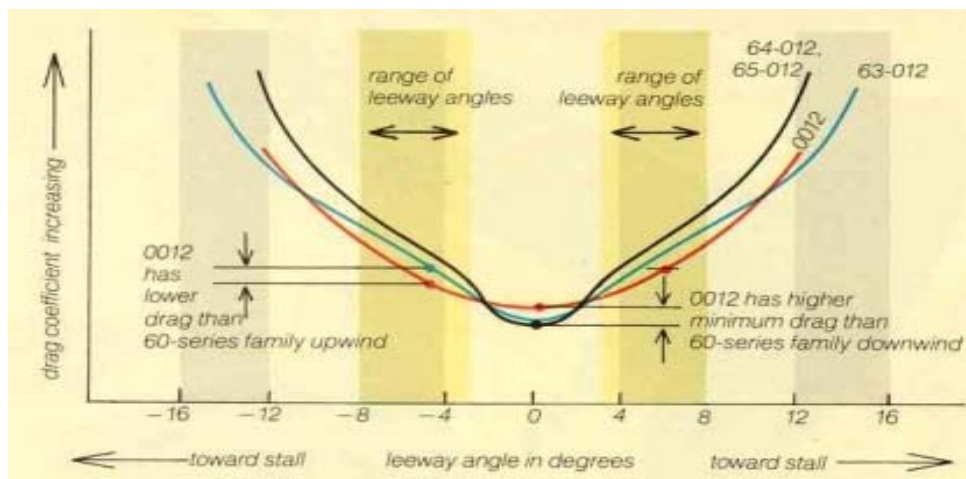


Figure 2.15 : The Drag Coefficient as a Function of angle of attack (leeway) [25]

CHAPTER 3

METHODOLOGY

3.1 Introduction

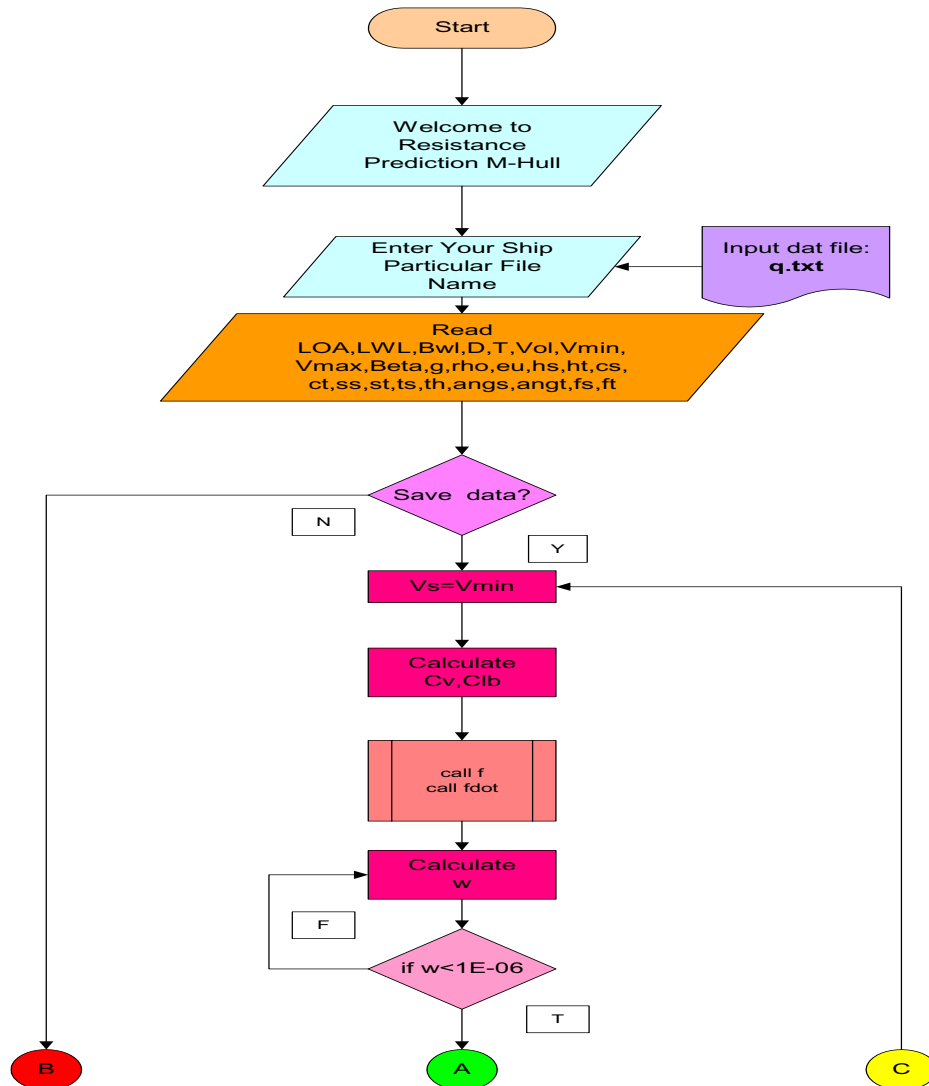
The main purpose of this research is to obtain the most favourable resistance and to improve the feasibility and effectiveness of the stern foil as means of pitch and heave reducing. The research is conducted to proof that by incorporating the stern foil, it will reduce the heave and pitch motion which resulted in good performance on the vessel. In this research methodology approaches categorized into the theoretical prediction for resistance by using Savitsky method and seakeeping predictions for the response heave and pitch motion of vessel is using the strip theory (Geritsma and Beukelman II Added Resistance Method).

Basically the methodology of this study is separated into two different areas whereas explained as follows:

3.2. Resistance

The study will be focused on theoretically for predicting the resistance of with and without stern foil. Prediction of resistance of the vessel will be compute by using FORTRAN software which the source code of this program based on Savitsky method for hull and 2D formulations for foil and strut. The calculation will

determined at various speed in calm water. The flowchart of FORTRAN programming for calculation ship resistance with stern foil as shown in figure 3.1 and 3.2 below:



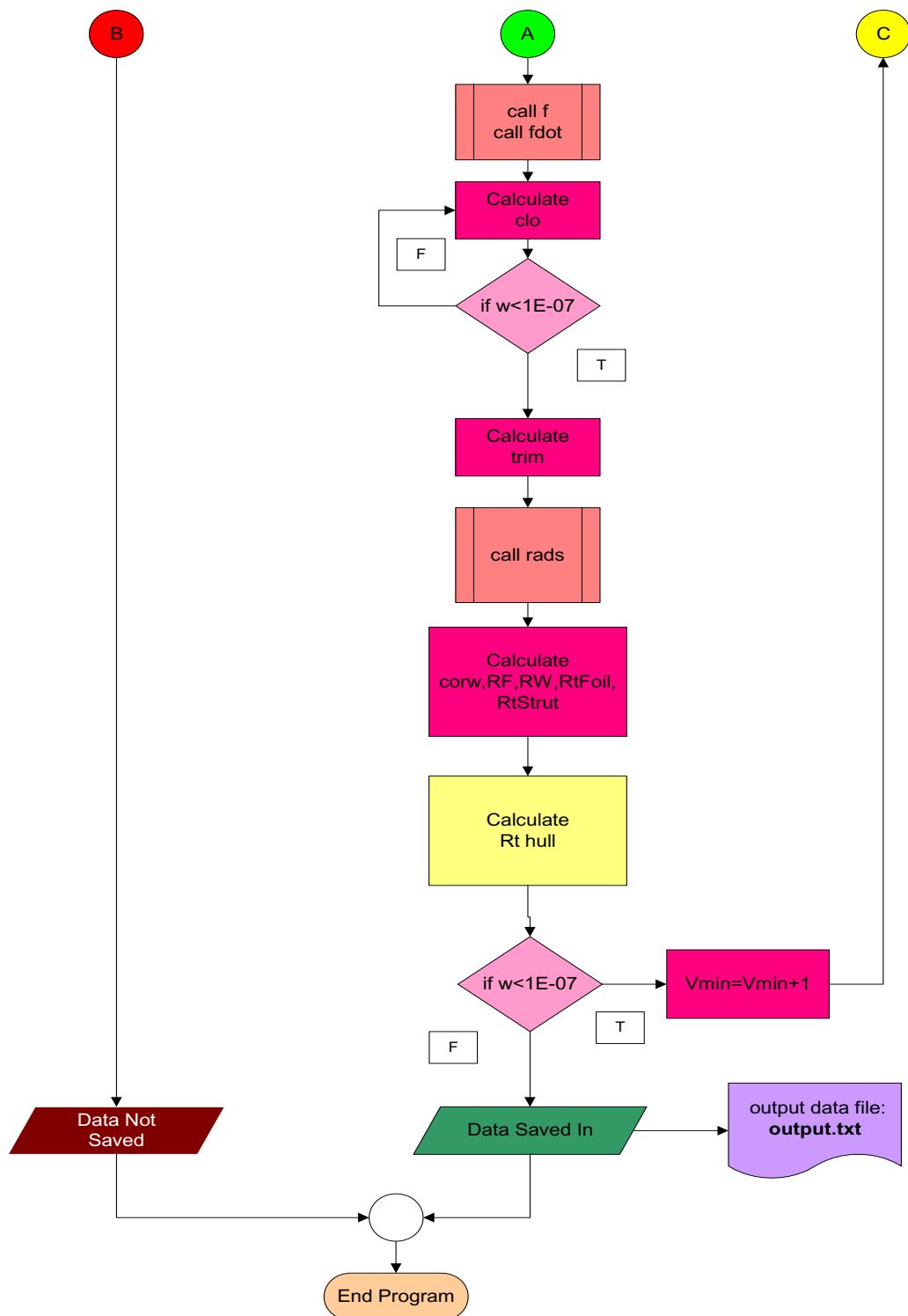


Figure 3.1 : The Flowchart of FORTRAN Programming for Resistance Prediction

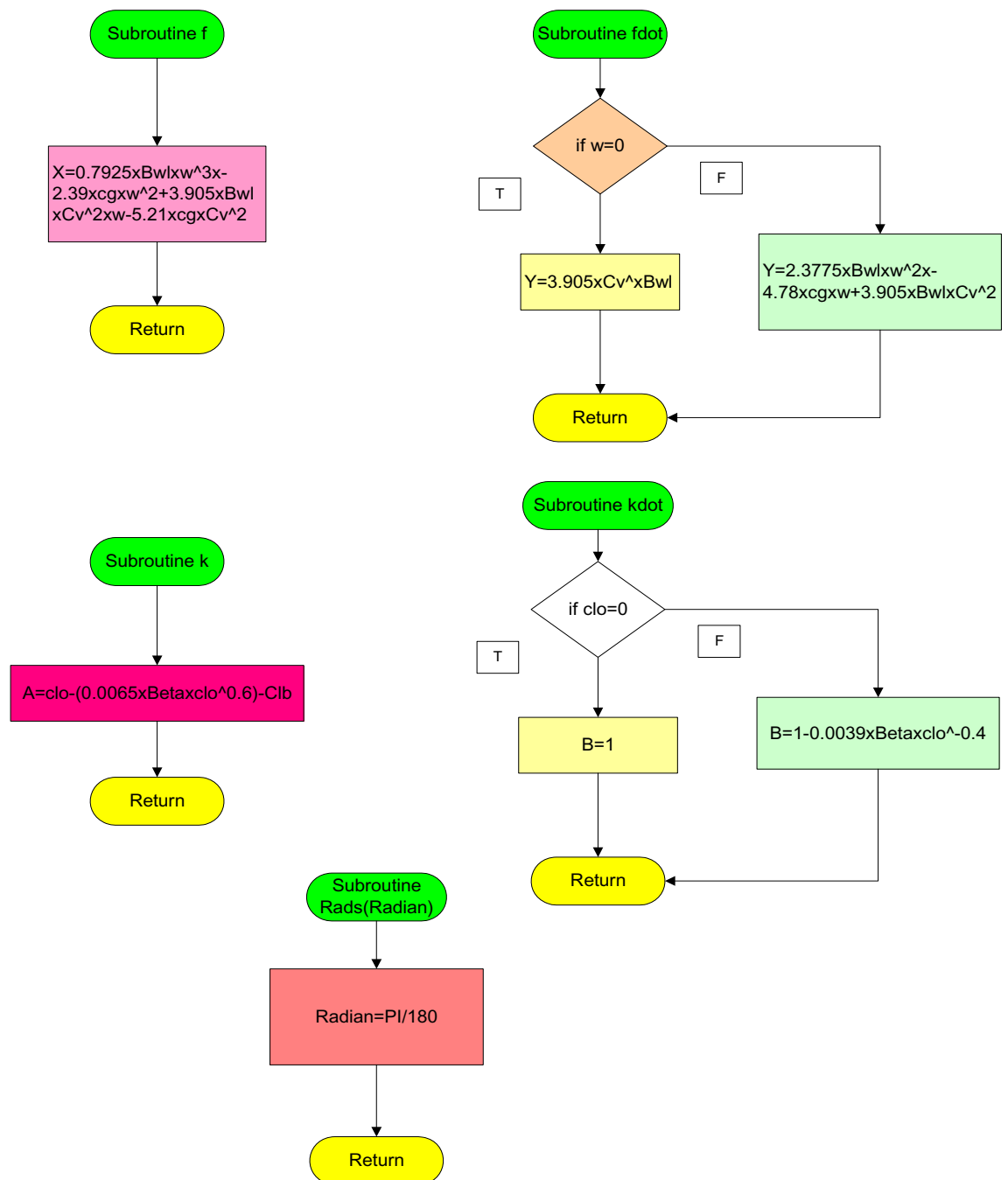


Figure 3.2 : The Flowchart Subroutine of FORTRAN Programming for Resistance Prediction

In term of experiment, the resistance test was carried out by using towing carriage facilities at Marine Technology Laboratory, Universiti Teknologi Malaysia. The experiment method is utilized for two difference modes of test with and without

stern foil at various speeds in calm water condition. The procedure of resistance test is described in Appendix B. The flowchart of resistance test procedure is shown in figure 3.3 below:

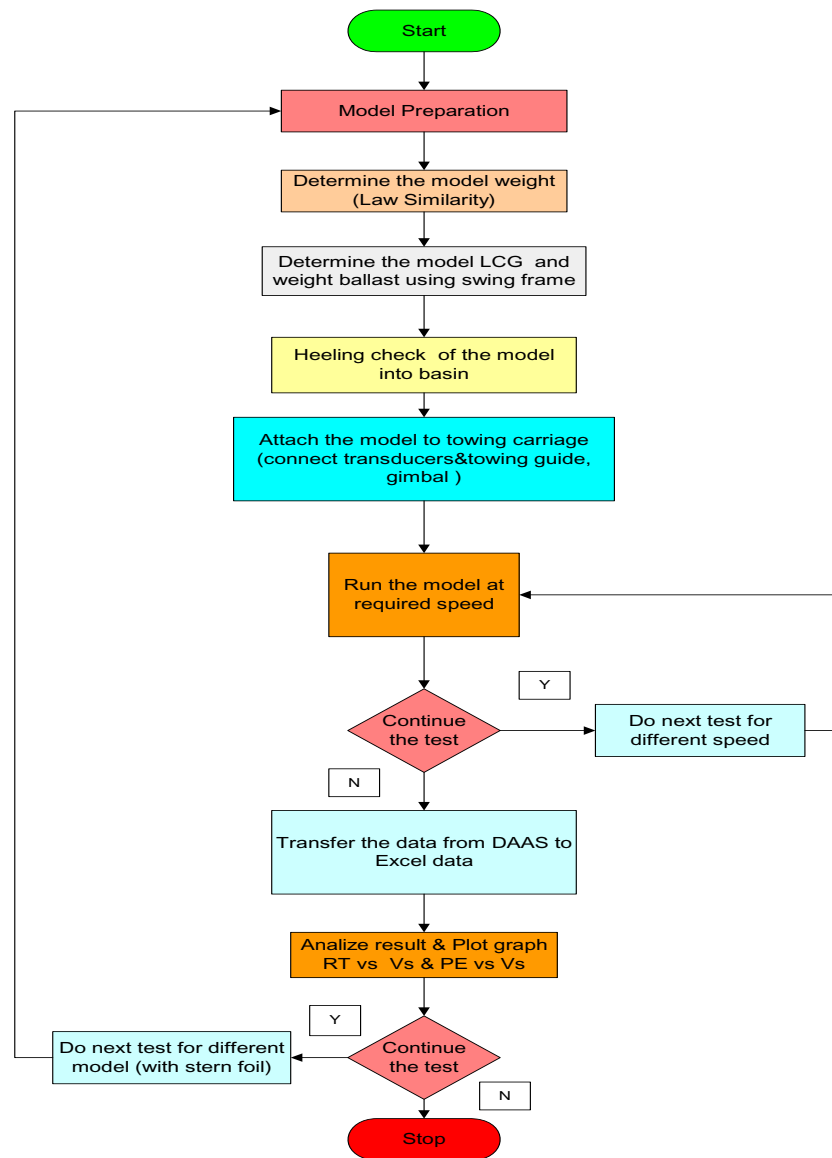


Figure 3.3 : The Resistance Test Procedure

3.3 Seakeeping

For seakeeping analysis, the study is to examine of motions criteria which dominate in high speed region such as heaving and pitching. The characteristic of the motions will investigate in strip theory method which using computational method Maxsurf SEAKEEPER in early prediction of these motions. SEAKEEPER used Strip Theory (Geritsma and Beukelman II Added Resistance Method) [26] to predict the coupled of heave and pitch response of a vessel in a seaway. To calculate the global equations of motions the vessel is split into transverse sections. These are treated as two-dimensional sections in order to compute their hydrodynamic characteristics and these are then integrated along the length of the vessel to obtain global coefficients of motions.

The prediction using SEAKEEPER validates the motions result of hull form incorporating with and without stern foil of the vessel respectively. As a preliminary requirement, offset data of the particular hull need to be prepared in MSD file format. The supplementary of the offset data together with principal particulars and some basic hydrostatic data would make the Maxsurf SEAKEEPER predicts the seakeeping analysis. The module [26] able to generate data on the following analysis;

1. Regular Response
2. Irregular Response
3. Added Resistance
4. Dynamic Loads
5. Motion Sickness Index

The Maxsurf SEAKEEPER module is considered as a part of frequency domain based and applies Strip's theory in order to perform analysis in regular and irregular wave conditions. This analysis has been undertaken at the top of sea state code four, which has a characteristic wind speed of 20 knots. Using a Pierson-Moskowitz spectrum, the wave spectrum has the following details:

Table 3.1 : Wave Spectrum Details

Parameters	
Characteristic Wind Speed (knots)	20
Characteristic Wave Height (m)	1.848
Modal Period, T_0 (sec)	7.52
Average Period, T_{ave} (sec)	5.802
Zero Crossing Period, T_z (sec)	5.317

The spectral co-ordinates of the Pierson-Moskowitz wave spectrum are defined by the following formula:

$$S_{PM\zeta}(\omega) = \frac{A}{\omega^5} e^{\left(\frac{-B}{\omega^4}\right)} \quad (3.1)$$

Where $A=8.11 \times 10^{-3} \text{ g}^2$ and $B = \frac{0.74g^2}{U_{wind}^4}$

The analysis will be carryout in 63 regular waves with maximum 200 frequencies to produce Response Amplitude Operators in order to calculate motion spectrum. The flowchart of procedure to analyze the specific motion characteristics as follows:

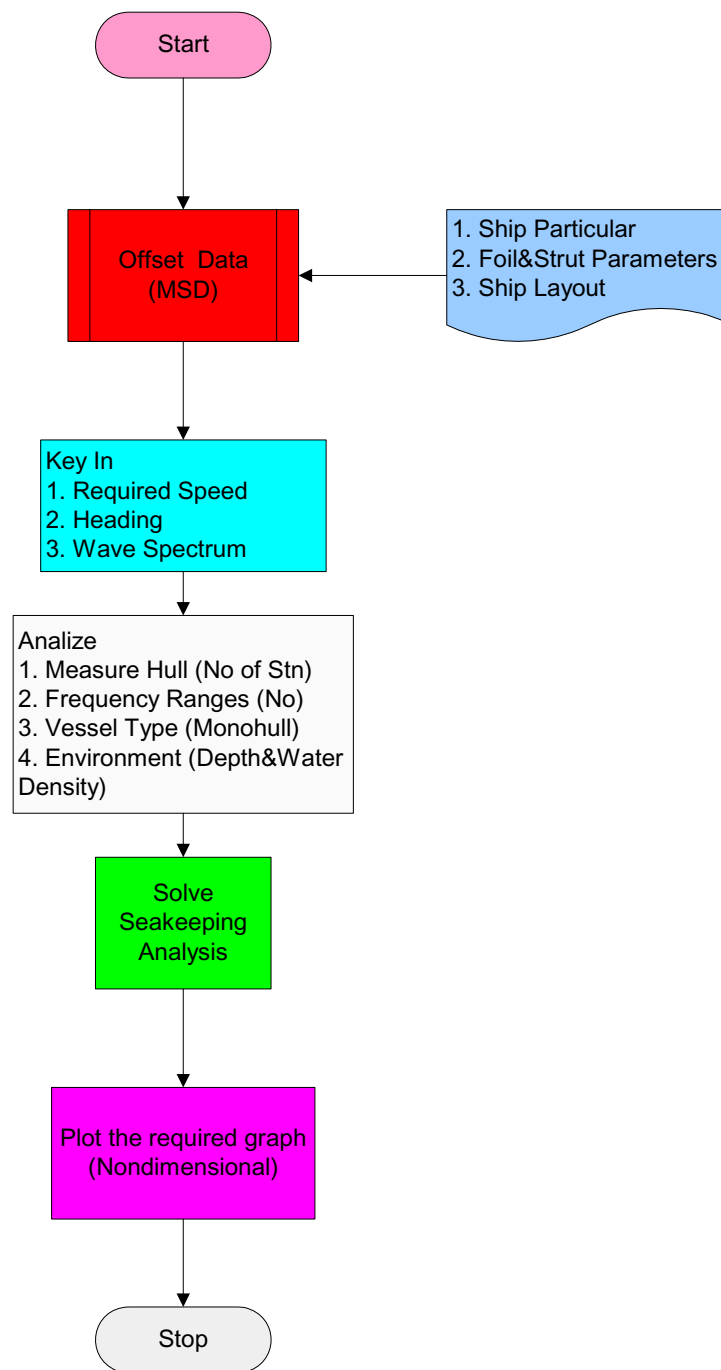


Figure 3.4 : The Flowchart of Motion Prediction using SEAKEEPER

Seakeeping test will be conducted on Head Seas at regular waves according to its characteristic. Particular models will be tested at difference wave length to ship length ratio (L_w/L_m) forward speed 25 knots with and without stern foil at regular

wave's condition in order to obtain RAO's pitch and heave motions. The procedure of seakeeping test is described in Appendix B. The parameters of experiments are described below:

Table 3.2 : Summaries of Experiment Data

L_w/L_m	L_w	Wave period, T_w	H_w (m)
Wave Characteristics		$[(2\pi*\lambda/g)]^{0.5}(s)$	$H_w = L_w \times 0.02$
0.50	1.0215	0.8089	0.0204
0.60	1.2258	0.8861	0.0245
0.80	1.6344	1.0231	0.0327
1.00	2.0430	1.1439	0.0409
1.20	2.4516	1.2531	0.0490
1.40	2.8602	1.3535	0.0572
1.60	3.2688	1.4469	0.0654
1.80	3.6774	1.5347	0.0735
2.00	4.0860	1.6177	0.0817
2.20	4.4946	1.6967	0.0899

The seakeeping test procedure at corresponding speed 25 knots and wave characteristic in regular waves at head sea condition are described in flowchart as show in figure 3.5:

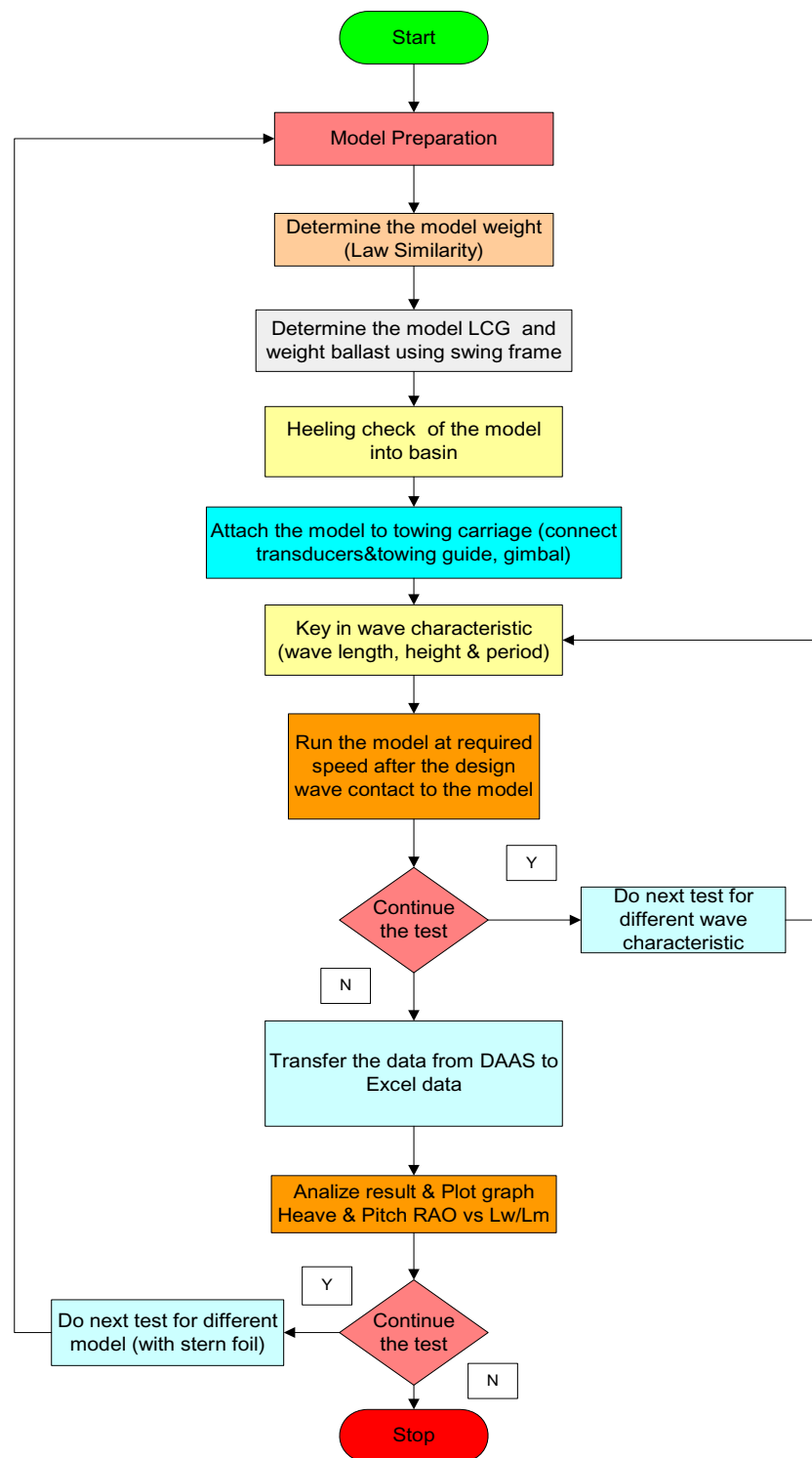


Figure 3.5 : The Seakeeping Test Procedure

3.4 Concluding Remarks

The procedure in theoretically and experimentally basis are important to ensure that project outline are in scope of work and method that has been used to materialized the research. In this case, the resistance and seakeeping test were conducted in order to validate the theoretically prediction and to check the ship behaviour at specific speed and wave characteristic.

CHAPTER 4

RESISTANCE

4.1 Introduction

There are numerous methods available today by which a ship designer can obtain an estimate of the resistance of high speed craft. Studies made by various authors and the present study show that no single method is accurate to predict the resistance over a wide range of speeds. Research made over the last few years show that some of the modern regression methods are sufficiently accurate over the speed range for which they had been developed while some of the other methods have been less than satisfactory. While using regression methods designers generally tend to satisfy the non-dimensional range for their particular hullform. The most important aspect i.e., the hull shape needs to be considered bearing in mind the limitations applicable to a particular method. To ensure confidence in accuracy it is imperative to investigate the regression methods in detail and compare results of a number of vessels for whom models have been previously tested.[27]

Generally, perhaps the most significant contributor to good prediction reliability is the appropriate selection of the prediction method. The selected prediction method should be built from hulls that share the same basic character as the vessel under review. Referring to drawings of the method's hull forms is the first step to selecting a suitable method. After principal hull type, the method's range of data set parameters must be considered. The most critical parameter to watch is speed (typically Froude number), then the hull form parameters. The obvious way to avoid

difficulty is to evaluate many different methods and to select one that offers a good correlation between a ship and the method.

4.2 Resistance Components

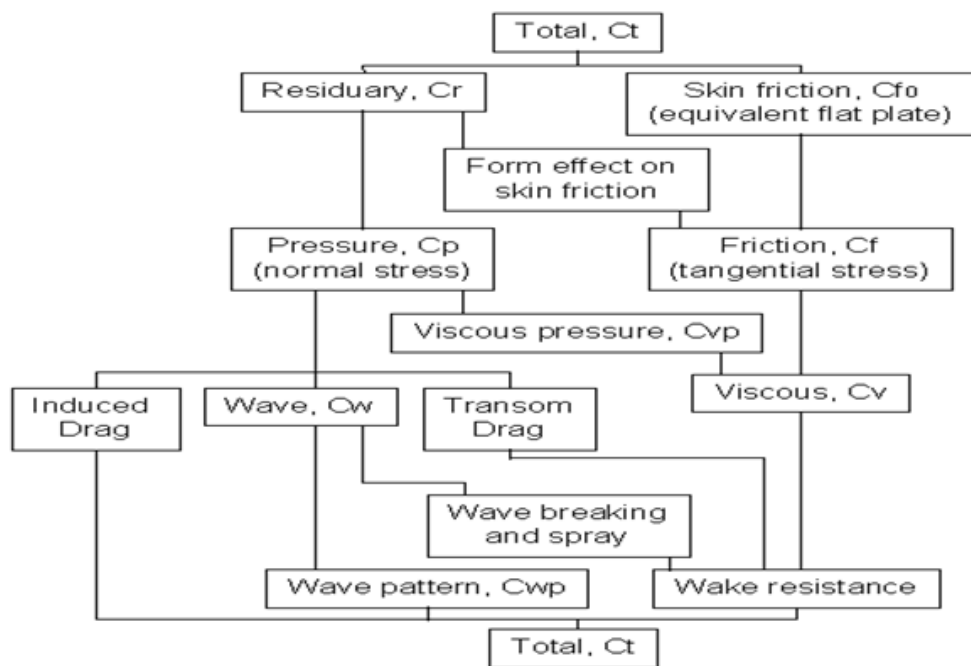


Figure 4.1 : Breakdown of Resistance into Components

The resistance of a surface vessel may be separated into components attributed to different physical processes, which scale according to different scaling laws. Such a breakdown is presented in figure 4.1. The resistance of a vessel (neglecting air resistance) is due to shear and normal fluid stresses acting on the vessel's underwater surface. The shear stress component is entirely due to the viscosity of the fluid, whilst the normal stress component may be separated into two major components: wave making, due to the generation of free surface gravity waves (inviscid) and a viscous pressure component caused by the pressure deficit at the stern due to the presence of the boundary layer (viscous). The transom stern presents a special case and this has been included as a pressure drag component.

The total resistance can be break down into viscous resistance dependent on Reynolds number and non viscous resistance i.e. wave making resistance or called it residuary resistance dependent on Froude number components. This is described in equation (4.1). The non viscous, R_R , contains the inviscid component and the viscous resistance, R_F , includes the resistance due to shear stress (Friction drag) and the viscous pressure component (discussed above). In practice the viscous resistance is usually estimated using the ITTC-57 correlation line (C_F) together with a suitable form factor (1+k). Here C_F is an approximation for the skin friction of a flat plate, the form factor is used to account for the three dimensional nature of the ship hull. This includes the effect of the hull shape on boundary layer growth and also the viscous pressure drag component. It should be noted that the ITTC-57 correlation line is an empirical fit and that some form effect is included. A number of researchers have attempted to measure the individual resistance components. Apart from total resistance, it is also possible to measure the viscous resistance and the wave pattern resistance to a reasonable degree of accuracy; from these measurements, the form factor may be derived:

$$R_T = R_F(R_n) + R_R(F_n) \quad (4.1)$$

The viscous resistance component may be derived from measurements of the velocity field behind the hull. The transverse extent of the wake survey will determine how much of the viscous component is measured. For slow speed forms, the viscous debris is concentrated in a wake directly astern of the model. However, for high-speed vessels significant viscous debris, probably originating from the spray sheet, may be observed to extend several times the model maximum beam either side of the model centre line. In addition, this component may also be investigated in the wind tunnel.

The friction resistance is assumed to be dependent on the wetted surface S , the square of the ship speed V , the mass density of the water ρ and the coefficient of friction C_F as follows:

$$R_F = \frac{1}{2} \rho V^2 S C_F \quad (4.2)$$

$$C_F = \frac{0.075}{(\log R_n - 2)^2} \text{ where, } R_n = \frac{VL}{\nu} \quad (4.3)$$

The friction coefficient C_F is dependent on the value of the R_n . This formula is used to estimate the frictional resistance for the model in order to determine the residual resistance and for the ship which to predict the total resistance from the residual resistance. The resistance of the model is equal to the residual resistance of the ship when models tests are carried out at full scale values of the F_n .

However, the residual resistance can be determined by several methods based on statistical and experimental data especially for estimate the resistance of high speed craft which is dominate by wave resistance as major component in residual resistance. The formulation for calculate the residual resistance as per equation (4.4). Theoretically the residual resistance dependent to F_n but it also dependent on hull form itself. The most important hull form parameter is the length-displacement ratio $L/\nabla^{1/3}$.

$$R_R = \frac{1}{2} \rho V^2 S C_R \quad (4.4)$$

The friction formulation as was originally used in arriving at the published residual resistance value of the model-ship correlation factor C_A and it given no serious errors need to occur. The procedure for calculating the resistance and effective power will be used for proposed design. The total resistance is calculated from following equation:

$$R_T = \frac{1}{2} \rho S V^2 (C_F + C_A) + \frac{R_{RA}}{\Delta} \quad (4.5)$$

Another factor contributing the resistance is the ship model-ship correlation C_A must be to take account for the effect on resistance structural such as plate seams, welds, and paint roughness. Typical values of C_A given in table 4.1. [28]

Table 4.1 : Typical values of Ship Model-Ship Correlation C_A

Waterline Length in Metres	C_A
12.5	0.00060
25.0	0.00055
50	0.00045
100	0.00035

The effective power P_E can be predicted by using formula as follows:

$$P_E = R_T V \quad (4.6)$$

4.3 Savitsky Method

The initial manual method attempted for calculating resistance was the Savitsky method [10]. This method utilized a table to organize and consolidate calculations; however, several assumptions were made in the derivation of the theory which prevented it from being applicable to the displacement hull design. These assumptions included: the vessel being a planing design having constant dead rise, and being able to predict trim angle to a reasonable degree of accuracy.

Savitsky method most often refers to long and short form methods presented in 1964. This method is oriented toward pure-planing hull operating at hump and beyond. Since these methods have been numerous modified versions from his previous work, there are many formulations were added in order to suit the formulation to high speed regime [29,30,31]. Every version of these methods will give different answers. The Savitsky method consist of important element i.e. lift and torque which it useful method for predicting the performance of high speed craft. The formulation balances the torques from the drag, weight and thrust of the craft.

However, the method assumed that the thrust is parallel to the axis of the thruster which it not occurs in real scenario. The results of the assumption not always correct due to spray drag not including in this version.

Then, Hardler [32] presented a method to predict the performance of planing hull (high speed). He used savitsky's formulas to calculate the hydrodynamic forces on the propeller and open water diagram to evaluate the propeller forces. The solution of the three equations of the equilibrium where sum of forces in X and Z direction and the moments be equal to zero yield the unknown; trim angle τ , wetted length L_m and rate of revolutions as shown in figure 4.2. The summaries of the method are given in table 4.2.

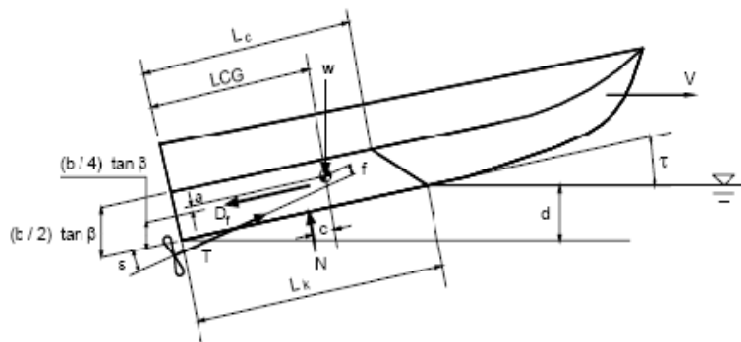


Figure 4.2 : Forces act on Planing Hull [10]

Table 4.2 : The Summary of Savitsky Method

Parameters	$L_{WL}, L/B, B/T, ie, LCB, LCG/L, \beta, C_v, \lambda$
Constraints	$3.7 < \frac{L}{v^{1.3}} < 12.4$ $2.52 < L/B < 18.6$ $-1.7 < B/T < 9.8$ $-0.016 < \frac{LCG}{L} < 0.0656$ $0 < \frac{AT}{Ax} < 1$

	$3.7 < ie < 28.6$
Speed range	$1.0 < F_{nb} \left(\frac{V}{\sqrt{gb}} \right) < 7.0$

The following approach to predict the resistance is follows:

$$\text{Speed Coefficient } C_v = \frac{V}{\sqrt{gb}} \quad (4.7)$$

$$\text{The lift coefficient for finite deadrise } C_{L\beta} = \frac{\rho g \nabla}{1/2 \rho V^2 b^2} \quad (4.8)$$

Then to determine trim angle τ for equilibrium, V is speed in m/s, b is beam of planing area, ρ is a density of seawater 1025 kg/m^3 , and g is acceleration of gravity 9.81 m/s^2 . The lift coefficient for finite deadrise $C_{L\beta}$, can be calculated from:

$$C_{L\beta} = C_{Lo} - 0.0065\beta C_{Lo}^{0.6} \quad (4.9)$$

Where β is the deadrise angle at the mid-chine position (in degree). Then Flat plate lift coefficient can be determined by: C_{Lb} can be calculating by numerical method (Newton Raphson).

$$C_{Lo} = \tau^{1.1} \left(0.0120\sqrt{\lambda} + \frac{0.0055\lambda^{5/2}}{C_v^2} \right) \quad (4.10)$$

$$\tau = 1.1 \sqrt{\frac{C_{Lo}}{(0.0120\sqrt{\lambda} + \frac{0.0055\lambda^{5/2}}{C_v^2})}} \quad (4.11)$$

Where λ is wetted length beam ratio $\lambda = \frac{L_m}{b}$ and can be determined as follow:

$$\frac{lcg}{\lambda B} - 0.75 + \left(\frac{1}{\frac{5.21C_v^2}{\lambda^2} + 2.39} \right) = 0 \quad (4.12)$$

This requires a numerical solution (Newton Raphson). Savitsky gives a formula to correct the mean wetted length ratio λ , to the keel wetted length ratio, λ_k which can be calculated by:

$$\lambda_k = \lambda - 0.03 + \frac{1}{2} \left[0.57 + \frac{\beta}{1000} \right] \left[\frac{\tan \beta}{2 \tan \tau} - \frac{\beta}{167} \right] \quad (4.13)$$

λ_k shall be less than $\frac{L_{WL}}{b}$ so that the bow is essentially clear of the water and the resistance can be predicted from the following equation:

$$R_T = W \tan \tau + \frac{\frac{1}{2} \rho V^2 \lambda b^2 C_F}{(\cos \tau \cos \beta)} \quad (4.14)$$

Which the C_F can be estimate by:

$$C_F = \frac{0.075}{(\log R_{nb} - 2)^2} + \Delta C_F \quad (4.15)$$

$R_{nb} = \frac{V_1 \lambda b}{\nu}$, $\Delta C_F = 0.0004$ which obtained from ATTC Standard Roughness and $W = \rho g \nabla$.

V_1 is the average velocity, which less than the forward planing velocity V owing to the fact that the planing bottom pressure is larger than the free free-stream pressure. The average velocity V_1 can be expressed as:

$$V_1 = V \left(1 - \frac{0.0120 \tau^{1.1}}{\sqrt{\lambda \cos \tau}} \right)^{1/2} \quad (4.16)$$

4.4 Controllable Transom Flaps (Trim Tab)

Controllable transom flaps (trim tabs) have become accepted as a means controlling the trim of high speed craft to optimize the performance. Controllable transom flaps (trim tabs) that can be automatically controlled and may be used to

minimize the trim angle which it is beneficial for resistance in order to minimize drag over the range of speed and loading conditions and dynamic stability in heave and pitch. The simpler fixed flap or wedge will be minimize the drag at the cruising speed and also reduce the vertical ship motions. This is less costly installation but still allows the designer a choice of longitudinal center of gravity positions without concern for penalties since the craft can subsequently be trimmed out with the flap.

A study of flap effectiveness by Brown et al [33] resulted in simple expression for the increase in lift, drag and moment. The flap expression have been validated by [33] over the following ranges as shows in table 4.3 whilst, figure 4.3 show the planing craft equipped with the flap.

Table 4.3 : Parameters of Flap Ranges [33]

Parameters	Ranges
Flap chord (percent of mean wetted length)	0 – 10
Flap deflection (deg)	0 – 15
Trim (deg)	0 – 10
Speed coefficient (C_v)	2 – 7

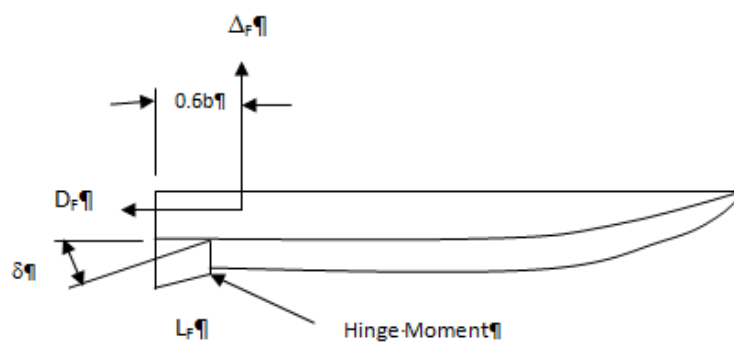


Figure 4.3 : Planing Hull with Transom Flap [33]

4.5 Stern Foil

Basically, the function of the stern foil was similar to trim tab, transom flap and inceptors. But many research found that the foils are suitable to install forward of the vessel which able to reduce the resistance and motion by decrease the trim of the vessel. For example a deeply submerged T-foil is an advantage in avoiding slamming, cavitation and ventilation. The coupling of cavitation and ventilation also depends on the local flow around the foil, which affected by the foil design, the angle of attack of the incident flow to the foil and the foil motion. Probability of cavitation and ventilation are difficult to avoid when the foils are operating at speeds higher than 50 knots.

The last decade has shown an increase in the use of foils on high speed craft. Foils may not only be used to lift the hull partially or fully out of the water but also to provide forces to dampen the motions of the craft when operating in waves. Viscosity effects on foil drag are then estimated by means of empirical formulations while viscosity effects on lift are usually neglected. For model testing at relatively low Reynolds numbers viscosity plays a larger role than at full scale conditions and it may be difficult to derive reliable full scale results from model tests.

In developing a lifting effect for the planing vessel, the overall design objective was to provide benefits to ship operations through increased lift-to-drag ratio, and improved motions in a seaway at all speeds. Other design considerations included waterjet inlet ventilation, lifting body support structure, ship static stability at cruising speed. The starting point in selecting the lifting effect for the planing vessel was a previous concept design that had similar design objectives as shown in figure 4.4. This configuration used a deep-vee hull which located forward of amidships.

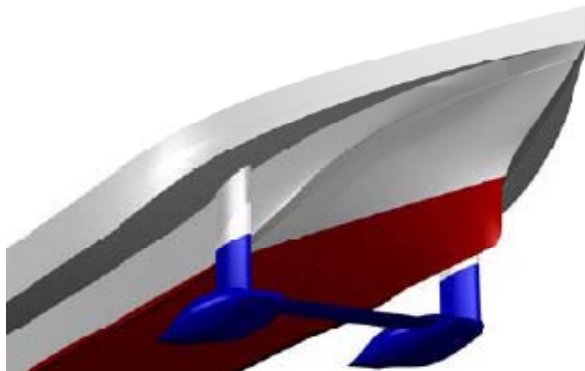


Figure 4.4 : Lifting Effect on Planing Vessel [24]

Although the major reason for the employment of hydrofoils is to lift the hull out of the water to reduce the effect of waves and to reduce the drag at high speed, a planing craft incorporated with foil spends a considerable portion of its life hullborne and must have an efficient hull form to keep the drag low at low speed.

The resistance on the foil is derived by numerical methods, using conservation of fluid momentum. A fluid volume exterior to the foil surface and the trailing vortex sheet is considered and it is bounded far away from the foil by the surfaces of a box with sides that are parallel to either the xy , xz or yz plane which is shown in figure 4.5. At the control surface, S_c that is perpendicular to the vortex sheet far downstream of the foil and the equation can be expressed as:

$$I_w = \frac{z}{2\pi} \int_{-0.5s}^{0.5s} d\eta \frac{[\varphi^+ - \varphi^-]_{TE}}{[(y-\eta)^2 + z^2]} \quad (4.17)$$

For $x-c/2 \rightarrow +\infty$

$$w_i = \frac{1}{4\pi} PV \int_{-0.5s}^{0.5s} \frac{d\Gamma}{d\eta} \frac{d\eta}{y-\eta} \quad (4.18)$$

By using equation 4.18, the following equation can be written in a way similar, that is:

$$I_w = \frac{1}{2\pi} \int_{-0.5s}^{0.5s} \frac{d\Gamma}{d\eta} \tan^{-1} \frac{z}{y-\eta} d\eta \quad (4.19)$$

For $x-c/2 \rightarrow +\infty$

The flow due to the vortex sheet is two dimensional in the yz plane at S_c when the parameters $\Gamma = (\varphi^- - \varphi^+)_{T.E}$ which means that the longitudinal velocity is U at S_c . The momentum flux through the surface enclosing the control volume is zero. The longitudinal pressure force acting on S_c :

$$\frac{\rho}{2} \iint_{S_c} \left[\left(\frac{\partial \varphi}{\partial y} \right)^2 + \left(\frac{\partial \varphi}{\partial z} \right)^2 \right] dydz \quad (4.20)$$

Where $\varphi = I_w$. The additional longitudinal force acting on the control volume is the force opposite the drag force D on the foil. This can be expressed as:

$$D = \frac{\rho}{2} \iint_{S_c} \left[\left(\frac{\partial \varphi}{\partial y} \right)^2 + \left(\frac{\partial \varphi}{\partial z} \right)^2 \right] dydz \quad (4.21)$$

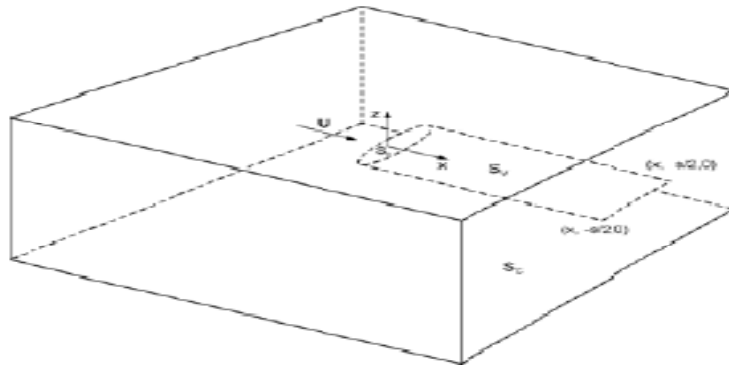


Figure 4.5 : The Drag Force on the Foil by using Conservation of Fluid Momentum[34]

4.6 Resistance Components on the Foil and Strut

The resistance components in the planing craft condition for this study consist of:

4.6.1 Viscous Resistance

The formulation of the viscous resistance coefficient on the foil and strut can be expressed as:

$$C_{Dv} = 2C_F[1 + 2(t/c) + 60(t/c)^4] \quad (4.22)$$

Where:

$$C_F = \frac{0.075}{(\log_{10} R_n - 2)^2} \quad (4.23)$$

$$R_{nc} = \frac{Vc}{\nu} \quad (4.24)$$

The viscous resistance can be calculated as follows:

$$R_v = 0.5C_{Dv}\rho V^2 A \quad (4.25)$$

Where:

C_{Dv} = Viscous resistance coefficient

C_F = Friction coefficient

R_{nc} = Reynolds number

t/c = Foil thickness to chord ratio

V = Ship speed

c = Chord length

ν = Kinematic viscosity

$\rho = 1025 \text{ kg/m}^3$

A = Planform area (the projected area of the foil in the direction of the lift force for zero angle of attack). In the two dimensional, A is equal to the chord length c of the foil.

In the two dimensional analysis, there are very small lift force occur which is it can be assumed that the lift is nearly zero. Both foil and strut are using by similar equation in order to predict the viscous resistance.

4.6.2 The Induced Resistance

The induced resistance due to the trailing vortex sheet of high aspect ratio with zero camber and elliptical loading in infinite fluid were identified in this study. This condition of foil and strut are going well with the Prandtl lifting theory which it will derive the lift and resistance coefficient. Figure 4.6 show the flow is locally two dimensional in the transverse cross sectional xz plane. Then it will give effect to the vertical inflow velocity that changes the angle of attack of the incident flow.

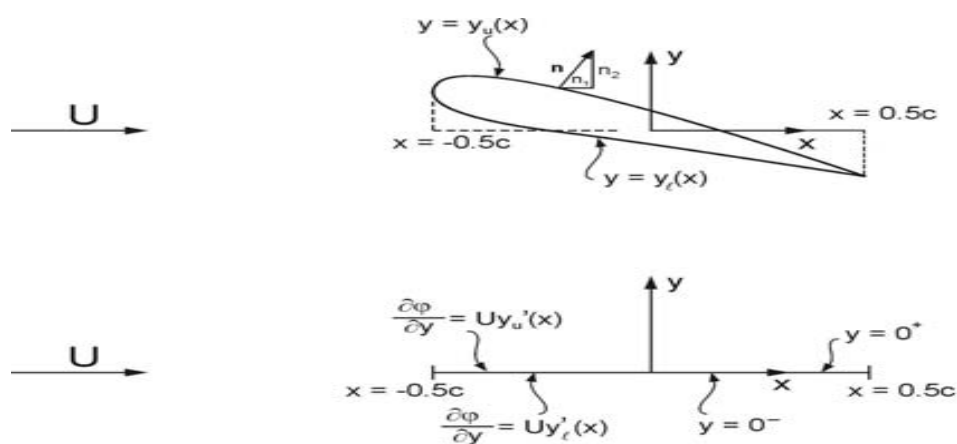


Figure 4.6 : Steady Potential Flow Past a Two Dimensional Infinite fluid [34]

The induced resistance and lift force for foil and strut can be described as:

$$R_i = 0.5C_{Di}\rho V^2 A \quad (4.26)$$

$$C_{Di} = \frac{4\pi\alpha\Lambda}{(\Lambda+2)^2} \quad (4.27)$$

$$L_i = 0.5C_{Li}\rho V^2 A \quad (4.28)$$

$$C_{Li} = \Lambda \frac{2\pi\alpha}{2+\sqrt{\Lambda^2+4}} \quad (4.29)$$

Where:

C_{Di} = Induced resistance coefficient

C_{Li} = Lift coefficient

$\Lambda =$ Aspect ratio, $\left(\frac{4s}{\pi c_0}\right)$

$\alpha =$ Angle of attack

$V =$ Ship speed

$c_0 =$ Chord length at midspan

$A =$ Planform area (projected area of the elliptical foil, $\left(\frac{s^2}{\Lambda}\right)$)

$s =$ Span length.

$\rho = 1025 \text{ kg/m}^3$

4.6.3 Wave Resistance

The wave resistance is caused by the lift and thickness of the foil or strut itself. The wave resistance equation for vertical hull surface i.e. strut can be expressed by Tuck Parabolic Strut [35] theory, as follows:

$$\frac{y}{L} = \frac{B}{2L} \left(1 - \left(\frac{x}{0.5L}\right)^2\right) \quad (4.30)$$

When $y \geq 0$

Then, the complex wave amplitude function $A(\theta)$ for waves with propagation direction θ can relate to the wave resistance by Newman [36] according to Michell [37] thin ship theory. The equation $A(\theta)$ can be expressed as:

$$A(\theta) = \frac{2}{\pi} v \sec^2 \theta X \iint \frac{\partial \eta}{\partial x} e^{[v \sec^2 \theta (z + ix \cos \theta)]} \quad (4.31)$$

By integrating equation 4.31 in the z direction, $A(\theta)$ can be expressed as:

$$A(\theta) = \frac{2}{\pi} \sec \theta \int_{-L/2}^{L/2} \eta_x(x, 0) e^{iv \sec \theta x} [1 - e^{-v \sec^2 \theta D}] \quad (4.32)$$

$$R_w = \frac{1}{2} \pi \rho V^2 \int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} |A(\theta)|^2 \cos \theta^3 d\theta \quad (4.33)$$

However, $A(\theta)$ in term of $(1 - e^{-v \sec^2 \theta D})$ in equation 4.32 is dependence on the draft (D) of the strut ($Fn_D = \frac{V}{\sqrt{gD}}$). When transverse waves correspond to smaller

θ than the divergent waves, then $(1 - e^{-v \sec^2 \theta D}) = (e^{-\sec^2 \theta / Fn_D})$ on Fn_D matters for larger Fn_D for the transverse waves. According to equation (4.33), there has connection between $A(\theta)$ and wave resistance, therefore this is relevant Fn_D on the contribution from transverse and divergent waves to wave resistance. In that case, $A(\theta)$ is proportional to the B for a parabolic strut and gives result to the wave resistance that proportional to B^2 . These mean that, the nondimensional wave resistance can be expressed as:

$$\frac{R_w}{0.5\rho V^2 B^2} \quad (4.34)$$

Which is the nondimensional wave resistance not dependence on B but is a function of $\frac{V}{\sqrt{Lg}}$ and $\frac{V}{\sqrt{Dg}}$. Nevertheless, the nondimensional wave resistance will become negligible or in other word very small effect to the wave resistance when $Fn_c = \frac{V}{\sqrt{gc}} \geq 3$, based on the chord length c.

The wave resistance due to the foil also must be considered which is it gives significant effect to the total resistance at high speed. The wave resistance due the thickness and lift on a single foil can be illustrated as a thin flat foil at a submergence h below the mean free surface. The linearized lift force can be expressed as:

$$L = -\rho V \int_{-c/2}^{c/2} \gamma(\xi) d\xi \equiv -\rho V \Gamma \quad (4.35)$$

The wave resistance dependence on the submergence Froude number of the foil ($Fn_h = \frac{V}{\sqrt{gh}}$) and the prediction of wave resistance based on a linear body boundary condition and the linear free surface condition. Referring to Figure 4.7 gives illustration about the flow due to the foil by two vortices with opposite circulation Γ . One vortex has a center in the foil at distance $c/4$ from the leading edge and the other vortex has a center at the image point about the mean free surface. The figure also explanation about velocity increases on the suction side of the foil that result the circulation sign to the foil. Because the two vortices have opposite signs i.e. Γ , the rigid free surface condition is satisfied. According to the Weissinger

[38] approximation, the boundary condition has to satisfy only at one point, at distance $\frac{3}{4}$ from the leading edge. The image vortex effectively causes an increase in the angle of attack at this point. This means a higher lift coefficient, C_{Lw} value and also the closer image vortex that increase the C_{Lw} value.

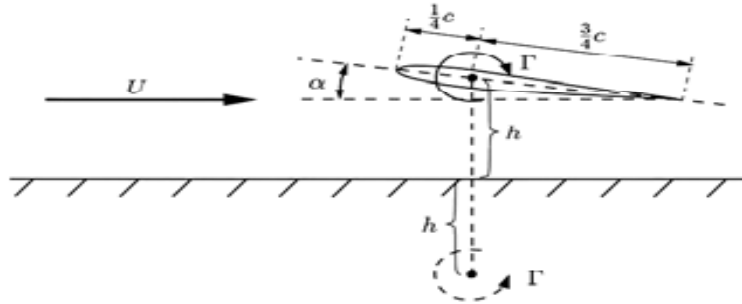


Figure 4.7 : The Flow due to the Foil By Two Vortices with Opposite Circulation Γ [34]

The circulation Γ of the foil can be calculated as follow:

$$\frac{\Gamma}{\pi c} - \frac{0.5c\Gamma}{2\pi(4h^2 + 0.25c^2)} = V \left(\alpha + \frac{2f}{c} \right) \quad (4.36)$$

Then the lift on the foil can be evaluated by using the Kutta-Joukowski formula below.

$$L_w = -\rho V \Gamma \quad (4.37)$$

and lift coefficient can be expressed as:

$$C_{Lw} = \frac{L_w}{0.5\rho V^2 c} \quad (4.38)$$

The wave resistance due to the foil can be described as:

$$R_w = \frac{\rho g \Gamma^2}{V^2} e^{\left(\frac{-2}{Fn_h^2} \right)} \quad (4.39)$$

and the wave resistance coefficient can be expressed as:

$$C_{Dw} = \frac{R_w}{0.5\rho V^2 c} \quad (4.40)$$

Where:

C_{Dw} = Wave resistance coefficient

C_{Lw} = Lift coefficient

f = Maximum camber

α = Angle of attack (deg)

V = Ship speed

c = Chord length

ρ = 1025 kg/m³

4.6.4 Spray Resistance

This situation is arising when the Froude number is larger approximately than 0.5 which the occurrence of the spray increases with the speed. The spray resistance is caused both by potential flow and viscous flow effect. Hoerner [39] simply set the spray resistance as follow:

$$R_s = 0.12\rho V^2 t^2 \quad (4.41)$$

This equation is valid when the Froude number based on the chord length ($Fn_c = \frac{V}{\sqrt{gc}}$) ≥ 3 . Where V is ship speed and t is thickness of the foil or strut.

4.7 The Combination Total Resistance

The total resistance for the vessel on planing hull taking account from resistance on barehull and the foil and strut. In this prediction of resistance, different methods are use in order to calculate the resistance. Savitsky method is use for prediction resistance of barehull while numerical method in two dimensional is use

for predict the resistance of foil and strut. The sample calculation of resistance prediction for model with stern foil is shown in Appendix C.

The formulation of total resistance can be expressed as:

$$R_T = R_{\text{barehull}}(\text{vessel}) + (R_v + R_i + R_w + R_s)_{\text{foil}} + (R_v + R_i + R_w + R_s)_{\text{strut}} \quad (4.42)$$

4.8 Sinkage and Trim

The hydrodynamic force put forth on ship's hull due to its forward motion causes forces which are capable of altering the attitude of the ship, inducing both a vertical movement or 'sinkage' (positive downward) and a rotation or 'trim' angle (positive when bow up). The combination of these two quantities is referred to as 'squat'.

Squat phenomenon is small for displacement vessels but it effect to small craft especially at high speed craft, involving vertical displacements of only at most a few percent of the vessel's draught at most speeds. Sinkage is generally positive at normal speeds, i.e. the ship's effective draught is increased. The positive sinkage reaches a maximum at Froude numbers of the order of 0.5, the reduction and may go negative at very high speeds, although the latter phenomenon has seldom being investigated for displacement vessels. However, at high speeds there is a connection between the squat phenomenon and the mechanism of planing, as a negative sinkage or rise in the water is induced by positive hydrodynamic lift. In that case however, for vessels capable of planing, the magnitude of squat is no longer small relative to the draught.

4.9 Program Development of Resistance Prediction

Basically computer programming is able to calculate the mathematical formulation when input data are specified by the user. In this case, by using FORTRAN (Formula Translation) program the resistance prediction can easily determine by specifying the formulation. In order to run the FORTRAN programme, it may have the source code to carry out this calculation. Before that, this program may require the input data of vessel. The input data are saved into notepad file *q.txt* and the input data contain of ship particular and foil strut parameters that have written in the program.

Basically the formula that used in the program resistance prediction for barehull is Savitsky Method and while for stern foil is 2D Method. The detail formulation of this program please refers to section 4.3 and 4.6 of this chapter. The result of calculation in this program was saved into *output.txt*. The source code, input and output data as illustrate in Appendix D.

4.10 Concluding Remarks

The resistance of the ship is able to predict by right method which must be decided at preliminary design stage. If incorrect or wrong method was selection then it will result the unenthusiastic answer either it give underestimate or overestimate calculation. In this case, the prediction performance using Savitsky method and 2D methods are used in order to calculate the resistance of the ship incorporating with stern foil.

CHAPTER 5

SEAWORTHINESS

5.1 Introduction

In the operation and design of high speed crafts, seakeeping performance is an important task because it has been proven that large motions and accelerations can degrade the operational capabilities of the ship. The high speed craft especially planing hull, sometimes exhibit an effect such as deck wetness, slamming and loss speed caused by the couple of heave and pitch. The vertical motions of high speed craft have negative consequences that limit the speed. It is not only a matter of structural damage or safety risks. When the frequencies of vertical motions are around 1 rad/sec effect by couple heave and pitch motion, they contribute in a cumulative way to sea-sickness of the crew.

From these research projects it became evident that the behavior of high speed craft in waves was strongly nonlinear by nature. This makes applying normal ship motion calculation routines and analysis procedures limited applicable. Also the limiting criteria for establishing the operability of these fast craft were found to be vastly different from those used for low speed displacement ships. From full scale observations it was evident the most of the speed reduction applied in fast craft operations in waves was voluntary. In order to improve the seaworthiness of the vessel, by it mean the vessel are able to move at maximum speed in rough condition, there are required equipment for reducing these motion. But this is much more difficult than for reducing roll because of the large pitching inertia moment of the vessel.

Various methods have been used to improve the seaworthiness of a boat, such as changing the main dimensions, choosing the type of lines and changing the distributing of weight. But the most effective way is by using a stabilizer. Recently stabilizing fins have been used to reduce roll. Heaving and pitching of boats brings on a series of effects, such as seasickness, deck wetness, slamming and speed loss. In order to reduce these effects, equipment for effectively reducing pitching is needed. But this is much more difficult than that for reducing roll, because of the very large pitching inertia moment of the boat. For a high speed planing boat, the pitching moment is relatively smaller, and a high speed moving hull can supply large dynamic forces, so it should be possible to install equipment for reducing pitch.

The improvement of seaworthiness in planing craft was studied by numerous researchers in reducing the motion. However, Savitsky and Brown (1976) [33] were initially derived the formulation for reducing the motion in heave and pitch at high speed by using wedge or flap in their experiment i.e. empirical method. In 1985, Long Wen [22] was performed tests with the planing craft model with transom flap for improved seakeeping ability for a single boat with a load coefficient of 0.54 over a range of speed coefficient from one to three. Other test he performed includes measuring the steady state normal force and hinge moment of transom flaps. In his research used the model without flap and with 12.5% flaps (flaps length or chord equal to 12.5% of the beam) with an angle of 0° and 3° , frequency response plot for the vessel on smooth water with oscillating flaps.

Unfortunately the hull form which Long Wen used was not a simple prismatic hull and the speed range used is on the low side for the equation used in developing the theoretical model. Figure 5.1 shows the hull model which Long Wen used in performing his test and figure 5.2 is the prismatic hull form which used for theoretical calculations. Long Wen's model has varying deadrise angles in front of 40% of the length, keel line curvature in front of 40% and varying beam along the complete length. The beam of Long Wen's model was 0.37m, the length was 1.5m and the mass was 27.3kg. For prismatic model used in calculations has constant beam equal to the mean beam of Long Wen's model and constant deadrise angle of 24° as the same as that for the last 60% of Long Wen's model.

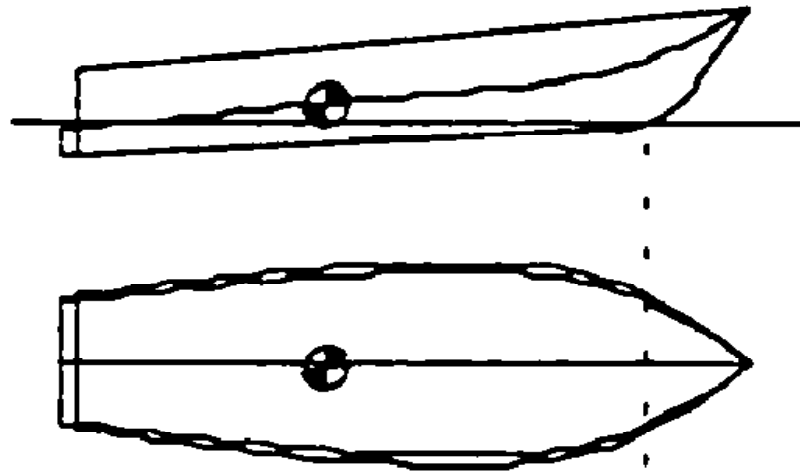


Figure 5.1 : Long Wen's Experiment Model [22]

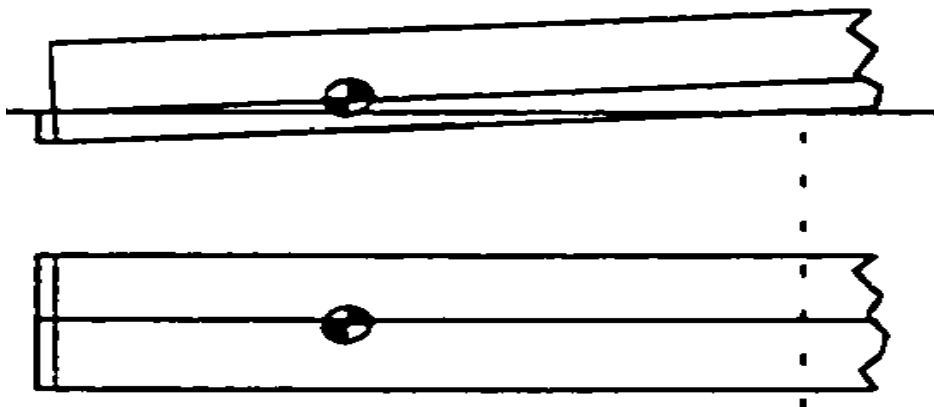


Figure 5.2 : Prismatic Hull Model [22]

In this research, the foil with strut is used in order to search out better seaworthiness especially for high speed craft (M Hull). Basically the idea of installation is obtain from hydrofoil vessel which it resulting positive in term of seakeeping performance. The advantageous of this foil or so called stern foil it able to minimize the vertical motion of heave and pitch that will be maintained the speed.

The ship in this research basically has its own direction of motion which has six degrees of freedom (6 D.O.F) and these motions are classified as movements of the centre of gravity. The motions of ship could be divided into two categories that

are translational and rotational. Figure 5.3 shows the complete set of ship motions (6 D.O.F) while table 5.1 describes ship motions term [40].

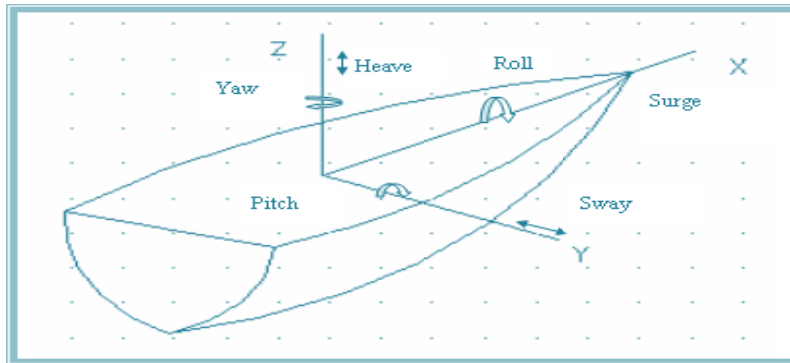


Figure 5.3 : Types of Ship Motions [40]

Table 5.1 : Ship's Six Degree of Freedom (6 D.O.F)

Movements	Description
Translational	Surge in the x direction of ship and acts forward
	Sway in the y direction of ship and move towards starboard
	Heave in the direction z axis and moving downwards
Rotational	Roll about x axis and makes the starboard side up and down
	Pitch about y axis which causes the bow up
	Yaw about z axis and the movements of ship is bow to starboard

The roll and pitch motion of ship are having dynamic equivalent of heel and trim. While in translational and rotational motion, there would be no residuary moment or force generates for sway, surge and yaw, provided the displacement is constant as long the ship is in neutral equilibrium condition.

Evaluation of a vessel's seakeeping performance depends heavily upon the environmental conditions (wave spectra) and the defined criteria, and this is the principal reason why any comparison for alternative vessel speeds, incident wave a

heading, loading conditions, etc is a complex matter. Seakeeping analysis is essentially a three part problem:

1. Estimation of the likely environmental conditions encountered by the vessel, based on hindcast or predicted weather data as applicable (e.g. wind, wind wave, swell).
2. Determination of the vessel's response characteristics (response amplitude operator, RAO).
3. Specification of the criteria being used to assess seakeeping behaviour (e.g. cargo acceleration, deck wetness, motions sickness).

Head sea waves are considered in this research which the RAO of heave refers to steady-state amplitude of heave divided by the incident wave amplitude in regular incident waves. Similarly, the RAO of pitch it refer to mean amplitude of pitch divided by the wave slope multiply with waves amplitude or regular waves.

5.2 Regular Waves

The regular waves or irregular waves are proportionally caused by wind with various speeds at sea surface. By right, the waves that always hit or cross the ship are considered as irregular waves and no regular waves could be generated at sea naturally. The waves that generally effect the ships characteristic might be in irregular waves or sometime in random.

The regular waves could not be generated in sea or ocean but this type of waves is taking into account during the seakeeping assessment. Most preferably this wave is known as an ideal. This ideal wave mentioned above is possible to be generated using towing tank facilities in laboratory. The experimental methods could be used to analyze ship model on seakeeping assessment. Other than this, there is another important fact by assumption where the theory of irregular waves can be

represented by superimposing or adding together a suitable assembly of regular waves.

The regular wave condition can be concluded that the characteristics of a ship at rough weather conditions could be determined by just referring to the regular waves criterion. The conditions or results are still acceptable although the ship does not encounter or face real situations at the sea. Therefore, an understanding of the theory and nature becomes important criteria to conduct seakeeping studies on any particular ships. An image of regular waves generated using towing tank facilities in laboratory is illustrated in figure 5.4. The waves are in two-dimensional and they travel ahead in x direction while the crests are 90° to x – axis [41].

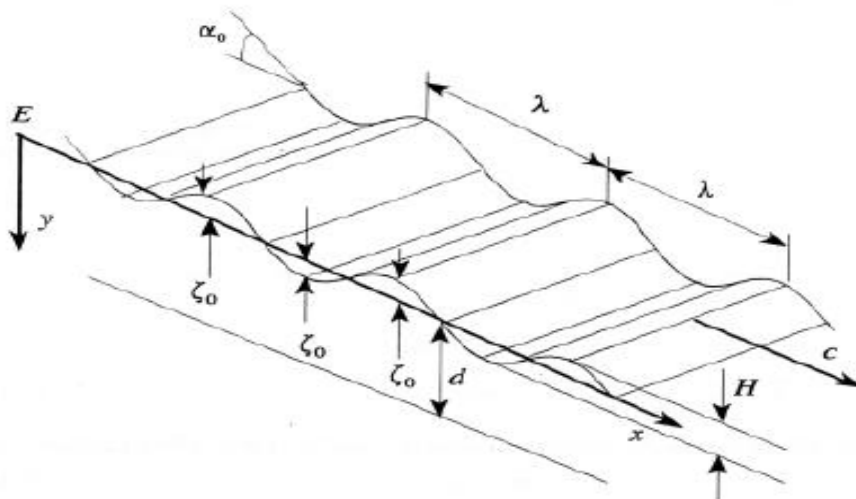


Figure 5.4 : Regular Waves Generated in Towing Tank [41]

5.3 Motion in Regular Waves

5.3.1 Lateral Plane Motion in Regular Beam Seas

The roll motion of a ship occurs when the starboard side is exposed to the regular (model test condition) or irregular (real sea condition) waves at beam angle 90° . Roll motion of a ship occurs about the x direction by rotation at starboard or

portside. Thus, the clear illustration is in figure 5.3. At this moment, the waves will be having resultant force and the force acting on a particle in the wave surface remains equal to the wave surface. In the case of wave length longer than the beam of ship, it is reasonable to assume that the ship is acted by a resultant force normal to an ‘effective wave surface’ which takes into account all the sub – surfaces interacting with the ship.

By the assumption made by R.E .Froude, the effective wave slope is subsurface occurs at the centre of buoyancy when rolls. Then, the general equation of motion for rolling in waves becomes [40]. However in this study only emphasizes on the vertical plane such as pitch and heave which gives much impact on high speed condition especially on planing craft (M Hull).

5.3.2 Vertical Plane Motion in Regular Head Waves [42]

Ship motion in regular head seas contributes motions such as heave, pitching and surge. At this stage, roll, yaw and sway are in the neutral condition and these three motions could not be analyzed. At the event of long waves, the encounter frequency, ω_e seems to be very low and at this moment, dynamic effects, added mass, and damping are virtually negligible. Thus, excitation of the ship is fully influenced by the buoyancy changes, provided the long waves. Figure 5.4 shows the maximum heave and pitch during the event in long waves [43]

In short waves, the excitation is not very significant and this occurs because in short waves, the buoyancy forces assist together with the ship hull. Here, it can be concluded that ships having significant excitation if the wave’s length is about three quarter of ship length.

5.4 Couple Heave and Pitch Motion in Head Sea

As described earlier, the pitch motion of a ship in the waves act in rotational motion about y - direction as shown in figure 5.3. It is always concentrates on head

seas where the heading angle will be 180° opposing the waves. The pitch motion normally having an encounter frequency due to its direction touches the successive wave crests [43]. The heave motion in regular waves is considered as movement of ship in vertical plane motion and the motion acts transitional. Since its buoyant force is then greater than its weight, the ship will move vertically up. Whilst the equilibrium is reached, the ship continue rising because of its momentum. Then the weight is greater than the buoyant force and it will tend to slow the motion. The process of a ship move vertically up and down will be continue indefinitely if there is no damping force acts in the opposite direction of motion and this known as a heaving. This condition occurs when the ship reach at extreme position at velocity is finally zero and the ship will move vertically downward since the weight is greater than buoyant force.

This couple of motion was dealing with vertical motion which can be investigated in a model basin. Actually the coupled heave and pitch for the head sea condition also can be investigated analytically by the strip theory method. Basically the strip theory was derived two equations for heave and pitch respectively. According to Newton's second law, at any instant all vertical forces on the ship are in dynamic equilibrium. Thus, the heaving and pitching equation can express as:

$$m\ddot{z} = \sum F \quad (5.1)$$

$$I\ddot{\theta} = \sum M \quad (5.2)$$

Where

$\sum F$ = The sum of various fluid forces (vertical hydrodynamic forces as well as the wave excitation force)

$\sum M$ = The sum of corresponding moments acting on the vessel because of relative motion of vessel and wave.

5.4.1 Basic Concept of Couple Heave and Pitch Motion

In order to simplified this complex motion problem several assumptions and limitations have been made as follows:

1. The ship must be heading into the waves in a direction transverse to their crest line.
2. The seaway is considered to consist of regular harmonic waves.
3. The ship motions of surge, sway, yaw and roll are neglected.

Korvin-Kroukovsky [44] approach was to consider a ship hull as a series of transverse segment or strip and to treat the flow adjacent to each as two dimensional in nature. The effect was to reduce the problem from three to two dimensions with each strip assumed to be a part of an infinite cylinder having two dimensional around it. The vertical motion of each strip is assumed to be composed of the combined pitching and heaving motions. When the response of each strip calculated, the total ship response can be found by integrating the component reactions of all the strips along the ship's length. The basic mathematical equation obtained in this way is a linear second order differential equation describing the inertial, hydrodynamic and hydrostatic components of force acting on each strip and caused by relative vertical motion between the ship and the wave.

The general equation coupled of heaving and pitching can be expressed as:

$$\begin{aligned} \left(\frac{\Delta}{g} + a_{zz}\right)\ddot{z} + b_{zz}\dot{z} + c_{zz}z + a_{z\theta}\ddot{\theta} + b_{z\theta}\dot{\theta} + c_{z\theta}\theta &= F_{(t)} \\ (I_{\theta\theta} + a_{\theta\theta})\ddot{\theta} + b_{\theta\theta}\dot{\theta} + c_{\theta\theta}\theta + a_{\theta z}\ddot{z} + b_{\theta z}\dot{z} + c_{\theta z}z &= M_{(t)} \end{aligned} \quad (5.3)$$

Which each term represents the moment of force instead of the force.

5.5 Calculation Method for Vertical Motions by effect of Stern Foil

Figure 5.5 show the planing craft (M Hull) was adapted with the stern foil in order to minimize the motion during cruising speed.



Figure 5.5 : The High Speed Craft (M Hull) incorporated with Stern Foil

For determining the effect of stern foil, calculating the coupling movement of heave and pitch in regular waves, the effect of foil on the vessel is considered as a small perturbation, and the motion of planing hull vessel with stern foil in waves may be described by the following equation:

$$\left(\frac{\Delta}{g} + a_{zz}\right)\ddot{z} + b_{zz}\dot{z} + c_{zz}z + a_{z\theta}\ddot{\theta} + b_{z\theta}\dot{\theta} + c_{z\theta}\theta = F_{(t)} + \Delta F$$

$$(I_{\theta\theta} + a_{\theta\theta})\ddot{\theta} + b_{\theta\theta}\dot{\theta} + c_{\theta\theta}\theta + a_{\theta z}\ddot{z} + b_{\theta z}\dot{z} + c_{\theta z}z = M_{(t)} + \Delta M \quad (5.4)$$

Where,

\ddot{z}, \dot{z}, z = heave acceleration, velocity and displacement, respectively.

$\ddot{\theta}, \dot{\theta}, \theta$ = pitch angular acceleration, velocity and displacement respectively.

Δ/g = mass of vessel.

I_{yy} = pitch inertia moment of vessel.

$a_{zz}, b_{zz}, c_{zz}, a_{z\theta}, b_{z\theta}, c_{z\theta}, a_{\theta\theta}, b_{\theta\theta}, c_{\theta\theta}, a_{\theta z}, b_{\theta z}, c_{\theta z}$ = stability derivatives

ΔF = Foil excited force

ΔM = Foil excited moment

In this study, the calculation used is strip theory method for the hull dynamics and for estimating the values of ΔF and ΔM or calculating the stern foil by using Savitsky and Brown's empirical plus two dimensional methods. The equation for calculating forces and moments of stern foil by using empirical method will be added to the dynamics equation in order to suit the characteristic and study objectives. The selection of stern foil parameter has been decided according to its resistance increment due to the seakeeping performance. The parameters of the stern foil was designed will be described in Chapter IV.

5.5.1 Exciting Forces and Moments due to Stern Foil for Planing Hull

The total lift force can be expressed as:

$$\Delta L_t = L_v + L_i + L_w \quad (5.5)$$

The equation for lift force due to viscous drag as follows:

$$L_v = 0.5C_{L_v}\rho V^2 A \quad (5.6)$$

$$C_{L_v} = 2\pi(\alpha - \alpha_0) \quad (5.7)$$

Where:

C_{L_v} = Lift coefficient due to Viscous

α = Angle of attack in radian

V = Ship speed

α_0 = Uncamber foil and zero flap=0

A = Chord length (c)

$\rho = 1025 \text{ kg/m}^3$

While for lift force due to induced drag can be expressed as:

$$L_i = 0.5C_{L_i}\rho V^2 A \quad (5.8)$$

$$C_{Li} = \Lambda \frac{2\pi\alpha}{2 + \sqrt{\Lambda^2 + 4}} \quad (5.9)$$

Where:

C_{Li} = Lift coefficient due to Induced Drag

Λ = Aspect ratio, $\left(\frac{4s}{\pi c_0}\right)$

α = Angle of attack

V = Ship speed

c_0 = Chord length at midspan

A = Planform area (projected area of the elliptical foil, $\left(\frac{s^2}{\Lambda}\right)$)

s = Span length.

$\rho = 1025 \text{ kg/m}^3$

Then the lift force due to wave resistance can be described as:

$$\frac{\Gamma}{\pi c} - \frac{0.5c\Gamma}{2\pi(4h^2 + 0.25c^2)} = V \left(\alpha + \frac{2f}{c} \right) \quad (5.10)$$

$$L_w = -\rho V \Gamma \quad (5.11)$$

and lift coefficient can be expressed as:

$$C_{Lw} = \frac{L_w}{0.5\rho V^2 c} \quad (5.12)$$

According to large aspect-ratio theory of wing in figure 5.6, the lift of a stern foil on planing craft is:

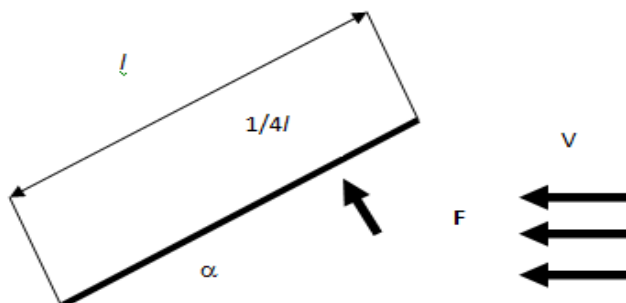


Figure 5.6 : Theory of Wing

The detailed expression of lift can be obtained as follows:

$$\Delta F = L_v + L_i + L_w \quad (5.13)$$

In this equation the moment about the center of gravity is,

$$\Delta M_G = \Delta F l_a \quad (5.14)$$

Where effective length l_a of stern foil to center of gravity as shows in figure 5.7 can be expressed as follows:

$$l_a = \frac{1}{4}c + \frac{zg+d}{\sin \alpha} \quad (5.15)$$

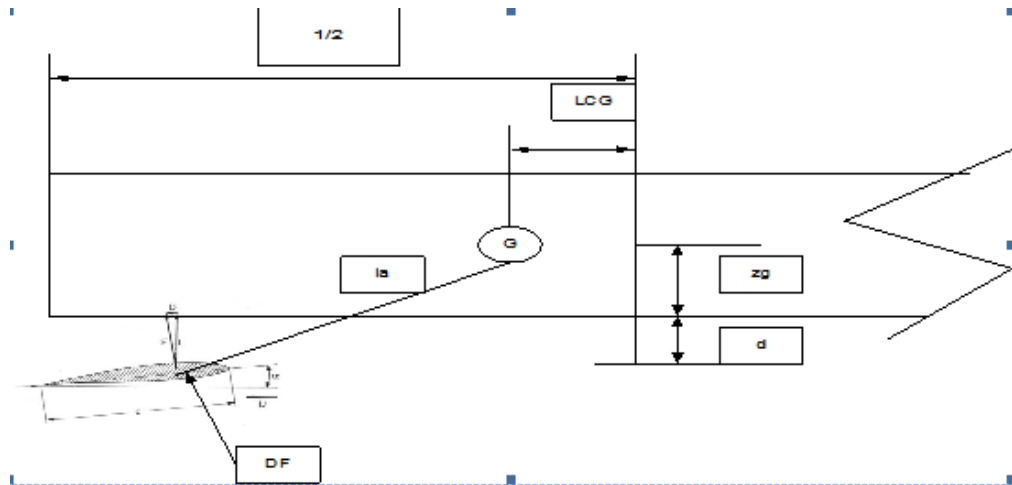


Figure 5.7 : Effective Length of Stern Foil

5.5.2 Solution of the Motion Equation with Stern Foil

Substituting equation (5.13) and equation (5.14) into equation (5.16) the motion of planing craft with stern foil in waves may be described by the following equation:

$$\begin{aligned} \left(\frac{\Delta}{g} + a_{zz}\right)\ddot{z} + b_{zz}\dot{z} + c_{zz}z + a_{z\theta}\ddot{\theta} + b_{z\theta}\dot{\theta} + c_{z\theta}\theta + \Delta F &= F_{(t)} \\ (I_{\theta\theta} + a_{\theta\theta})\ddot{\theta} + b_{\theta\theta}\dot{\theta} + c_{\theta\theta}\theta + a_{\theta z}\ddot{z} + b_{\theta z}\dot{z} + c_{\theta z}z + \Delta F l_a &= M_{(t)} \end{aligned} \quad (5.16)$$

In its simplest form, the vessel may be considered like an electronic filter. It takes an input signal (the ocean waves), filters it, and produces an output (the vessel motions). In most cases, this simple method is quite valid and produces useful results. The vessel's filter function which is also named Response Amplitude Operators (*RAO*) are different for the six, rigid-body degrees freedom of ship motions. Each motion has its own characteristics and *RAO*.

RAO are derived from the amplitudes by dividing the motion response by the wave input parameter:

$$RAO = \frac{\text{Motion Response}}{\text{Wave Input Parameters}}$$

The *RAO* for pitching and heaving are defined as below:

Pitching *RAO* = Pitch Amplitude / Wave Slope

$$= \frac{\theta_a}{k\zeta_a}$$

Heave *RAO* = Heave Amplitude / Wave Amplitude

$$= \frac{z_a}{\zeta_a}$$

5.6 SEAKEEPER Program [26]

SEAKEEPER is the seakeeping analysis program in the Maxsurf software package. It uses the Maxsurf geometry file to calculate the response of the vessel to user-defined sea conditions. In order to calculate the vessel response, linear strip theory based on the work of Salvesen et al (1970), is used to calculate the coupled heave and pitch response of the vessel. The roll response is calculated using linear roll damping theory. In addition to graphical and tabular output of numerical results

data, SEAKEEPER is also able to provide an animation of the vessel's response to specified sea conditions. Strip theory, and SEAKEEPER, is able to provide reasonably accurate seakeeping predictions particularly for a relatively wide range of vessel types. The speed of the analysis and its integration into the rest of the Maxsurf suite, make SEAKEEPER particularly useful in initial design stage. For a greater insight into the accuracy that can be achieved by SEAKEEPER.

The SEAKEEPER computational method, an application which may be used to predict the motion and seakeeping performance of vessels designed using Maxsurf. The outlines of fundamental principles used in seakeeping analysis are follows:

5.6.1 Coordinate System

The coordinate system used by SEAKEEPER is the same as for Maxsurf. The coordinate system has the origin at the user defined zero point as show in figure 5.8.

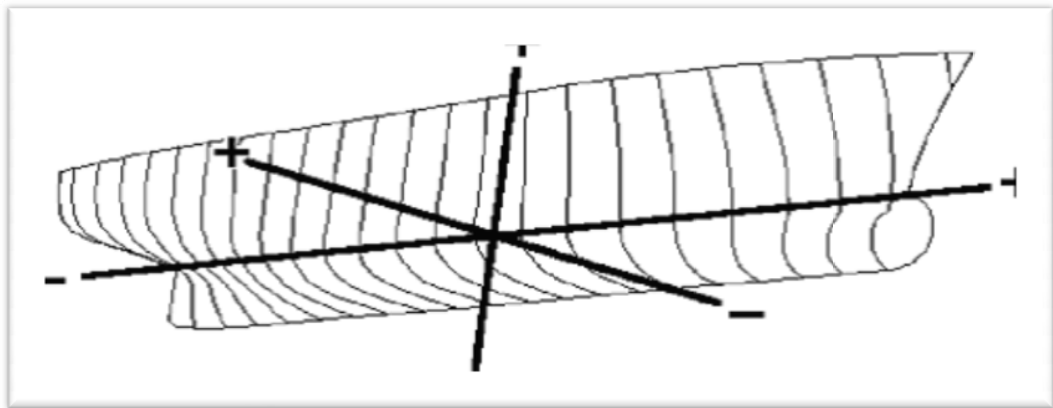


Figure 5.8 : Coordinate System in SEAKEEPER Program [26]

The coordinate that used for SEAKEEPER User Coordinate System similar to view Windows, which explain below:

1. X-axis +ve forward -ve aft
2. Y-axis +ve starboard -ve port

3. Z-axis +ve up -ve down

When calculating motions at remote locations, the vessel is assumed to rotate about the centre of gravity. Hence the distance of the remote location from the centre of gravity is of interest. SEAKEEPER calculates this distance internally and all positions are measured in the coordinate system described above. A vessel has six degrees of freedom, three linear and three angular. These are: surge, sway, heave (linear motions in x, y, z axes respectively) and roll, pitch, yaw (angular motions about the x, y, z axes respectively). For convenience, the degrees of freedom are often given the subscripts 1 to 6; thus heave motion would have a subscript 3 and pitch 5.

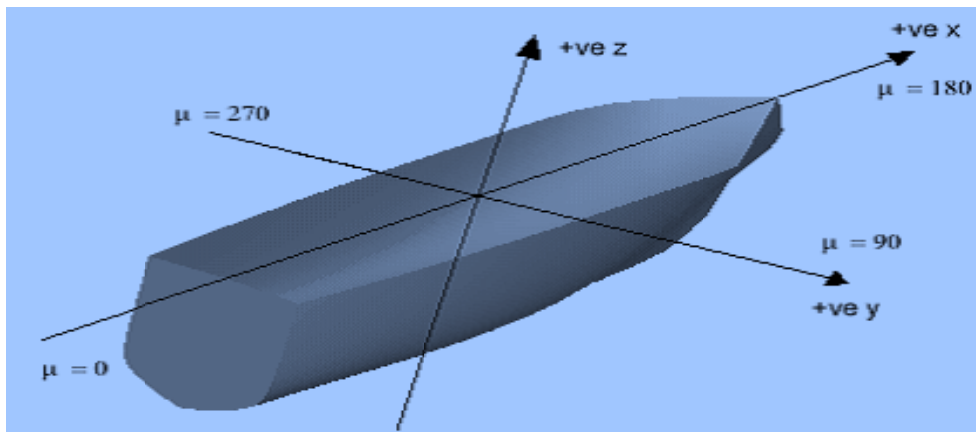


Figure 5.9 : Wave Directions in SEAKEEPER System [26]

Wave direction is measured relative to the vessel track and is given the symbol μ . Thus following waves are at $\mu = 0^\circ$; starboard beam seas are 90° ; head seas 180° and port beam seas 270° as shown in figure 5.9.

5.6.2 Wave Spectra

Irregular ocean waves are often characterised by a "wave spectrum", this describes the distribution of wave energy (height) with frequency. Characterisation Ocean waves are often characterised by statistical analysis of the time history of the irregular waves.

Irregular ocean waves are typically described in terms of a wave spectrum. This describes a wave energy distribution as a function of wave frequency. The continuous frequency domain representation shows the power density variation of the waves with frequency and is known as the wave amplitude energy density spectrum, or more commonly referred to as the wave energy spectrum. The spectral ordinates (or wave spectral density) are given the symbol: $S_{\zeta}(\omega)$. (This is similar to the power spectral density, PSD, used in electronics and communications analysis.) A typical wave spectrum is shown in figure 5.10 below:

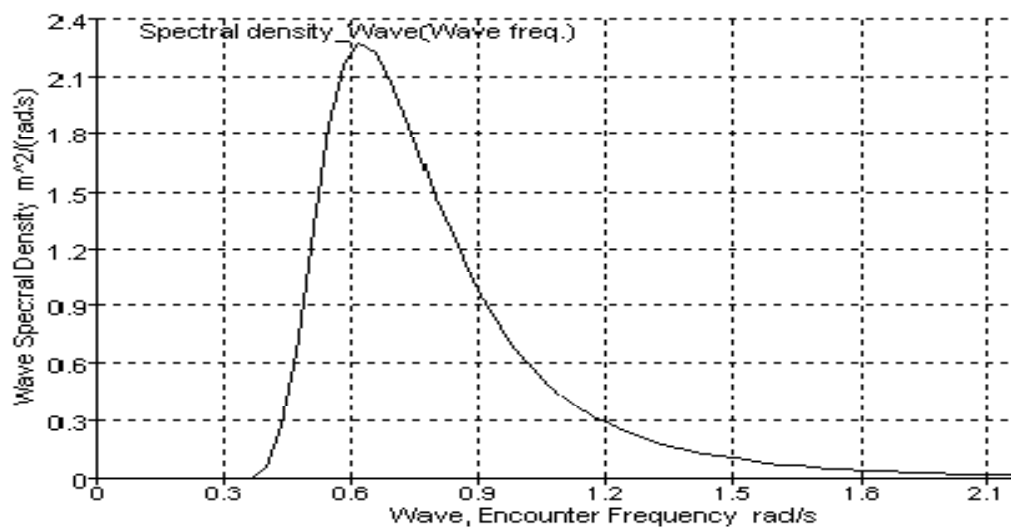


Figure 5.10 : Typical Wave Spectrums [26]

5.6.3 Idealised Spectra

It is often useful to define idealised wave spectra which broadly represent the characteristics of real wave energy spectra. Several such idealised spectra are available in SEAKEEPER and are described below:

1. Bretschneider or ITTC two parameter spectrum.
2. One parameter Bretschneider.
3. JONSWAP.
4. DNV Spectrum.
5. Pierson Moskowitz.

5.6.4 Encounter Spectrum

An important concept when calculating vessel motions is that of the encountered wave spectrum. This is a transformation of the wave spectrum which describes the waves encountered by a vessel travelling through the ocean at a certain speed. This is effectively a Doppler shift of the spectrum which smears the spectrum towards the higher frequencies in head seas and towards the lower frequency in following seas.

5.6.5 Characterising Vessel Response

This section outlines the method used to describe a vessel's response in a seaway. Harmonic Response of Damped, Spring, Mass system for most purposes, it is sufficient to model the vessel as a set of coupled spring, mass, damper systems undergoing simple harmonic motion. This is assumed by SEAKEEPER and most other seakeeping prediction methods. This method may be successfully applied to the analysis of the vessel's motions provided that these motions are linear and that the principle of superposition holds. These assumptions are valid provided that the vessel is not experiencing extremely severe conditions.

5.6.6 Response Amplitude Operator (RAO)

The Response Amplitude Operator (RAO), also referred to as a transfer function (this is similar to the response curve of an electronic filter), describes how the response of the vessel varies with frequency. These are normally non-dimensionalised with wave height or wave slope. Typical heave and pitch RAOs are shown in figure 5.11 below:

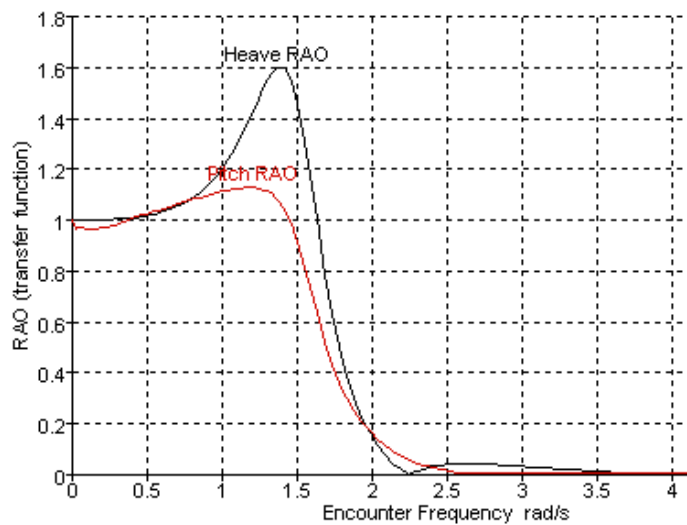


Figure 5.11 : Typical Heave and Pitch RAO's [26]

It may be seen that the RAOs tend to unity at low frequency, this is where the vessel simply moves up and down with the wave and acts like a cork. At high frequency, the response tends to zero since the effect of many very short waves cancel out over the length of the vessel. Typically the vessel will also have a peak of greater than unity; this occurs close to the vessel's natural period. The peak is due to resonance. An RAO value of greater than unity indicates that the vessel's response is greater than the wave amplitude (or slope).

5.6.7 Calculating Vessel Motions

Assuming linearity, the vessel's RAOs depend only on the vessel's geometry, speed and heading. Thus once the RAOs have been calculated the motion of a vessel in a particular sea state of interest may be calculated. It is hence possible to obtain a spectrum for a particular vessel motion in a particular sea spectrum:

5.7 Concluding Remarks

An engineering tool for calculating hydrodynamic forces on the foil systems and for modeling boat dynamics is being developed. It will be used for designing

hydrofoil-assisted craft, which SEAKEEPER is used in this study. This relatively simple approach is suitable for parametric studies of the influence of foil elements on hydrodynamic seakeeping performance. Stern foil sections that improve boat performance will also be incorporated into the tool. Planing hull elements will be added to model transitional regimes of pure hydrofoil boats and service regimes of hydrofoil assisted ships. The theoretical in seakeeping also important in order to cross check the seakeeping prediction.

CHAPTER 6

RESEARCH OBJECT

6.1 Introduction

The research to be done is intended on the resistance and seakeeping test and is conducted in the Marine Technology Laboratory, Universiti Teknologi Malaysia. In this case, the research will be conducted on resistance in calm water and seakeeping test for planing craft at speed 25 knots with and without stern foil in regular waves. As previously stated these models are based on full sized ships. When expanding the model results to full scale, full scale use was made of the ITTC 57 friction line as well as applying a ship-model correlation allowance of 0.0006981. A summary of the ship and model including stern foil parameters features are shown below in table 6.1 and table 6.2. This craft was designed for high speed i.e. 25 knot at maximum continuous speed. The body plan with and without stern foils of the vessel are shown in figure 6.1 and figure 6.2 respectively. The offset data of vessel is shown in Appendix E. Figure 6.3 and 6.4 shows the resistance tests which were conducted for both conditions with and without stern foil at various speeds.

Seakeeping test is meant to measure the behaviour of the vessel at seas. Model testing provides an attractive alternative. The seakeeping test can predict the characteristics of the vessel in term of motion which is it gives significant effect on its dynamics stability. The success of a ship design depends ultimately on its performance in a seaway. However, in a realistic seaway it is such a complex problem that ship designers are generally forced to select their hull forms and ship

dimensions on the basis of calm water performance without much consideration of the sea and the weather conditions. Figure 6.5 and 6.6 show seakeeping test which was conducted according to the test protocol in order to determine the RAO for heave and pitch.

Table 6.1 : Main Particular of Planing Craft (M Hull)

Parameters	Ship	Model
Length overall L_{OA}	34	2.194
Length of waterline L_{WL} (m)	31.667	2.043
Waterline beam (m)	7.4167	0.4785
Design Draught T(m)	1.365	0.0881
Depth (m)	3.300	0.2129
Deadrise angle β (deg)	14	14
Midship area coefficient, C_m	0.695	0.695
Block coefficient, C_b	0.46	0.46
Waterplane area coefficient, C_{wp}	0.817	0.817
Wetted Surface Area (m^2)	220.000	0.9157
Displacement, Δ (Tonne)	130.275	0.0341
LCG from Midship (m)	-2.258	-0.1457
VCG from keel (m)	2.124	0.1370
Linear scale ratio, λ	1:15.5	

Table 6.2 : Stern Foil Parameters

	Model	Prototype
Foil		
Height (h) m	0.071	1.1005
Chord Length (c) m	0.065	1.0000
Span (s) m	0.452	7.0000
Thickness (t) m	0.008	0.1194
Angle of Attack deg	0.000	0.0000
<i>Camber (f)</i>	0.000	0.0000

	Model	Prototype
Strut		
Height (h) m	0.071	1.1005
Chord Length (c) m	0.021	0.3255
Thickness (t) m	0.006	0.0930
Angle of Attack deg	0.000	0.0000
<i>Camber (f)</i>	0.000	0.0000

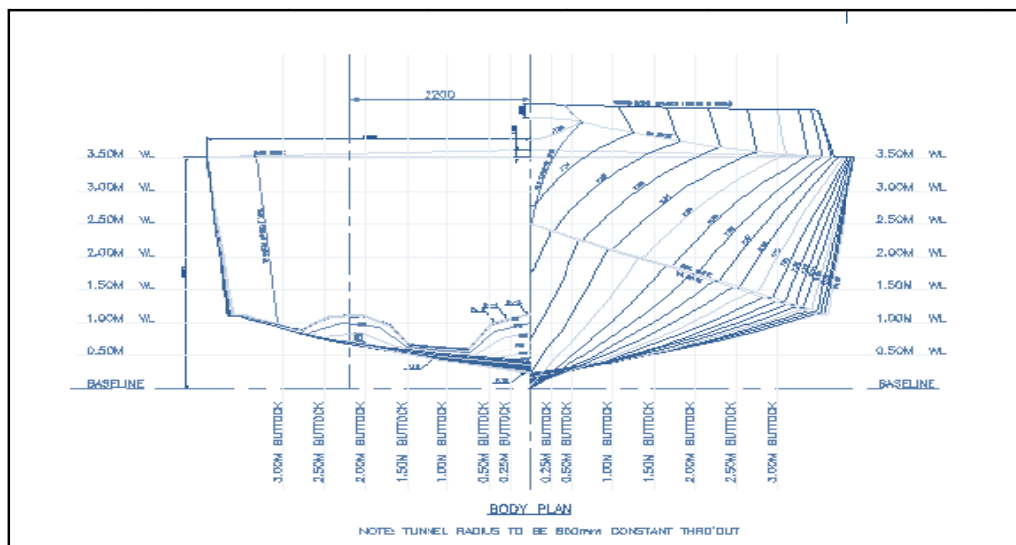


Figure 6.1 : Body Plan of Planing Craft (M Hull) without Stern Foil

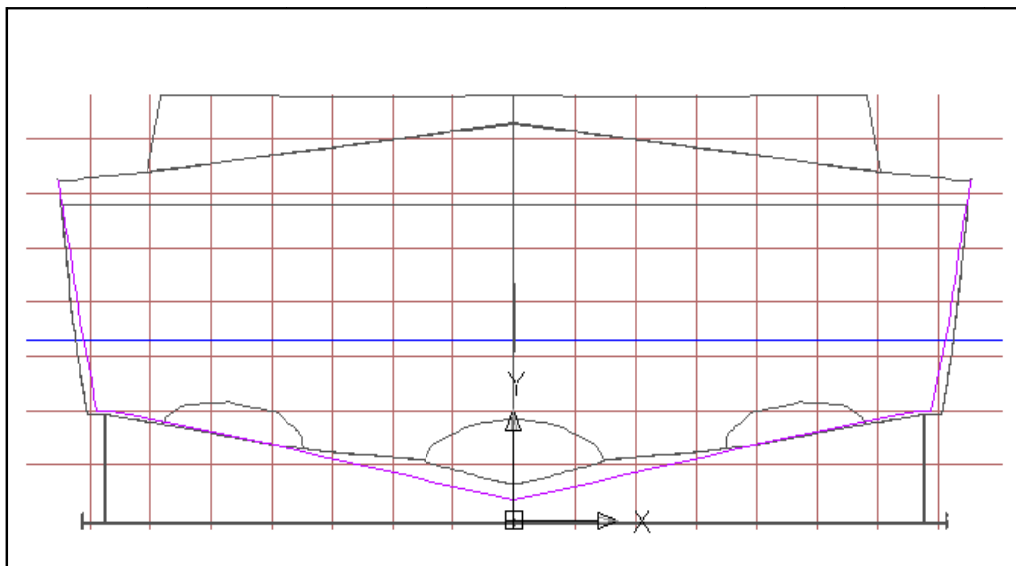


Figure 6.2 : Body Plan of Planing Craft (M Hull) with Stern Foil

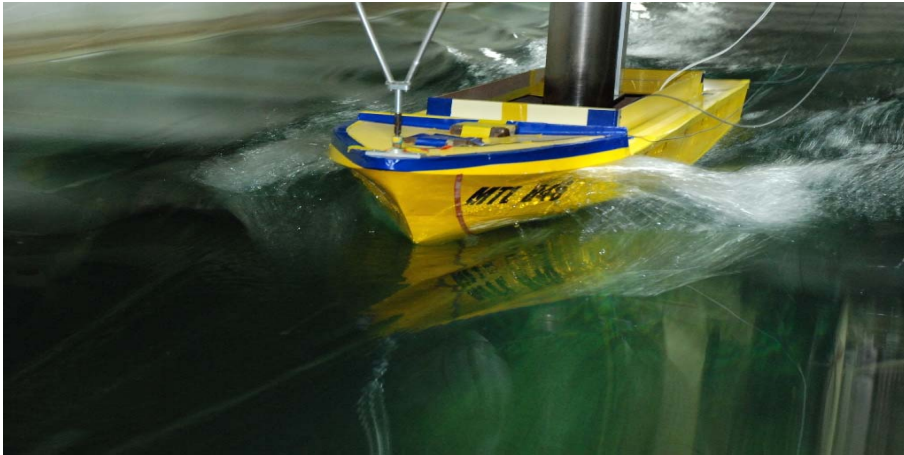


Figure 6.3 : Resistance test without Stern Foil



Figure 6.4 : Resistance test with Stern Foil



Figure 6.5 : Wave Contour in Seakeeping Test



Figure 6.6 : Seakeeping Test in Progress

6.2 Concluding Remarks

According to Savitsky formulation for a ship that is fully planing, the ship will be categorized as a high speed craft when $C_v > 1.5$ or $F_{nb} > 1.5$. Based on the calculation, this model is run at speed 25 knots which is equivalent to $C_v = 1.507$, meaning that the model meet the criteria of high speed craft.

CHAPTER 7

ANALYSIS

7.1 Introduction

In this chapter, the research analysis are divided into two part i.e. resistance and seakeeping analysis which is consist of performance prediction of planing craft (M Hull). This research concentrates on using stern foil to improved the heave and pitch motion in seaway meaning that the vessel is able to operate at high speed in certain level of sea state code. Besides that, comparison in term of heave and pitch motion and resistance are made between with and without stern foil. Basically, there are advantages by incorporating the stern foil at aft of the vessel which it gives significant effect to motion reduction. Figure 7.1 show the vessel which has been modified by adapting the stern foil located at aft portion.

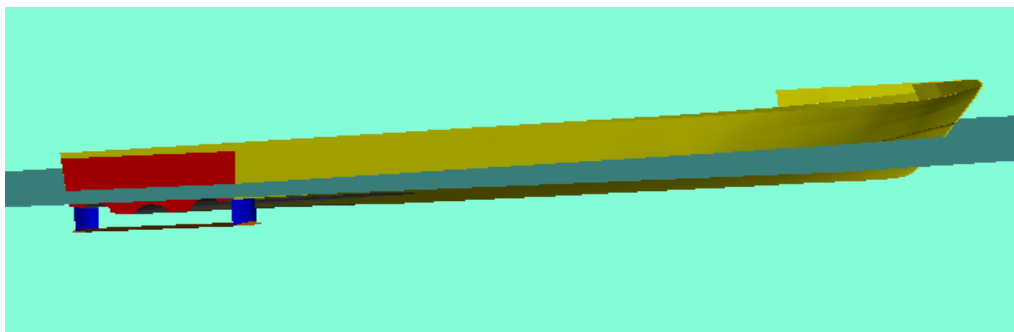


Figure 7.1 : The Vessel installed with Stern Foil.

7.2 Resistance Analysis

An example of resistance calculation using Savitsky Method and 2D formulations are shown in Appendix C. While the source code and input data for calculation using FORTRAN program is shown in Appendix D which consist of input data, source code and output data. In order to validate the resistance prediction, the resistance test was conducted for both models (with and without stern foil) where the comparisons can determine either the formula are appropriate or not.

Table 7.1 and figure 7.2 shows the comparison between theoretical and experimental resistance result for a ship without incorporating stern foil at calm water condition. From the result it show that the variance of resistance values between theory and experiment. The variance resistance value between both methods is 37.89%, at speed 23 knots which is calculation using theoretical (Savitsky Method) resulting high value compare to the resistance value that was carryout by experiment. However, this value is large due to linearized formulation by Savitsky equation. At speed 26 knots the variance of resistance between both methods are 15.62%, where the result of resistance test from experiment closely agreed to the resistance value obtained by theory (Savitsky Method). The theory prediction resulting better performance at speed 28 knots where the discrepancy value is 5.32%.

Table 7.1 : Resistance Result for a ship without Stern Foil

Vs(knots)	Resistance (N)	
	Theoretical	Experiment
23	95346.63	59244.09
24	99696.57	66787.36
25	104081.14	77299.11
26	108480.54	91535.98
28	117240.33	123473.10
30	125808.06	141895.51

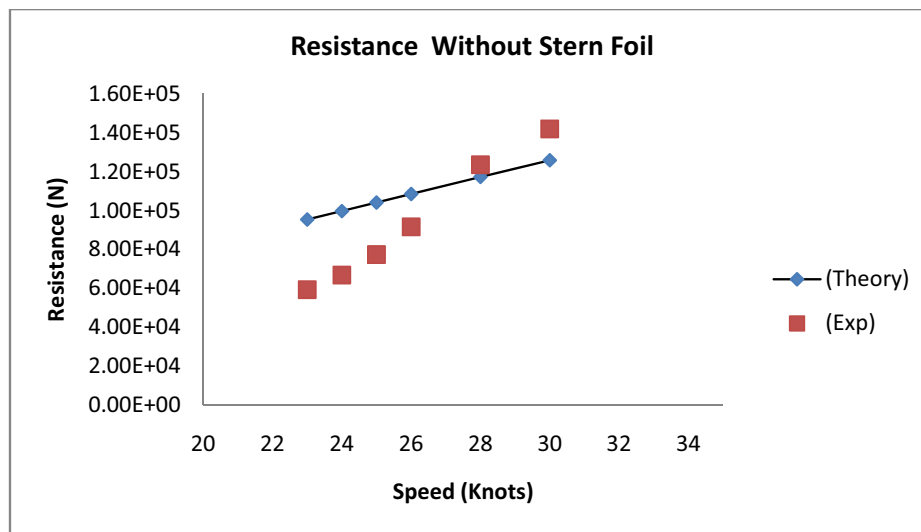


Figure 7.2 : Graphs on Comparison between Theory and Experiment for Resistance without Stern Foil

From table 7.2 and figure 7.3 shows the resistance value for both theoretical and experiment methods for a ship incorporating with stern foil at 0° degree angle of attack. The result between theoretical and experiment resulting small variance where the percentage ranges for every speed are at 7-20%. From this table and graph the resistance values are slightly higher when compare to theoretical prediction.

Table 7.2 : Resistance Result for a ship with Stern Foil (0°)

Vs(knots)	Resistance (N)	
	Theoretical (N)	Experiment (N)
23	97063.33	104980
24	101556.01	110070
25	106088.69	125860
26	110641.57	140620
28	119724.28	144670
30	128636.11	154240

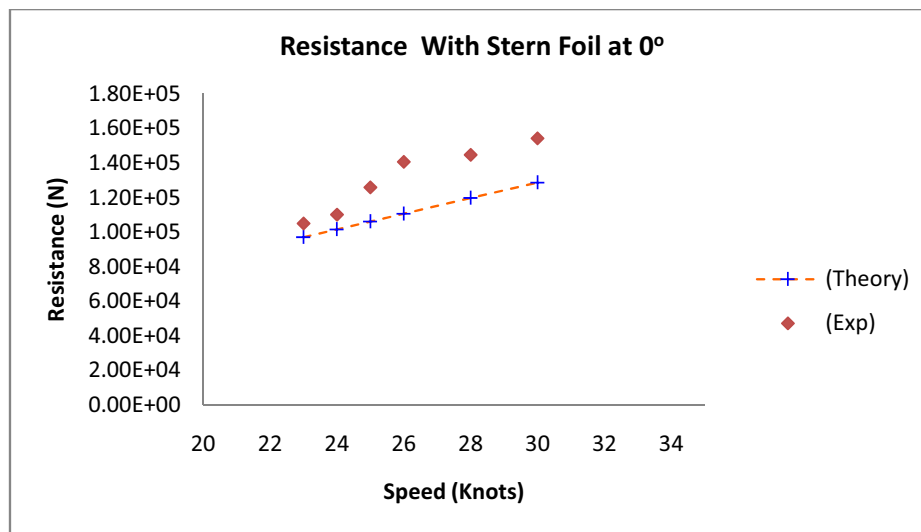


Figure 7.3 : Graphs on Comparison between Theory and Experiment for Resistance with Stern Foil at 0° angle of attack

Table 7.3 and figure 7.4 shows the resistance value for both theoretical and experiment methods for a ship incorporating with stern foil at 3° degree angle of attack. The result comparison between theoretical and experiment resulting quite large variance which is the percentage ranges for every speed are at 26-41% where the largest value of resistance discrepancy is obtained at 30 knots.

Table 7.3 : Resistance Result for a ship with Stern Foil (3°)

Vs(knots)	Resistance (N)	
	Theoretical (N)	Experiment (N)
23	116742.67	158750
24	122964.15	178210
25	129298.71	196920
26	135726.59	208870
28	148778.83	220440
30	161953.10	277410

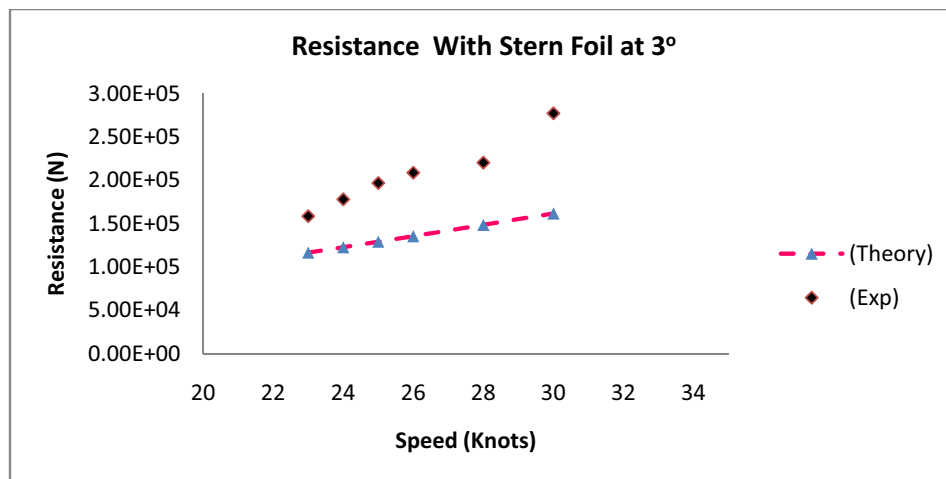


Figure 7.4 : Graphs on Comparison between Theory and Experiment for Resistance with Stern Foil at 3° angle of attack

From figure 7.5, it show that the values of resistance nearly similar for theoretical prediction (Savitsky and 2D Method) and Hullspeed program for resistance prediction with and without stern foil.

Table 7.4 : Resistance Result for a Ship with and without Stern Foil (Hull Speed Program)

Resistance Without Stern Foil			Resistance With Stern Foil (0°)		
Hullspeed Program			Hullspeed Program		
Vs(knots)	Resistance (kN)	Resistance (N)	Vs(knots)	Resistance (kN)	Resistance (N)
23	98.993017	98993.02	23	102.04	102035.19
24	103.041826	103041.83	24	106.15	106152.40
25	107.138408	107138.41	25	110.32	110318.51
26	111.264434	111264.43	26	114.51	114514.30
27	115.399866	115399.87	27	118.72	118718.68
28	119.523182	119523.18	28	122.91	122908.90
29	123.611827	123611.83	29	127.06	127061.05
30	127.642912	127642.91	30	131.15	131150.83

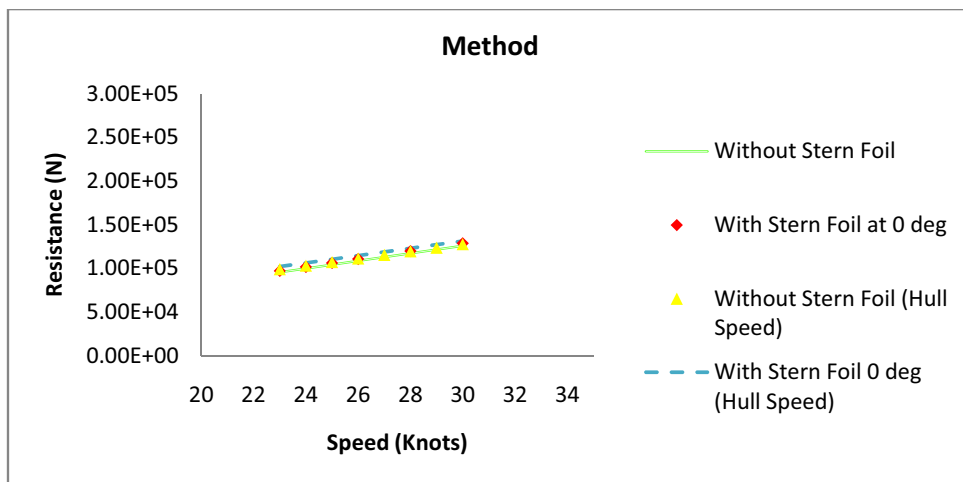


Figure 7.5 : Graphs on Comparison between Ship Resistance for with and without Stern Foil (Savitsky Method and Hull Speed Program)

The resistance value in figure 7.6 which is obtain from experiment showing a quite large variance at different test condition. Figure 7.7 show that the increasing pattern of resistance value between ship with stern foil at 0° and ship without stern foil is smaller due to speed increasing.

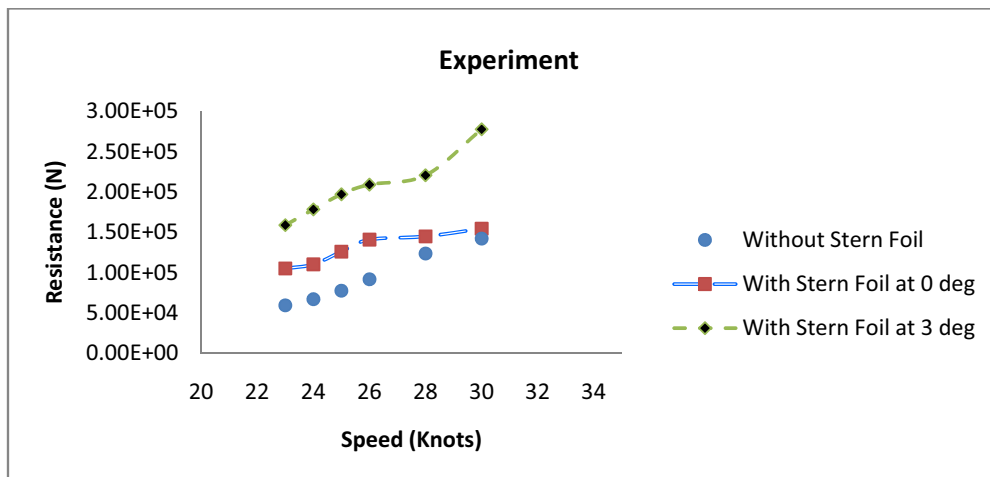


Figure 7.6 : Graphs on Comparison of Resistance between a ship with and without Stern Foil (Exp)

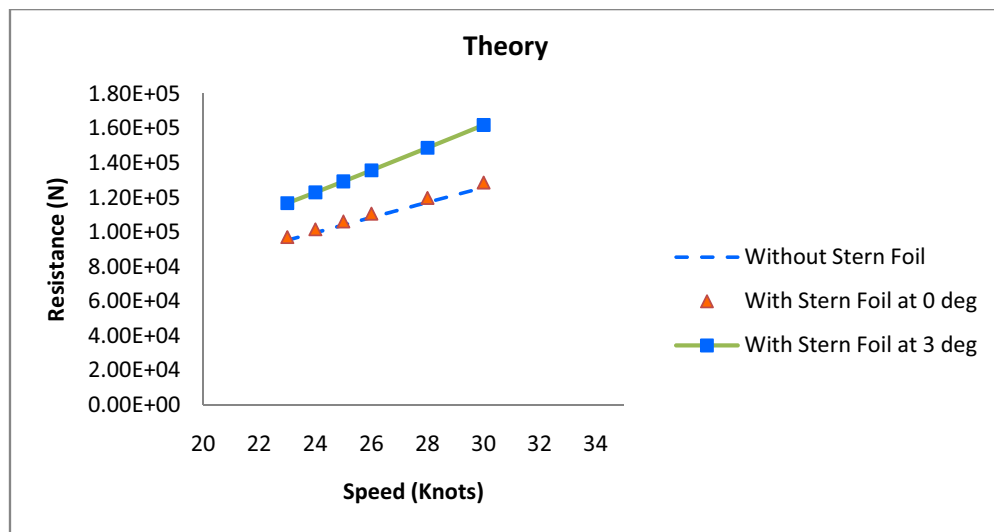


Figure 7.7 : Graphs on Comparison of Resistance between a ship with and without Stern Foil (Theory)

The values of sinkage and trim result test in calm water condition at various speeds are presented in table 7.5. While in figure 7.8 show the graph of sinkage at different test condition. Every condition of sinkage gives inconsistent values at every speed. At condition of ship with 3° angle of attack, the sinkage becomes negative values. This condition also occurs to the trim as shown in figure 7.9.

Normally sinkage is positive at normal speeds, i.e. the ship's effective draught is increased. The positive sinkage reaches a maximum at Froude numbers of the order of 0.5, which is this reduction may achieve a negative value at very high speeds. However, at high speeds there is a connection between the squat phenomenon and the mechanism of planing, as a negative sinkage or rise in the water is induced by positive hydrodynamic lift. In this case, the negative value for both sinkage and trim at condition with stern foil at 3° angle of attack are generated by lift force.

Table 7.5 : Sinkage and Trim Result for a Ship with and without Stern Foil

EXPERIMENT						
Vs(knots)	Without Stern Foil		With Stern Foil 0°		With Stern Foil 3°	
	Sinkage	Trim	Sinkage	Trim	Sinkage	Trim
23	-0.0018	0.9975	0.0019	1.6361	-0.0009	-0.5620
24	-0.0018	0.9580	0.0024	1.4928	-0.0002	-0.6279
25	-0.0006	0.9003	0.0020	1.4572	-0.0002	-0.8416
26	-0.0004	0.9477	0.0062	1.4970	0.0001	-1.0479
28	0.0018	0.8569	0.0059	1.5462	-0.0004	-1.2637
30	0.0028	0.9352	0.0075	1.6442	0.0031	-1.6898

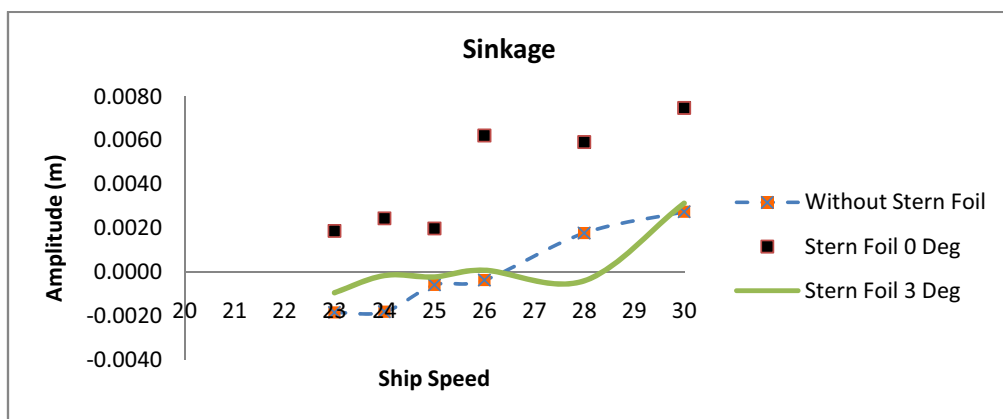


Figure 7.8 : Graphs on Comparison of Sinkage between a ship with and without Stern Foil

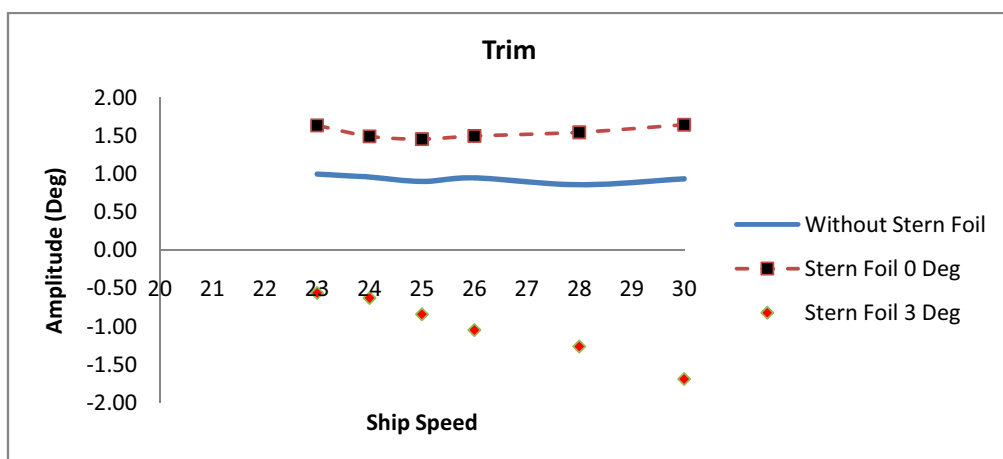


Figure 7.9 : Graphs on Comparison of Trim between a ship with and without Stern Foil

7.3 Seakeeping Analysis

In seakeeping analysis, there are several conditions to be investigated, which in this case the idea of the study is to determine the heave and pitch RAO. Practically, the result from the model analysis is a curve of RAO where the curve is the value of RAO versus wave length/ship length ratio. The table and graph of result RAO for heave and pitch and vertical acceleration forward and aft are present below. The seakeeping analysis for heave and pitch were analyse by computational method i.e. SEAKEEPER software and Strip Theory Method in theoretically while the experiment was conducting using towing carriage for validating the result calculation.

Table 7.6 shows the summaries of experiment result data heave and pitch RAO at various wave length/ship length ratios in condition with and without stern foil at zero degree angle of attack. The result data were obtained by conducting the seakeeping test at Froude Number 0.7296 ($V_m=3.2664$ m/s) in regular waves for both a ship model with and without stern foil.

Table 7.6 : Experiment Result Heave and Pitch RAO

WITHOUT STERN FOIL			WITH STERN FOIL		
Lw/Ls	HEAVE	PITCH	LW/LS	HEAVE	PITCH
0.50	0.2805	0.4878	0.50	0.2941	0.3903
0.60	0.2947	0.4863	0.60	0.1935	0.4314
0.80	0.2983	0.4903	0.80	0.3190	0.4574
1.00	0.2450	0.5245	1.00	0.2616	0.4587
1.20	0.2178	0.5667	1.20	0.2156	0.5223
1.40	0.3823	0.6753	1.40	0.2976	0.5938
1.60	0.5641	0.7980	1.60	0.4420	0.6954
1.80	0.7286	0.9533	1.80	0.5937	0.7874
2.00	0.9137	1.0797	2.00	0.8017	0.8741
2.20	1.0349	1.1518	2.20	0.9947	0.9340
2.30	1.0232	1.1023	2.30	1.0213	0.9223

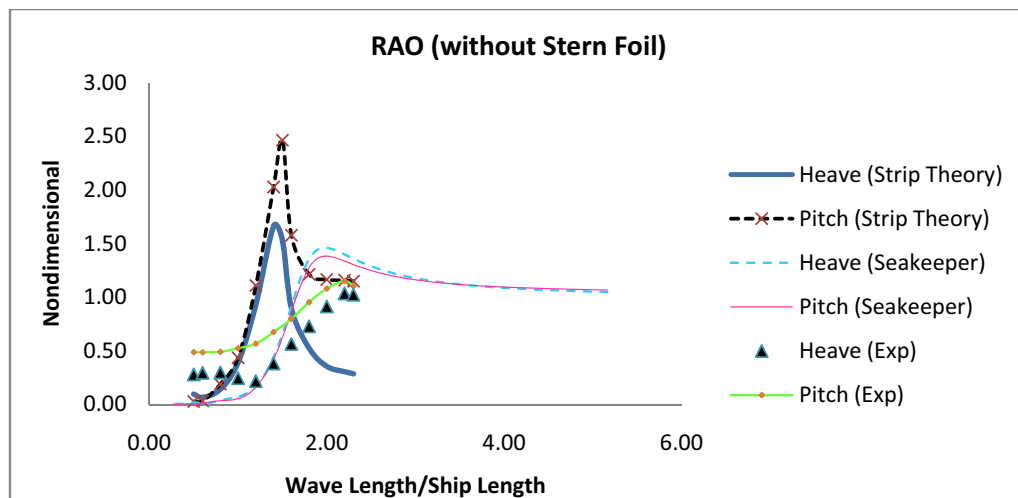


Figure 7.10 : Graphs on Comparison of Heave and Pitch RAO in different method for a ship without Stern Foil

The comparison between heave and pitch RAO is illustrated in figure 7.10 for models without stern foil. The value from graph curve which is obtained from calculation Strip Theory Method shows that high variance in term of heave and pitch RAO when compared to the SEAKEEPER program. The peak of resonance at wave length and ship length ratio slightly difference which the peak are 1.5 and 2.0 for strip method and SEAKEEPER program respectively. By comparing between the experiment and SEAKEEPER program, it is observed that the peak of resonance quite close to each other i.e. 2.0 and 2.1 respectively.

Table 7.7 shows the summaries of result calculation using Strip Theory Method [44] for heave and pitch RAO at various wave length/ship length ratio for model with and without stern foil.

Table 7.7 : Calculation Result Heave and Pitch RAO (Strip Theory)

Lw/Lm	Strip Theory Method	
	Heave RAO	Pitch RAO
0.50	0.0970	0.0278
0.60	0.0678	0.0333
0.80	0.1407	0.1889
1.00	0.3921	0.4361
1.20	0.9304	1.1083
1.40	1.6631	2.0278
1.50	1.5314	2.4639
1.60	0.8961	1.5776
1.80	0.5249	1.2139
2.00	0.3555	1.1639
2.20	0.3060	1.1583
2.30	0.2869	1.1500

The comparison of SEAKEEPER program and experiment for heave RAO curves in regular waves with and without a stern foil are shown in figure 7.11 and 7.12. From experiment result, by adding a stern foil to the model causes reduction of heave RAO at 1.6 wave length to ship length ratio which the heave RAO reduce to 3.88% compared to model without stern foil. While SEAKEEPER program estimate the reducing of heave RAO is 21.59% at 2.0 wave length to ship length ratio. This condition occurs due to the hydrodynamic lift react to the foil as a damping in order to reduce the vertical movement of the model.

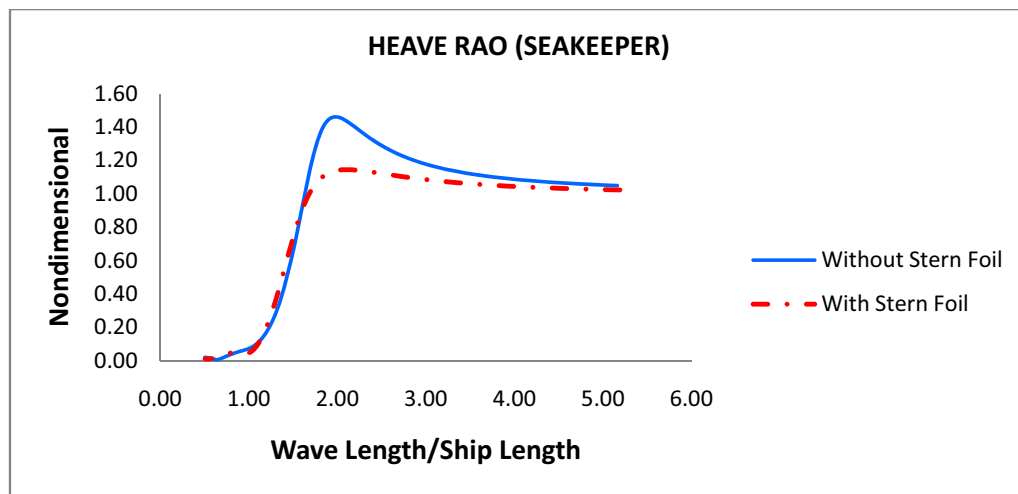


Figure 7.11 : Graphs on Comparison Heave RAO (SEAKEEPER)

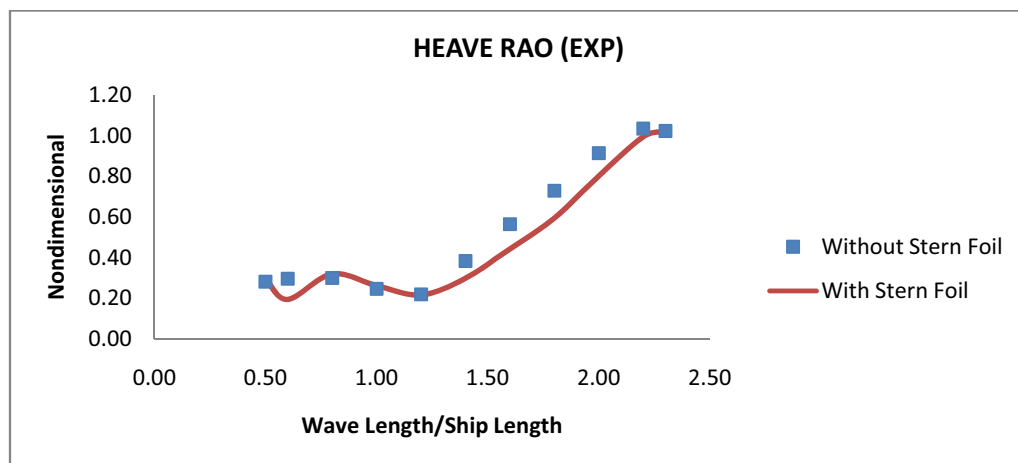


Figure 7.12 : Graphs on Comparison Heave RAO (Experiment)

The comparison of SEAKEEPER program and experiment for pitch RAO curves in regular waves with and without a stern foil are shown in figure 7.13 and 7.14. The performance of the model with stern foil in experiment results reduction of pitch RAO at 1.6 wave length to ship length ratio which the pitch RAO reduce to 18.91% compared to model without stern foil. While SEAKEEPER program estimate the reducing of pitch RAO is 41.12% at 2.1 wave length to ship length ratio. This condition occurs due to the hydrodynamic lift react to the foil as a damping in order to reduce the vertical acceleration of the model. This damping increases quite linearly with forward speed. However, this graph show for one speed i.e. 12.86 m/s (V_s) equivalent to $F_n:0.7296$. The pitch RAO of model with stern foil is smaller

compared to the model without stern foil, meaning that the pitch RAO of model incorporating with stern foil is able to reduce the motion.

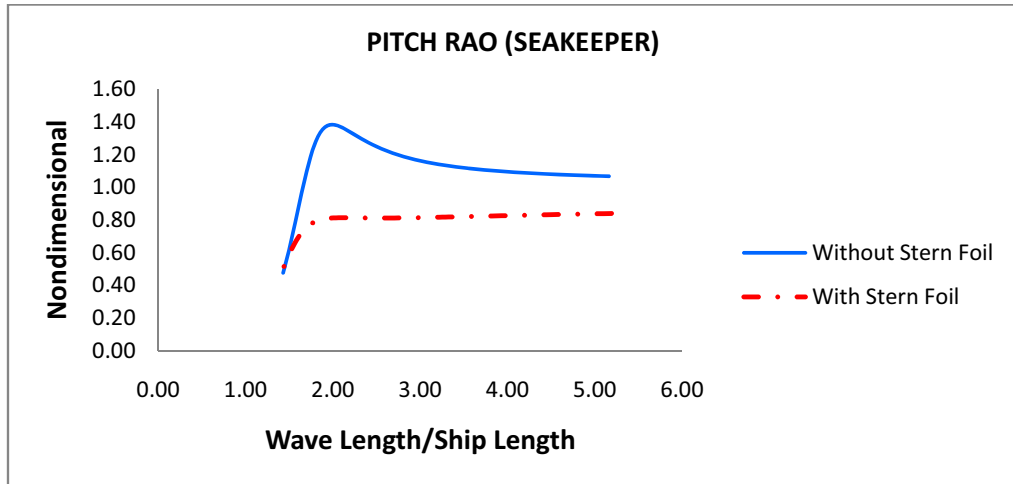


Figure 7.13 : Graphs on Comparison Pitch RAO (SEAKEEPER)

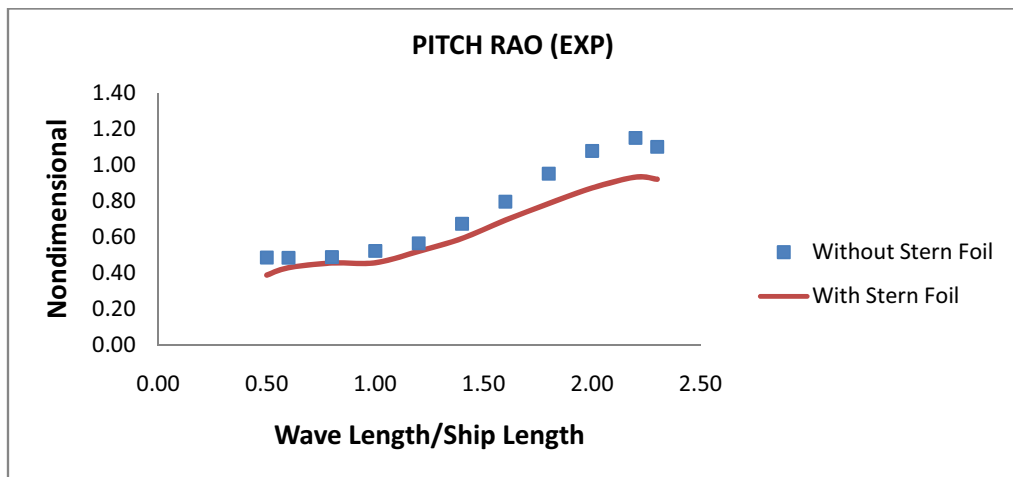


Figure 7.14 : Graphs on Comparison Pitch RAO (Experiment)

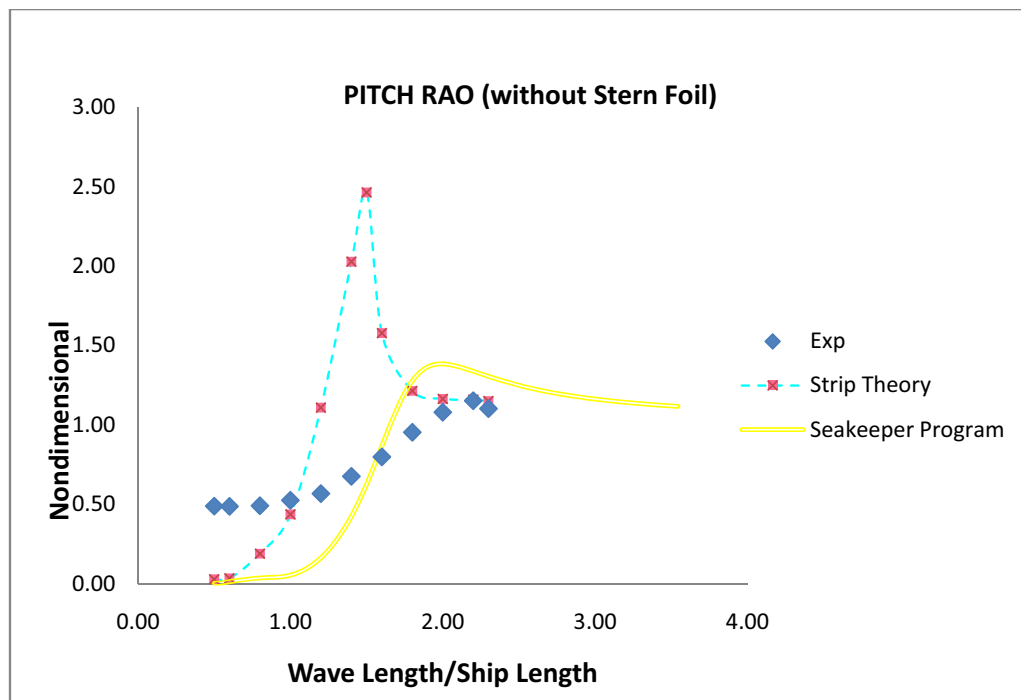


Figure 7.15 : Graphs on Comparison Pitch RAO between Experiment, Strip Theory and SEAKEEPER Program

Figure 7.15 and 7.16 show the heave and pitch RAO in term of comparison of method determination for the model without stern foil. Each graph curves had their own self characteristic where different methods are used to predict or calculates the response frequency of waves. From this graph curves, the result of Strip Theory show high different compared to the experiment and SEAKEEPER program. The peak resonance in every method is contradictory especially using Strip Theory. However, each heave and pitch RAO in experiment is lower compared to SEAKEEPER program and Strip Theory Method.

The result from Strip Theory become greater compare to other method due to transom and viscous effect that not taken into account in this calculation. Theoretically, the waves flow around a ship which the waves produce incident exciting forces, diffract when they reach the ship hull, which also produces forces on it, and the waves produce ship motion. These motions also produce radiated waves, which again produce forces.

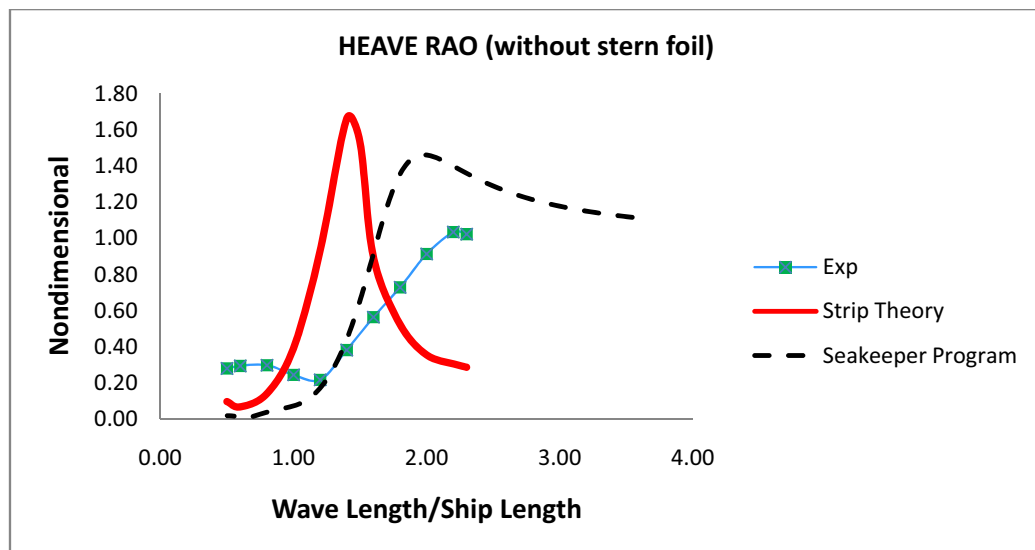


Figure 7.16 : Graphs on Comparison Heave RAO between Experiment, Strip Theory and SEAKEEPER Program

Table 7.8 : Forward and Aft Acceleration RAO with and without Stern Foil

Without Stern Foil			With Stern Foil		
Experiment					
Lw/Lw	Fwd Accs RAO	Aft Accs RAO	Lw/Ls	Fwd Accs RAO	Aft Accs RAO
0.50	4.1194	2.4490	0.50	2.4507	2.6186
0.60	7.0798	3.1874	0.60	2.6777	1.6635
0.80	12.5554	1.9384	0.80	9.0577	2.5430
1.00	24.4161	8.0260	1.00	20.4165	9.1626
1.20	39.2400	20.3977	1.20	36.6791	20.2481
1.40	78.4055	34.7158	1.40	63.0753	35.1755
1.60	116.2488	53.2275	1.60	91.7031	49.9581
1.80	155.9025	69.7581	1.80	122.7573	62.8907
2.00	189.4984	87.1439	2.00	154.0345	75.4872
2.20	213.0832	92.4476	2.20	181.0517	86.0142

The forward and aft acceleration RAO values of experiment of the model with and without stern foil are shown in table 7.8. The values are obtained from experiment at F_n 0.7296 in regular waves, head sea. From this data the values of

RAO either forward or aft acceleration are increased due to wave length. Both graph curves in figure 7.17 and 7.18 shows decreasing of forward and aft acceleration RAO by increasing the wave length when the model with stern foil. By adapting the stern foil at the model, the forward and aft RAO reduces to 21.1% and 6.14% respectively. The situation occurs due to reducing of trim angle where lift force exerted to the model by lifting effect .Hence a small reduction in trim angle may reduce the impact acceleration significantly. By inserting the stern foil can therefore be very effective in reducing the trim angle, thereby decreasing the impact forces.

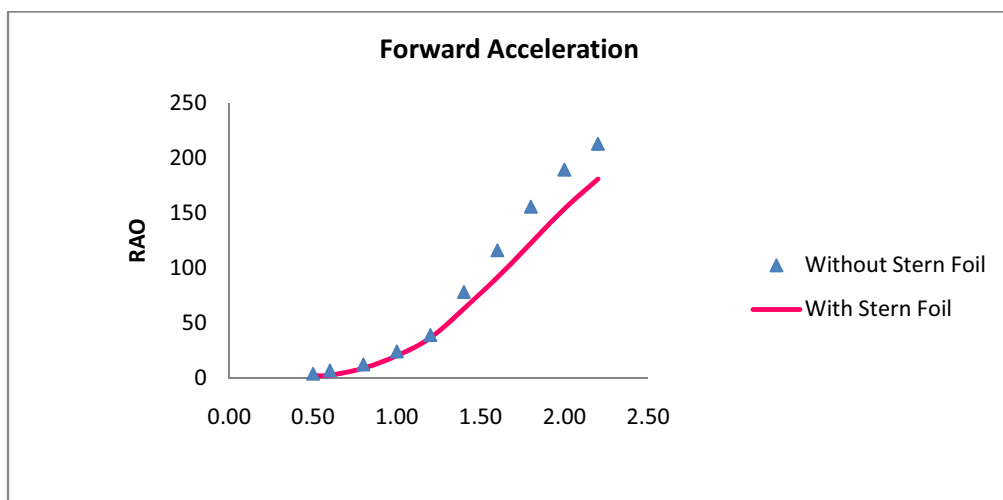


Figure 7.17 : Graphs on Comparison Forward Acceleration RAO

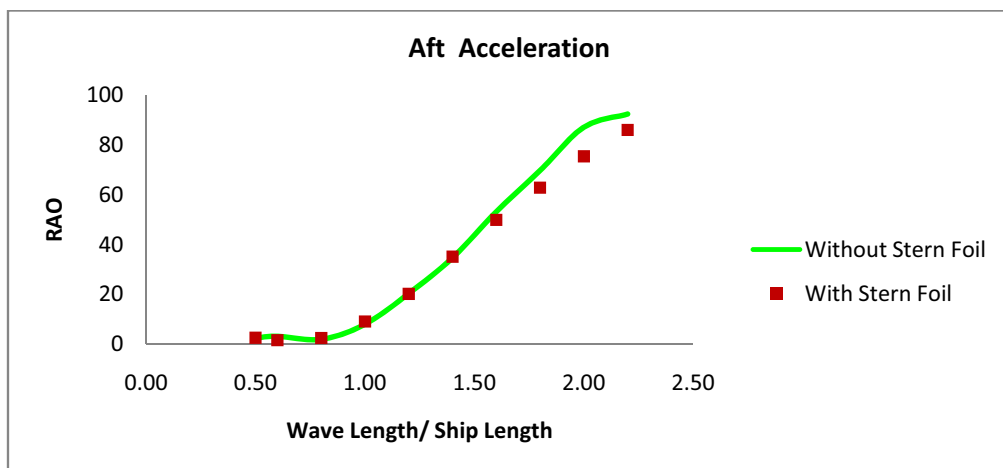


Figure 7.18 : Graphs on Comparison Aft Acceleration RAO

The stern foil is most efficient in damping vertical motion where it is proven in motion reduction technique. In this case, the stern foil is able to reduce the amplitude of heave, pitch, forward and aft acceleration. The experimental result show that at wave length 3.689 m, height waves 0.0654 m and wave period 1.45 s, the motion of the vessel automatically trim down. This condition resulting good indication that stern foil will reduced the motion in term of heave, pitch, forward and aft acceleration. Figure 7.19, 7.20, 7.21 and 7.22 show that the variance of amplitude in every RAO for model with and without stern foil.

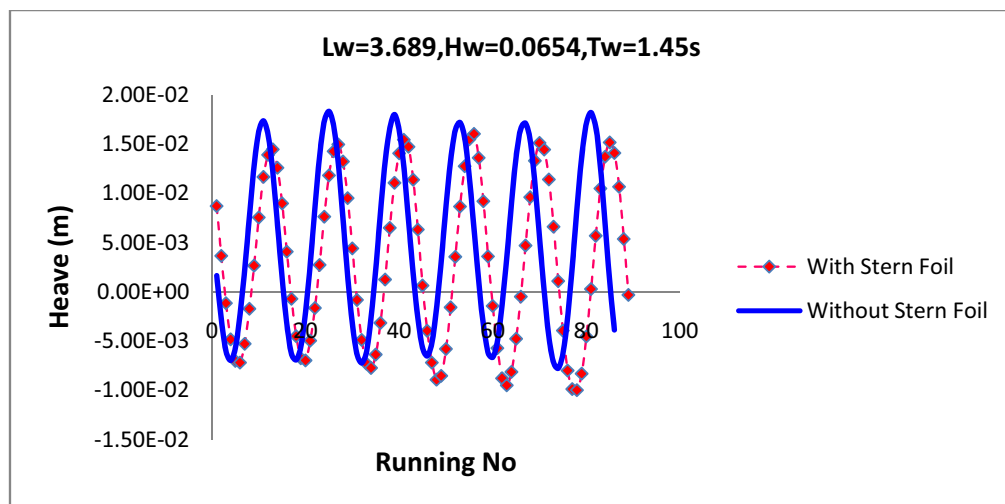


Figure 7.19 : Record Curves of Heave Amplitude

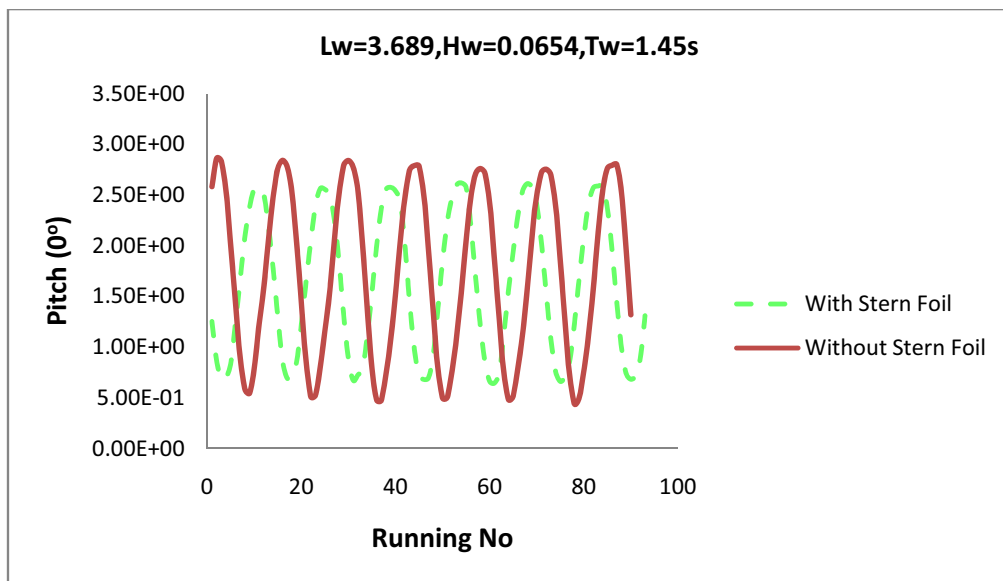


Figure 7.20 : Record Curves of Pitch Amplitude

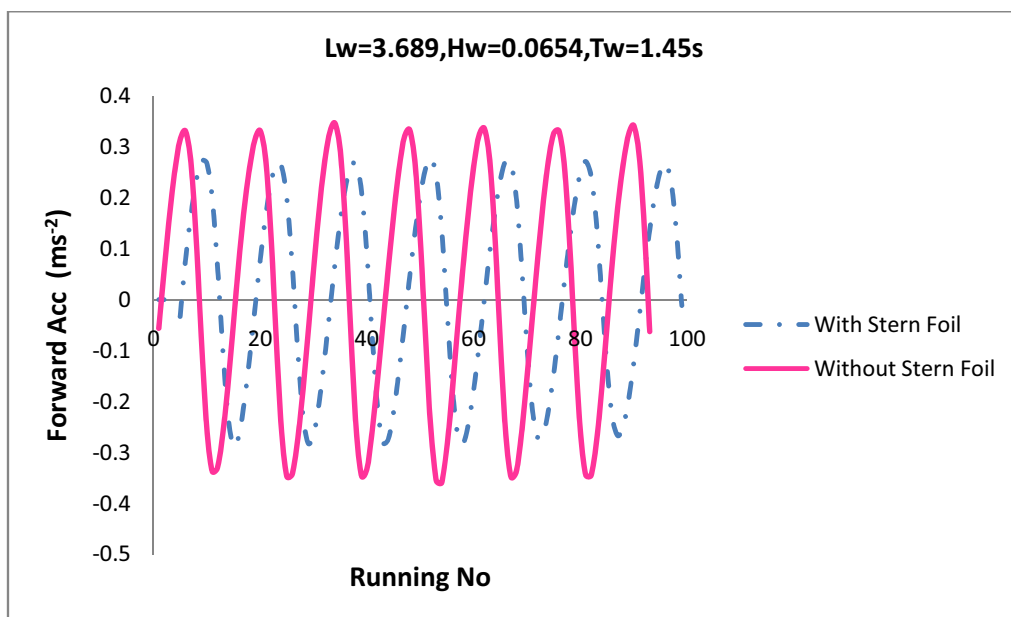


Figure 7.21 : Record Curves of Forward Acceleration Amplitude

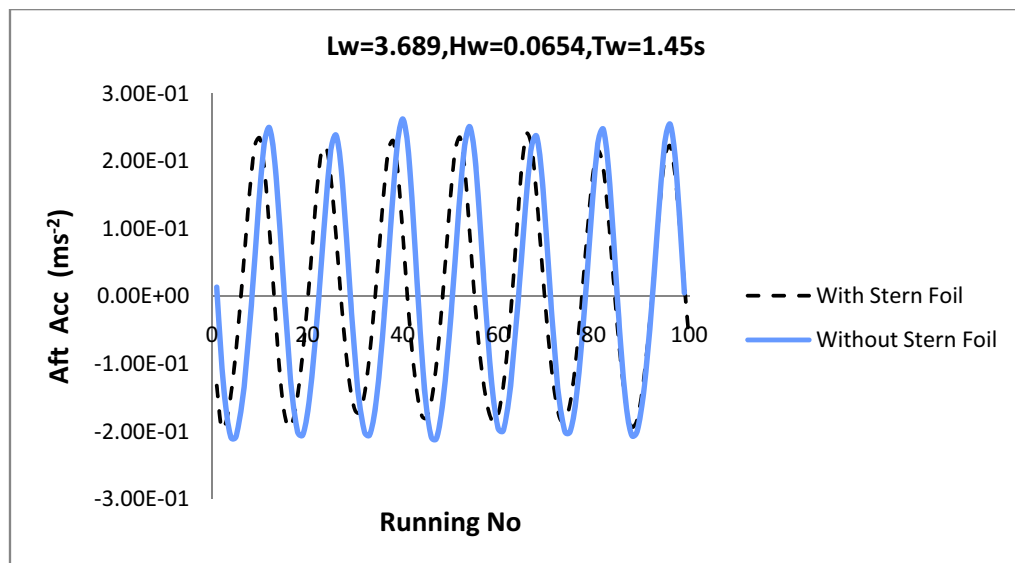


Figure 7.22 :Record Curves of Aft Acceleration Amplitude

7.4 Concluding Remarks

From the tables and graphs above, the performance prediction of high speed craft can be materialized. The performance prediction of high speed craft in term of resistance and seakeeping are a challenging work where many aspects or parameters should to be taken into account on calculation especially the ones which are involve in modification work. Even though Strip Theory and motion prediction software can be used to optimise hullforms for resistance and seakeeping performance but it is being simplified due to linearization of formulation. However, in this case, the SEAKEEPER program is quite precise in predicting the ship motion. By incorporating the stern foil it can have significantly better performance even though it will affect the resistance performance at slow speed but resulting good performance in seakeeping quality at high speed.

CHAPTER 8

CONCLUSION

8.1 Conclusion

Performance prediction on high speed craft especially in planing craft is difficult to achieve for both resistance and seakeeping. The performance becomes a challenging work if the craft has special or unique hull form. The study on planing craft was conducted on exclusive hullform which is called M Hull. The objective of this research is to predict the performance of the planing craft M Hull in term of resistance and seakeeping for different test condition (with and without stern foil).

Generally, many of the hypothesis and linearization that can be made for conventional craft are not applicable especially for vessel that has modification on it hullform, and great care has to be taken to ensure that the predictions are reliable and the way in which the experiment was conducted. In order to carry out the performance prediction for certain vessel, special techniques have to be developed for the experiments associated with the prediction of the performance of high speed craft.

By incorporating the stern foil (0° angle of attack) on the model, the resistance is slightly higher compared to the model without stern foil. However, the increasing patterns of resistance becoming less at high speed, which is at speed 30

knots, the resistance percentage is increased only at 8% (refer table 7.1 and 7.2). The resistance predictions using theoretical formulation (Savitsky and two dimensional Method) give nearly precise value to the experimental result for both models. The Hull Speed program and theory also resulting quite similar resistance value (see table 7.4 and figure 7.5).

Experiment and software program (SEAKEEPER) results show that the contributions of stern foil reduce the heave, pitch, forward and aft acceleration of the M-Hull planing craft in waves is important. In this study, the model speed at 3.2664 m/s ($F_n=0.7296$), SEAKEEPER result show heave and pitch RAO reducing 21.59% and 41.12% respectively. While the reducing of heave and pitch RAO in experiment are 3.8% and 18.91% respectively (see figure 7.12 and 7.14). Even though the percentage between SEAKEEPER is quite large but both method show the reducing pattern on heave and pitch RAO. The forward and aft acceleration also reduce when the model incorporate with stern foil (see figure 7.17 and 7.18).

8.2 Future Work

Based on analysis, the performance prediction especially on resistance can be improved for further development. These are some recommendations for further research of M-Hull planing craft with stern foil:

1. The stern foil replaced by controllable stern foil with Proportional Integration Derivatives (PID) control system.
2. Conduct the model test beyond $F_n > 1.0$.
3. Carry out advance research on dynamic instability, propulsion and manoeuvring performance for the model with Stern Foil.

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APPENDIX A

Details Planning for Project I and II

APPENDIX B

Resistance and Seakeeping Test

1.0 Designation

Model Test of a Planing Craft (M Hull), Model MTL- 046 (Resistance and Seakeeping Test)

2.0 Objectives

The resistance test of a planing craft model is carried out to determine:

- a. The required total resistance a full scale of bare hull of planing craft (M Hull) at corresponding speeds in calm water.
- b. The required total resistance of a full scale of planing craft of full scale of planing craft with and without stern foil at 25 knots in regular waves.

The seakeeping test for model is carried out to determine:

- a. The significant effect of motions (heave,pitch RAO) in regular waves (head seas) at various wavelength at design speed 25 knots with and without stern foil.
- b. Forward and aft acceleration in regular waves at design speed 25 knots with and without stern foil.
- c. To present the result of the Response Amplitude Operators for the both model according to a and b.

3.0 Introduction

Model testing in towing tank is one of the essential parts in ship design process and resistance test is needed in determining the resistance and power effective of the ship. To confirm design predictions of powering requirement, scaled model resistance test are carried out. Measurement of resistance of planing craft, model MTL- 046 is undertaken and the data/results are analyzed accordingly to obtain full-scaled resistance. Analysis is to be carried out using ITTC 1957 method.

Another model test that can be carryout is seakeeping test. Actually this test is to measure the behavior of the vessel at seas. Model testing provides an attractive alternative. The seakeeping test can predict the characteristics of the vessel in term of motion which is it gives significant effect of motion. The success of a ship design depends ultimately on its performance in a seaway. However, the in a realistic seaway is such a complex problem that ship designers are generally forced to select their hull forms and ship dimensions on the basis of calm water performance without much consideration of the sea and the weather conditions. Only very recently sophisticated experimental techniques such as seakeeping test and computer applications in ship motion theories have made it possible for the designer to consider the seakeeping qualities of his ship at an early stage.

In this model testing, the resistance test will be conducted at bare hull with and without stern foils in various speed at calm water, while seakeeping test will be carryout in three conditions at cruising speed (25 knots) in regular waves. The model test will be conducted in order to optimize the performance at high speed condition.

3.1. Resistance

The resistance of a ship at a given speed can be defined as a force required to tow the ship at that speed in smooth water, assuming no interference from the towing ship. The power necessary to overcome this resistance is called the tow rope or effective power. This total resistance is consists of a number of different components, which is caused by a variety of factors and which interact one with other in a complicated way. In simplified manner, it is usual to consider the total calm water resistance inclusive four main components;

- a. The **frictional resistance**, due to the motion of the hull through a viscous fluid.
- b. The **wave making resistance**, due to the energy created continuously on the surface of water by the ship to the wave system.
- c. **Eddy resistance**, due to the energy carried away by eddies shed from the hull or appendages. Local eddying will occur behind appendages such as bossing,

shaft struts, and from stern frames and rudders if these items are not properly streamlined and aligned with the flow.

d. **Air resistance** experienced by the above water line of the main hull and superstructures due to the motion of the ship through air.

The resistance from ii), iii) and iv) commonly can be taken together under the term known as residuary resistance. Figure 1 can be referred to give us clearer picture about the total resistance of the ship and its components related.

The given summary of the total resistance is convenient but not scientifically correct since the first three types all react one with the other. It can be seen where the skin drag gives rise to a boundary layer which virtually alters the shape of the hull and hence the pressure distribution and the resulting wave making resistance. Also the wave system alters the wetted surface area and so the skin friction of the ship resistance. Nevertheless, it is convenient to make such division for practical use, but the presence and interaction of all these factors well illustrated the complexity of the ship resistance problem.

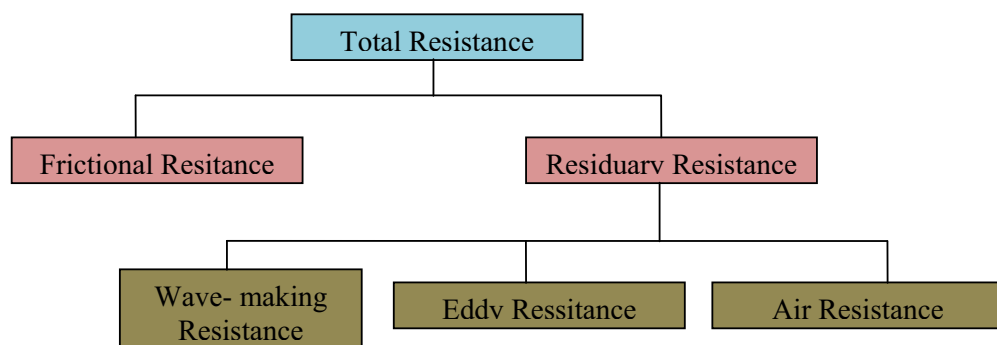


Figure 1 Basic component of Resistance

The practical methods whereby model tests are carried out at full scale Froude number and corrections made for the differences in Reynold's number are given. The method originally used by William Froude is applied, and then its modern derivative and finally that by Hughes is extended. The basic and early approach,

suggested by Froude which breakdown of resistance into several components, assuming:

Total Resistance = Skin Friction + “The Rest”, which he termed Residuary

i.e.

$$C_T = C_F + C_R$$

The skin friction coefficient (CF) is estimated from data for a flat plate of same length, wetted surface area and velocity of model or ship. The difference between total resistance and skin friction gives the residual resistance. Hence the part dependent on Reynold’s number is separately determined and the model test is carried out at the corresponding velocity which gives equality of Fn for ship and model. Hence dynamic similarities for wave-making (or residuary resistance) is obtained.

Hence if residuary resistance is considered,

$$R_R = \rho V^2 L^2 f_2 \left(\frac{V}{\sqrt{gL}} \right)$$

Model:

$$R_{Rm} = \rho V_m^2 L_m^2 f_2 \left(\frac{V_m}{\sqrt{gL_m}} \right)$$

Ship:

$$R_{Rs} = \rho V_s^2 L_s^2 f_2 \left(\frac{V_s}{\sqrt{gL_s}} \right)$$

ρ is constant, g is constant, f_2 same for ship and model (i.e. same Fn) and it follows that:

$$\frac{V_m}{\sqrt{L_m}} = \frac{V_s}{\sqrt{L_s}}$$

or

$$\frac{V_m^2}{L_m} = \frac{V_s^2}{L_s}$$

Then,

$$\frac{R_{Rm}}{R_{Rs}} = \frac{V_m^2 L_m^2}{V_s^2 L_s^2} = \frac{L_m^3}{L_s^3} = \frac{\nabla_m}{\nabla_s}$$

Froude's law; when speeds of ship and model are in ratio of square root of their lengths, then resistance due to wave-making varies as their displacements. (Speeds in ratio of square root of lengths called "corresponding speeds"). In coefficient form:

$$C_T = \frac{R_T}{\frac{1}{2}\rho S V^2}, \quad C_F = \frac{R_F}{\frac{1}{2}\rho S V^2}, \quad C_R = \frac{R_R}{\frac{1}{2}\rho S V^2}$$

$$S \propto L^2, V^2 \propto L \text{ (due to same Fn), hence } \frac{1}{2}\rho S V^2 \propto \rho L^3 \propto \nabla$$

Thus,

$$\frac{C_{Rm}}{C_{Rs}} = \frac{R_{Rm}}{\nabla_m} \frac{\nabla_s}{R_{Rs}} = \frac{\nabla_m}{\nabla_s} \frac{\nabla_s}{\nabla_m} = 1$$

for constant $\frac{V^2}{L}, \frac{V}{\sqrt{L}}$ i.e. $C_{RM} = C_{RS}$ same for model and ship at constant $\frac{V}{\sqrt{L}}$

$$\text{Now,} \quad C_{TM} = (1+k)C_{FM} + C_{RM}$$

$$\text{And} \quad C_{TS} = (1+k)C_{FS} + C_{RS}$$

$$\text{but,} \quad *C_{RM} = C_{RS}*$$

Hence, total ship resistance can be determined by following equation:

$$R_{ts} = C_{ts} * 0.5 * \rho * S * V^2$$

3.2. Seakeeping

Seakeeping is the dynamic response of the ship affected by environmental forces, primarily wind and waves. It is often a limiting factor in operability, specifically speed loss. By incorporating seakeeping into the initial ship design, the ship's performance and efficiency can improve. The primary parameters that affect seakeeping are the ship proportions, including waterplane geometry and weight distribution. Secondly, unique hull characteristics such as transom sterns, bulbous bows or motion damping devices also affect the ship's seakeeping abilities. There are some general seakeeping design guidelines that should be always kept in mind during the design process.

1. Longer lengths are better for a ship's seakeeping abilities
2. Wave excitation comes in through the waterplane area. Smaller waterplane area ships experience fewer motions, but also have less damping which results in pronounced resonant peaks.
3. Wave excitation also comes through pressure, which reduces exponentially with depth.

The seakeeping response of a ship is a random process and must combine several elements. The key elements are ship characteristics, required functions for mission achievement and a specified sea environment for the given mission.

3.2.1 Couple Heave and Pitch Motion in Head Sea

As described earlier, the pitch motion of a ship in the waves acts in rotational motion about y - direction. It is always concentrates on head seas where the heading angle will be 180° opposing the waves. The pitch motion normally having an encounter frequency due to its direction touches the successive wave crests. The heave motion in regular waves is considered as movement of ship in vertical plane motion and the motion acts transitional. Since its buoyant force is then greater than its weight, the ship will move vertically up. Whilst the equilibrium is reached, the ship continue rising because of its momentum. Then the weight is greater than the buoyant force and it will tend to slow the motion. The ship reach at extreme position when the velocity is finally zero and the ship will move vertically downward since the weight is greater than buoyant force. The process of a ship move vertically up and down will be continue indefinitely if there is no damping force acts in the opposite direction of motion and this known as a heaving.

This couple of motion was dealing with vertical motion which can be investigated in a model basin. Actually the coupled heave and pitch for the head sea condition also can be investigated analytically by the strip theory method. Basically the strip theory was derived two equations for heave and pitch respectively.

According to Newton's second law, at any instant all vertical forces on the ship are in dynamic equilibrium. Thus, the heaving and pitching equation can express as:

$$m\ddot{z} = \sum F$$

$$I\ddot{\theta} = \sum M$$

Where

$\sum F$ = The sum of various fluid forces (vertical hydrodynamic forces as well as the wave excitation force)

$\sum M$ = The sum of corresponding moments acting on the vessel because of relative motion of vessel and wave.

In order to simplified this complex motion problem several assumptions and limitations have been made as follows:

1. The ship must be heading into the waves in a direction transverse to their crest line.
2. The seaway is considered to consist of regular harmonic waves.
3. The ship motions of surge, sway, yaw and roll are neglected.

The general equation coupled of heaving and pitching can be expressed as:

$$\left(\frac{\Delta}{g} + a_{zz}\right)\ddot{z} + b_{zz}\dot{z} + c_{zz}z + a_{z\theta}\ddot{\theta} + b_{z\theta}\dot{\theta} + c_{z\theta}\theta = F_{(t)}$$

$$(I_{\theta\theta} + a_{\theta\theta})\ddot{\theta} + b_{\theta\theta}\dot{\theta} + c_{\theta\theta}\theta + a_{\theta z}\ddot{z} + b_{\theta z}\dot{z} + c_{\theta z}z = M_{(t)}$$

For determining the effect of stern foil, calculating the coupling movement of heave and pitch in regular waves, the effect of foil on the vessel is considered as a small perturbation, and the motion of planing hull vessel with stern foil in waves may be described by the following equation:

$$\left(\frac{\Delta}{g} + a_{zz}\right)\ddot{z} + b_{zz}\dot{z} + c_{zz}z + a_{z\theta}\ddot{\theta} + b_{z\theta}\dot{\theta} + c_{z\theta}\theta = F_{(t)} + \Delta F$$

$$(I_{\theta\theta} + a_{\theta\theta})\ddot{\theta} + b_{\theta\theta}\dot{\theta} + c_{\theta\theta}\theta + a_{\theta z}\ddot{z} + b_{\theta z}\dot{z} + c_{\theta z}z = M_{(t)} + \Delta M$$

Where,

\ddot{z}, \dot{z}, z = heave acceleration, velocity and displacement, respectively.

$\ddot{\theta}, \dot{\theta}, \theta$ = pitch angular acceleration, velocity and displacement respectively.

Δ/g = mass of vessel.

I_{yy} = pitch inertia moment of vessel.

$a_{zz}, b_{zz}, c_{zz}, a_{z\theta}, b_{z\theta}, c_{z\theta}, a_{\theta\theta}, b_{\theta\theta}, c_{\theta\theta}, a_{\theta z}, b_{\theta z}, c_{\theta z}$ = stability derivatives

ΔF = flap excited force

ΔM = flap excited moment

Basically the *RAO* are derived from the amplitudes by dividing the motion response by the wave input parameter:

$$RAO = \frac{\text{motionresponse}}{\text{waveinputparameter}}$$

The *RAO* for pitching, heaving and rolling are defined as below:

Pitching *RAO* = Pitch Amplitude / Wave Slope

$$= \frac{\theta_a}{k\zeta_a}$$

Heave *RAO* = Heave Amplitude / Wave Amplitude

$$= \frac{z_a}{\zeta_a}$$

4.0 Experiment Apparatus, Equipment and Facility

4.1 Towing Tank

Main dimensions of 120 m length, 4 m width and, 2.5 m depth. Detail particulars of towing tank are shown in figure 2 and 3.

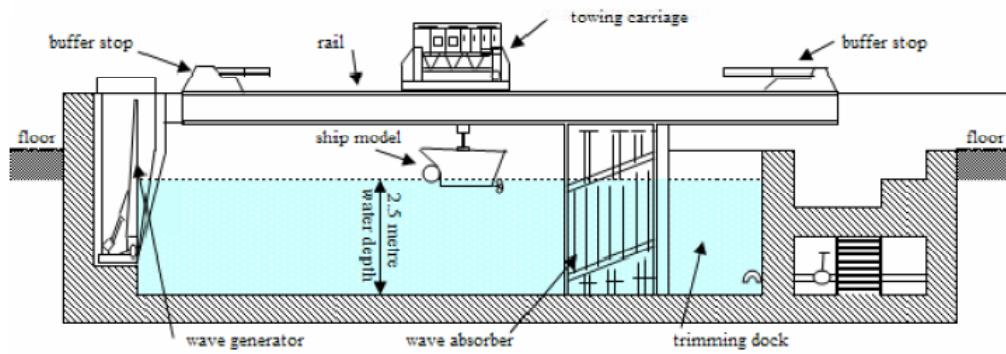


Figure 2 Side view of towing tank

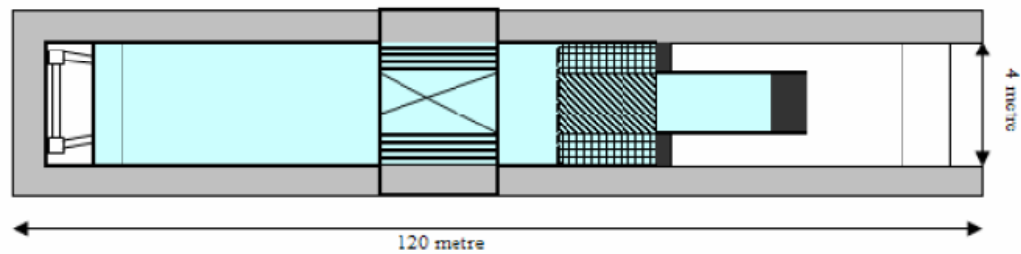


Figure 3 Plan view of towing tank

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.

4.3 Data acquisition system (refer Figure 4 for the DAAS block diagram)

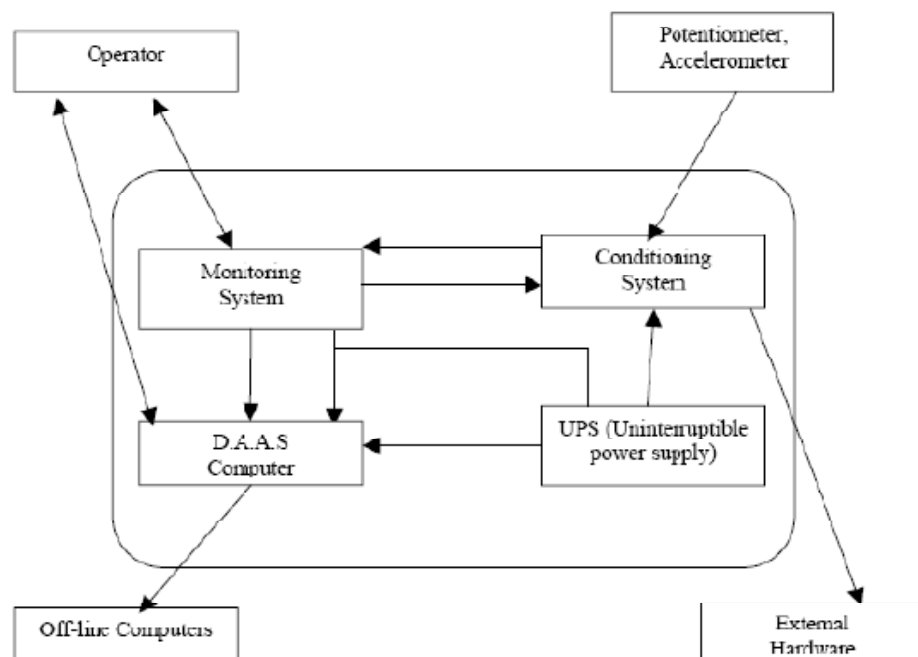


Figure 4 The block of diagram of Data Acquisition Analysis System (DAAS)

4.4 Ship's model. Refer to figure 5.

4.4 Airstrut

1. Consist of an aluminium tube which give the frictionless and stick-slip free result.
2. Connected to gimbals with 2 potentiometer each of them used to measure the pitch and roll action motion.

4.5 Gimbals

1. The gimbals permit the model to roll and pitch.
2. To measure the roll and pitch angle, the gimbals are equipped with the potentiometer.
3. The calibration device inside the potentiometer enables the pitch angles and roll angles in a range between -300 up to +300.

4.6 Towing Guide

1. Towing guide is attached to the model at the center of gravity of the model.
2. The towing guide enables the model to move freely in vertical and longitudinal direction but it keeps the model transversely in position.
3. The towing guide is fully balanced, so no force is acting on the model when there is no movement.

4.7 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 absorbed by a wave absorber at the other end of the tank.

The maximum wave height that can be achieved is 0.44m for wave period of 1.7sec. The wave generator of the basin is capable in generating regular and irregular waves over period range of 0.5sec to 2.5sec. The irregular waves; characterized by a Jonswap or a Pierson-Moskowitz spectrum can be generated up to a significant wave height of 0.25m. Basically it consists of a hinged dry back type flap, hydraulically driven and computer controlled.

Basically the wave generator consists of a hinged dry back type flap, hydraulically driven and computer controlled. The wave generator of the basin is capable of generating regular and irregular waves over a wave period range of 0.5sec to 2.5sec.

5.0 Procedures

5.1 Resistance Test

1. Prepare the model that to be tested. The model size is determined based on appropriate scale (λ) from full scale size. The model used is the Planing Craft M Hull, with scale factor (λ) value of 15.5. The ship particular and stern foil parameters for resistance test are shown in table 4 and table 5 respectively.
2. Determine the model weight using Law of Similarity. Install some weights (ballast weight) into the model so that the total weight of the model is same with the model displacement.
3. Put the model completed with its weight on the swing frame for ballasting and LCG determination.
4. Attached the completed model to the towing carriage. The model is attached to the carriage through the air strut with the help of the base plate at LCG of the model. The view of how the model is attached can be referred to the figure 4.

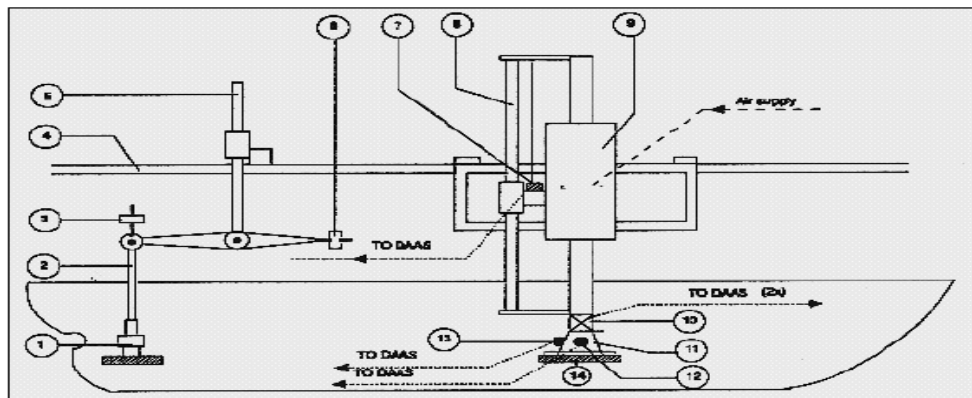


Figure 4 The Arrangement and Connection of completed Model to the Carriage

5. After all the attachments and connection of transducers are completed and prepared, key-in the required model speed which corresponding to the ship speed in the computer.

6. The model test will be carried out for 6 corresponding speeds without and with stern foil as shown in table 1, 2 and 3:

Table 1 Summaries of Resistance Test for without Stern foil

Ship Speed, Vs (knot)	Froude Number (F_n)	Ship Speed, Vs (ms^{-1})	Model Speed, Vm (ms^{-1})
23	0.6713	11.8312	3.0051
24	0.7004	12.3456	3.1358
25	0.7296	12.8600	3.2664
26	0.7588	13.3744	3.3971
28	0.8172	14.4032	3.6584
30	0.8756	15.4320	3.9197

Table 2 Resistance Test Summary for with Stern foil at angle of attack (α) 0°

Ship Speed, Vs (knot)	Froude Number (F_n)	Ship Speed, Vs (ms^{-1})	Model Speed, Vm (ms^{-1})	α
23	0.6713	11.8312	3.0051	0°
24	0.7004	12.3456	3.1358	0°
25	0.7296	12.8600	3.2664	0°
26	0.7588	13.3744	3.3971	0°
28	0.8172	14.4032	3.6584	0°
30	0.8756	15.4320	3.9197	0°

Table 3 Resistance Test Summary for with Stern foil at angle of attack (α) 3°

Ship Speed, Vs (knot)	Froude Number (F_n)	Ship Speed, Vs (ms^{-1})	Model Speed, Vm (ms^{-1})	α
23	0.6713	11.8312	3.0051	3°
24	0.7004	12.3456	3.1358	3°
25	0.7296	12.8600	3.2664	3°
26	0.7588	13.3744	3.3971	3°
28	0.8172	14.4032	3.6584	3°
30	0.8756	15.4320	3.9197	3°

7. Run the towing carriage at corresponding speed. Start the running with model speed, V_m at 3.0051ms^{-1} .
8. When the towing carriage speed becomes constant, approximately 10 seconds, click the “continue” button to measure and log the data into the computer.
9. After around 10 to 15 seconds running at the constant speed, click “stop” to stop logging the data and after that slow down the towing carriage and then stop.
10. Repeat the Step 7 until 9 for difference model speed.
11. Note down all results into a table.
12. Repeat the same procedures for model with stern foil with angle of attack() 0° and 3° .
13. From the result of the test, graph R_S vs V_S were plotted.

Table 4 The Main Parameters of Resistance Test

Description	Ship	Model
Scale ratio, λ	1	15.5
Length Waterline L_{WL} (m)	31.667	2.0430
Breadth Waterline (m)	7.4168	0.4785
Depth , D (m)	3.300	0.2129
Draught, T (m)	1.365	0.0881
Displacement (Tonne)	130.275	0.0341
Wetted surface area, S (m^2)	220.000	0.9157
Longitudinal center of gravity abaft \otimes , LCG (m)	2.258	0.1457
Vertical center of gravity (from keel) (m)	2.124	0.1370
Deadrise angle, β (deg)	14	14
Block coefficient, C_b	0.4	0.4

Table 5 Stern Foil Parameters

	Model	Prototype
Foil		
Height (h) m	0.071	1.1005
Chord Length (c) m	0.065	1.0000
Span (s) m	0.452	7.0000
Thickness (t) m	0.008	0.1194
Angle of Attack deg	0 and 3	0 and 3
<i>Camber (f)</i>	0.000	0.0000
	Model	Prototype
Strut		
Height (h) m	0.071	1.1005
Chord Length (c) m	0.021	0.3255
Thickness (t) m	0.006	0.0930
Angle of Attack deg	0.000	0.0000
<i>Camber (f)</i>	0.000	0.0000

5.2 Seakeeping Test

Seakeeping tests in regular wave were carried out in order to determine the RAO for the ship. Following are the experiment procedure:

1. The model was ballasted until the total weight reach to the condition 1, i.e. using a ballast of 34.1 kg where the weight ballast include the air strut (air strut = 16.12 kg).
2. The center of gravity of model is then being determined by using swinging frame.
3. Check the heeling of the model by placing the model into basin. If there is heeling, the weights are then moved sideways (port or starboard) until the model at even keel.
4. The model was then fixed to the gimbal.
5. Then run the towing carriage for the model at speed 3.2664 m/s (model speed) at selected wave characteristics, which was determined by using Law of Comparison.

6. Repeat step 5 for the same speed at difference wave characteristics.
7. For each running, the model seakeeping test values such as wave height, forward heave, pitch, forward and aft acceleration were then determined.
8. The RAO of the ship at each wave characteristics were determined and plot the RAO curve.
9. Repeat the procedure for the model with stern foils at same speeds and waves characteristics.
10. The model test will be carried out for 10 different wavelength characteristic at design speeds without and with stern foil as shown in table 6:

Table 6 Experiment Data for Seakeeping in Regular Wave (Head Seas)

Seakeeping Test Condition	Test No.	Wave Characteristics (Model)		
		L_w/L_s	L_w (m)	H_w (m)
Bare hull at even keel 1.365 m and running at 25 knots (12.86 m/s) at ship condition ($T_m = 0.0881$ m and $V_m = 3.2664$ m/s at model condition)	MTL No. 46	0.50	1.0215	0.0204
		0.60	1.2258	0.0245
		0.80	1.6344	0.0327
		1.00	2.0430	0.0409
		1.20	2.4516	0.0490
		1.40	2.8602	0.0572
		1.60	3.2688	0.0654
		1.80	3.6774	0.0735
		2.00	4.0860	0.0817
		2.20	4.4946	0.0899

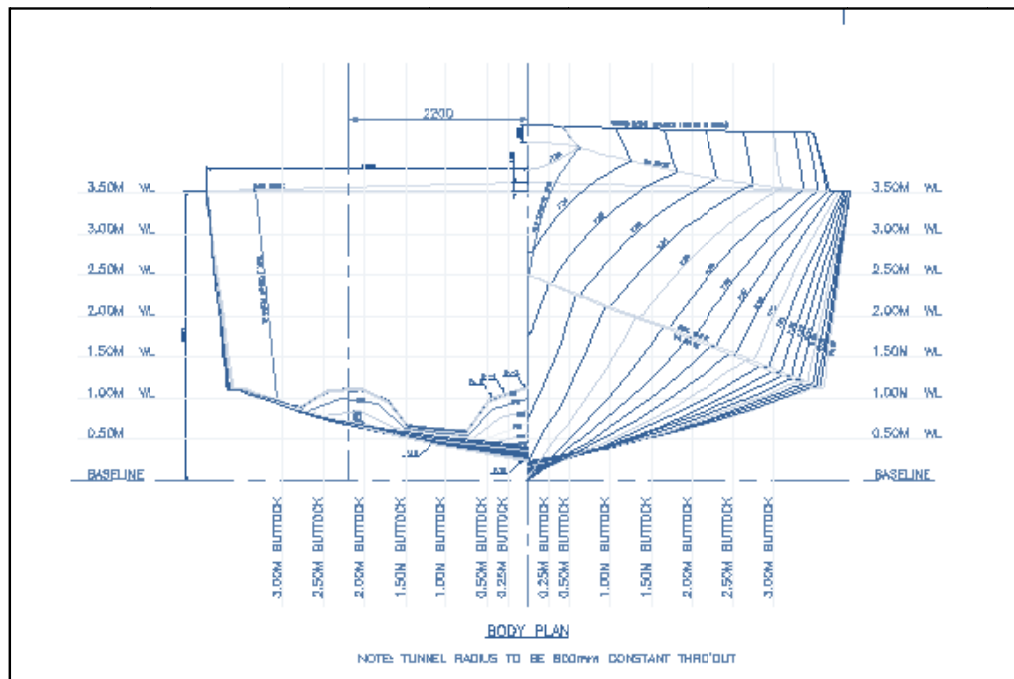


Figure 5 Ship Model Body Plan

APPENDIX C

Sample of Resistance Prediction

The example of calculation of resistance with stern foil using Savitsky and 2D methods are show below:

Ship Particular

LOA	34 m	β	14 deg
LWL	31.667 m	ρ	1025 kg/m ³
BOA	7.85 m	v	9.7117×10^{-7} m ² /s
T	1.365 m	g	9.81 m/s ²
BWL	7.4168 m		
Δ	130.275 tonne		
∇	127.0976 m ³		
L _{CG}	13.5755 m		

Stern Foil Parameters

Model			Prototype	
Foil				
h	0.071	m	1.1005	m
c	0.065	m	1.0000	m
s	0.452	m	7.0000	m
t	0.008	m	0.1194	m
α	0.00	deg	0.00	deg
<i>f (camber)</i>	0.00	m	0.00	m

Strut				
Model			Prototype	
h	0.071	m	1.1005	m
c	0.021	m	0.3255	m
t	0.006	m	0.0930	m
α	0.00	deg	0.00	deg
<i>f (camber)</i>	0.00	m	0.00	m

1. Resistance Prediction Barehull

For speed 25 knots equivalent to 12.86 m/s, the following formulation for resistance prediction is:

$$\begin{aligned} \text{Speed Coefficient } C_v &= \frac{V}{\sqrt{gb}} \\ C_v &= \frac{12.86}{\sqrt{9.81(7.4168)}} \\ &= 1.5076 \end{aligned}$$

The lift coefficient for finite deadrise

$$\begin{aligned} C_{L\beta} &= \frac{\rho g \nabla}{1/2 \rho V^2 b^2} \\ C_{L\beta} &= \frac{1025 * 9.81 * 127.0976}{\frac{1}{2} * 1025 * 12.86^2 * 7.4168^2} \\ &= 0.2741 \end{aligned}$$

Where β is the deadrise angle at the mid-chine position (in degree). Then Flat plate lift coefficient C_{Lo} and wetted length beam ratio λ can be calculating by numerical method (Newton Raphson).

$$\begin{aligned} C_{L\beta} &= C_{Lo} - 0.0065\beta C_{Lo}^{0.6} \\ f &= C_{Lo} - 0.0065\beta C_{Lo}^{0.6} - C_{L\beta} \\ &= C_{Lo} - 0.0065(14)C_{Lo}^{0.6} - 0.2741 \end{aligned}$$

$$\begin{aligned} \hat{f} &= 1 - 0.0039\beta C_{Lo}^{-0.4} \\ \hat{f} &= 1 - 0.0039(14)C_{Lo}^{-0.4} \end{aligned}$$

$$C_{Lo1} = C_{Lo} - \frac{f}{\hat{f}}$$

When $C_{Lo} = 0.1$

$$C_{Lo1} = 0.3283$$

$$C_{Lo2} = 0.3200$$

$$C_{Lo3} = 0.3200$$

Then from Newton Raphson, the value of C_{Lo} is

$$\text{Limit} < 0.00001$$

$$\begin{aligned} C_{Lo(new)} &= C_{Lo3} - C_{Lo2} \\ &= 0.3200 - 0.3200 \\ &= 0.00 \end{aligned}$$

So the answer for C_{Lo} is 0.3200

Then for λ is wetted length beam ratio $\lambda = \frac{L_m}{b}$ and can be determined using Newton Raphson formulation:

$$\frac{lcg}{\lambda B} - 0.75 + \left(\frac{1}{\frac{5.21Cv^2}{\lambda^2} + 2.39} \right) = 0$$

$$g = 0.7925\lambda^3 B_{wl} - 2.39\lambda^2 Lcg + 3.9075Cv^2 \lambda B_{wl} - 5.21Cv^2 Lcg$$

$$\dot{g} = 2.3775\lambda^2 B_{wl} - 4.78\lambda Lcg + 3.9075Cv^2 \lambda B_{wl}$$

$$\lambda_1 = \lambda_o - \frac{g}{\dot{g}}$$

$$\text{When } \lambda_o = 4.0$$

$$\lambda_1 = 4.4568$$

$$\lambda_2 = 4.3898$$

$$\lambda_3 = 4.3881$$

$$\lambda_4 = 4.3881$$

Then using Newton Raphson, the value of λ is

$$\text{Limit} < 0.00001$$

$$\lambda_{new} = \lambda_4 - \lambda_3$$

$$= 4.3881 - 4.3881$$

$$= 0.00$$

Then, $\lambda = 4.3881$ m

Using the values of λ and C_{Lo} , we can calculate the trim (λ) by equation below:

$$C_{Lo} = \tau^{1.1} \left(0.0120\sqrt{\lambda} + \frac{0.0055\lambda^{5/2}}{C_v^2} \right)$$

$$\begin{aligned} \tau &= \sqrt[1.1]{\frac{C_{Lo}}{\left(0.0120\sqrt{\lambda} + \frac{0.0055\lambda^{5/2}}{C_v^2} \right)}} \\ &= \sqrt[1.1]{\frac{0.3200}{\left(0.0120\sqrt{4.3881} + \frac{0.0055(4.3881)^{5/2}}{(1.5076)^2} \right)}} \\ &= 2.3921 \text{ deg} \end{aligned}$$

Savitsky gives a formula to correct the mean wetted length ratio λ , to the keel wetted length ratio, λ_k which can be calculated by:

$$\lambda_k = \lambda - 0.03 + \frac{1}{2} \left[0.57 + \frac{\beta}{1000} \right] \left[\frac{\tan\beta}{2 \tan\tau} - \frac{\beta}{167} \right]$$

$$\lambda_k = 4.3881 - 0.03 + \frac{1}{2} \left[0.57 + \frac{14\left(\frac{\pi}{180}\right)}{1000} \right] \left[\frac{\tan(14)}{2 \tan(2.3921)} - \frac{14\left(\frac{\pi}{180}\right)}{167} \right]$$

$$\lambda_k = 4.3881 - 0.03 + \frac{1}{2} [0.57 + 2.4438 \times 10^{-4}] \left[\frac{0.2493}{0.0835} - \frac{0.2438}{167} \right]$$

$$\lambda_k = 4.3881 - 0.03 + \frac{1}{2} [0.5702] [2.9856 - 1.4598 \times 10^{-3}]$$

$$\lambda_k = 5.2089 \text{ m}$$

λ_k shall be less than $\frac{L_{WL}}{b}$ so that the bow is essentially clear of the water.

Which the C_F can be estimate by:

$$\begin{aligned} C_F &= \frac{0.075}{(\log R_{nb}-2)^2} + \Delta C_F \\ &= \frac{0.075}{(\log(430.961 \times 10^6)-2)^2} + 0.0006981 \\ &= 2.4020 \times 10^{-3} \end{aligned}$$

$R_{nb} = \frac{v\lambda b}{\nu}$, $\Delta C_F = 0.0006981$ which obtained from ATTC Standard Roughness which is based on the R_{nb} number.

$$\begin{aligned} &= \frac{12.86(4.3881)(7.4168)}{9.7117 \times 10^{-7}} \\ &= 430.961 \times 10^6 \end{aligned}$$

Where weight of the vessel is:

$$W = \rho g \nabla.$$

The resistance can be predicted from the following equation:

$$\begin{aligned} R_T &= W \tan \tau + \frac{\frac{1}{2} \rho V^2 \lambda b^2 C_F}{(\cos \tau \cos \beta)} \\ &= 1025(9.81)(127.0976) \tan(2.3921) \\ &+ \frac{\frac{1}{2}(1025)(12.86)^2 (4.3881)(7.4168)^2 (2.4020 \times 10^{-3})}{[\cos(2.3921) \cos(14)]} \\ &= 53387.4722 + 50691.2447 \\ &= \mathbf{104078.72 \text{ N}} \end{aligned}$$

2.0 Resistance Prediction with Stern Foil

2.1 Foil

The formulation of the viscous resistance on the foil can be expressed as:

$$\begin{aligned} R_n &= \frac{Vc}{\nu} \\ &= \frac{12.86(1.0)}{9.7117 \times 10^{-7}} \\ &= 1.32345 \times 10^7 \end{aligned}$$

$$\begin{aligned} C_F &= \frac{0.075}{(\log_{10} R_n - 2)^2} \\ &= \frac{0.075}{(\log_{10}(1.32345 \times 10^7) - 2)^2} \\ &= 2.8591 \times 10^{-3} \end{aligned}$$

$$\begin{aligned} C_{Dv} &= 2C_F[1 + 2(t/c) + 60(t/c)^4] \\ &= 2(2.8591 \times 10^{-3})[1 + 2(0.1194/1.00) + 60(0.1194/1.00)^4] \\ &= 7.1534 \times 10^{-3} \end{aligned}$$

The viscous resistance can be calculated as follows:

$$\begin{aligned} R_v &= 0.5C_{Dv}\rho V^2 A \\ &= 0.5(7.1534 \times 10^{-3})(1025)(12.86)^2(1) \\ &= \mathbf{606.3042} \end{aligned}$$

The induced resistance can be described as follows which the foil at zero angle of attack,

$$\begin{aligned} \Lambda &= \left(\frac{4s}{\pi c_0} \right) \\ &= \left(\frac{4(7)}{\pi(1)} \right) \\ &= 8.9115 \end{aligned}$$

$$\begin{aligned} A &= \left(\frac{S^2}{\Lambda} \right) \\ &= \left(\frac{7^2}{8.9115} \right) \end{aligned}$$

$$= 5.4985$$

$$\begin{aligned} C_{Di} &= \frac{4\pi\alpha\Lambda}{(\Lambda + 2)^2} \\ &= \frac{4\pi \left(0\left(\frac{\pi}{180}\right)\right) 8.9115}{(8.9115 + 2)^2} \\ &= \mathbf{0.0} \end{aligned}$$

$$\begin{aligned} R_i &= 0.5C_{Di}\rho V^2 A \\ &= \mathbf{0.0} \end{aligned}$$

The circulation Γ for wave resistance of the foil can be calculated as follow:

$$\frac{\Gamma}{\pi c} - \frac{0.5c\Gamma}{2\pi(4h^2 + 0.25c^2)} = V \left(\alpha + \frac{2f}{c} \right)$$

At zero angle of attack and uncamber foil:

$$\begin{aligned} \frac{\Gamma}{\pi(1.0)} - \frac{0.5(1.0)\Gamma}{2\pi(4(1.005)^2 + 0.25(1.0)^2)} &= 12.86 \left((0) + \frac{2(0)}{(1.0)} \right) \\ &= \mathbf{0.0} \end{aligned}$$

The wave resistance due to the foil can be described as:

$$\begin{aligned} F_{nh} &= \frac{V}{\sqrt{gh}} \\ &= \frac{12.86}{\sqrt{9.81(1.1005)}} \\ &= 3.9139 \end{aligned}$$

$$\begin{aligned} R_w &= \frac{\rho g \Gamma^2}{V^2} e^{\left(\frac{-2}{Fn_h^2}\right)} \\ &= \frac{(1025)(9.81)(0.0)^2}{(12.86)^2} e^{\left(\frac{-2}{(3.9139)^2}\right)} \\ &= \mathbf{0.0} \end{aligned}$$

The spray resistance for foil system as follow:

$$\begin{aligned}
 R_s &= 0.12\rho V^2 t^2 \\
 &= 0.12(1025)(12.86)^2(0.1194)^2 \\
 &= 289.9984 \text{ N}
 \end{aligned}$$

This equation is valid when the Froude number based on the chord length ($Fn_c = \frac{v}{\sqrt{gc}}$) ≥ 3 .

The total resistance of the foil as follows:

$$\begin{aligned}
 R_{\text{Foil}} &= (R_v + R_i + R_w + R_s) \\
 &= \mathbf{606.3042 + 0.0 + 0.0 + 289.9984} \\
 &= \mathbf{896.3026}
 \end{aligned}$$

2.2 Strut

The formulation of the viscous resistance the strut can be described as:

$$\begin{aligned}
 R_n &= \frac{Vc}{\nu} \\
 &= \frac{12.86(0.3255)}{9.7117 \times 10^{-7}} \\
 &= 4.30784 \times 10^6
 \end{aligned}$$

$$\begin{aligned}
 C_F &= \frac{0.075}{(\log_{10} R_n - 2)^2} \\
 &= \frac{0.075}{(\log_{10}(4.30784 \times 10^6) - 2)^2} \\
 &= 3.4922 \times 10^{-3}
 \end{aligned}$$

$$\begin{aligned}
 C_{Dv} &= 2C_F[1 + 2(t/c) + 60(t/c)^4] \\
 &= 2(3.4922 \times 10^{-3})[1 + 2(0.0930/0.3255) + 60(0.0930/0.3255)^4] \\
 &= 1.3795 \times 10^{-2}
 \end{aligned}$$

The viscous resistance can be calculated as follows:

$$\begin{aligned}
 R_v &= 0.5C_{Dv}\rho V^2 A \\
 &= 0.5(1.3795 \times 10^{-2})(1025)(12.86)^2(0.3255) \\
 &= \mathbf{380.6030}
 \end{aligned}$$

The induced resistance can be described as follows which the strut at zero angle of attack. The span of strut is an equivalent to high of the strut.

$$\begin{aligned}
 \Lambda &= \left(\frac{4s}{\pi c_0} \right) \\
 &= \left(\frac{4(1.1005)}{\pi(0.3255)} \right) \\
 &= 4.3042
 \end{aligned}$$

$$\begin{aligned}
 A &= \left(\frac{S^2}{\Lambda} \right) \\
 &= \left(\frac{(1.1005)^2}{4.3042} \right) \\
 &= 0.2814
 \end{aligned}$$

$$\begin{aligned}
 C_{Di} &= \frac{4\pi\alpha\Lambda}{(\Lambda + 2)^2} \\
 &= \frac{4\pi \left(0 \left(\frac{\pi}{180} \right) \right) 4.3042}{(4.3042 + 2)^2} \\
 &= \mathbf{0.0}
 \end{aligned}$$

$$\begin{aligned}
 R_i &= 0.5C_{Di}\rho V^2 A \\
 &= \mathbf{0.0}
 \end{aligned}$$

The wave resistance for strut is considering small when $F_{nc} \geq 3.0$ which in this case F_{nc} is 3.0 so that the wave resistance equal to zero.

$$F_{nc} = \frac{V}{\sqrt{gc}}$$

$$\begin{aligned}
 &= \frac{12.86}{\sqrt{9.81(0.3255)}} \\
 &= 7.1967
 \end{aligned}$$

The spray resistance for strut as follow:

$$\begin{aligned}
 R_s &= 0.12\rho V^2 t^2 \\
 &= 0.12(1025)(12.86)^2(0.0930)^2 \\
 &= 175.9353 \text{ N}
 \end{aligned}$$

This equation is valid when the Froude number based on the chord length ($Fn_c = \frac{v}{\sqrt{gc}}$) ≥ 3 .

The total resistance of the **2** unit of strut that use as follows:

$$\begin{aligned}
 R_{\text{Strut}} &= 2(R_v + R_i + R_w + R_s) \\
 &= 2(380.6030 + 0.0 + 0.0 + 175.9353) \\
 &= 1,113.0766 \text{ N}
 \end{aligned}$$

3.0 The Combination Total Resistance

The total resistance for the vessel on planing hull taking account from resistance on barehull and the foil and strut. The total resistance at speed 25 knots (12.86 m/s) is:

$$\begin{aligned}
 R_T &= R_{\text{barehull}}(\text{vessel}) + R_{\text{foil}} + R_{\text{strut}} \\
 &= (104078.72 + 896.3026 + 1,113.0766) \\
 &= 106,088.10 \text{ N}
 \end{aligned}$$

APPENDIX D

FORTRAN Programming on Resistance Prediction

1.0 FORTRAN Programming (Source Code)

Program Resistance_Calculation

```

!NODEBUG
real LOA,Lwl,Bwl,D,T,Beta,cg,Vol,Vmin,g,rho,eu,w,corw,vmax
real hs,cs,ss,ts,ht,ct,st,th
character filename*10,yes*2,savefilename*10

print*, '
print*, 'Welcome to Resistance Prediction M - Hull'
print*, '
print*, 'Please Enter Your Ship Particular File Name'
print*, '
        read*,filename
        print*, '
        print*, 'Your Ship Particular File Name:',filename
        print*, '

! Open data file for input

Print*, "Your File Name is ",filename
open (unit=11, file=filename, status='old')
read(11,*) LOA
read(11,*) Lwl
read(11,*) Bwl
read(11,*) D
read(11,*) T
read(11,*) Beta
read(11,*) cg
read(11,*) Vol
read(11,*) Vmin

```

```

read(11,*) Vmax
read(11,*) g
read(11,*) rho
read(11,*) eu

read(11,*) hs,ht
read(11,*) cs,ct
read(11,*) ss,st
read(11,*) ts,th
read(11,*) angs,angt
read(11,*) fs,ft
read(11,*)

```

! Save output data to a user specified file

```

print*, "
Print*, "Save the data? (Y/N)"
read*,yes
if (yes.EQ."y".or.yes.EQ."Y") then
Print*, "Enter file name?"
read*,savefilename
    print*, "
open (unit = 12,file = savefilename, status="unknown")

```

```

write(12,*)'#####< INPUT DATA >#####'
    write(12,*)' '
    write(12,*)'Length Overall (m)    =',LOA
    write(12,*)'Length Waterline (m)  =',Lwl
    write(12,*)'Breadth Waterline (m)  =',Bwl
    write(12,*)'Depth (m)              =',D
    write(12,*)'Draught (m)           =',T

```

```

write(12,*)'Deadrise Angle (deg)  =',Beta
write(12,*)'Long.Center gravity(m)  =',cg
write(12,*)'Volume (m)            =',Vol
write(12,*)'Min Ship Speed (knots)  =',Vmin
write(12,*)'Max Ship Speed (knots)  =',Vmax
write(12,*)'Gravity (m/s2)         =',g
write(12,*)'Sea Water density(kg/m3) =',rho
write(12,*)'Kinematic Velocity(m2/s) =',eu
write(12,*)' '
write(12,*)'                        Foil   Strut'
write(12,*)''
write(12,*)'Height (m)             =',hs,ht
write(12,*)'Chord Length (m)       =',cs,ct
write(12,*)'Span (m)               =',ss,st
write(12,*)'Thickness (m)          =',ts,th
write(12,*)'Angle of Attack        =',angs,angt
write(12,*)'Maximum Camber         =',fs,ft

```

! Display the input on the screen and write the input on a user specified file

```

print*,'#####< INPUT DATA >#####'
print*,' '
print*,'Length Overall (m)   =',LOA
print*,'Length Waterline (m) =',Lwl
print*,'Breadth Waterline (m) =',Bwl
print*,'Depth (m)           =',D
print*,'Draught (m)         =',T
print*,'Deadrise Angle (deg) =',Beta
print*,'Long.Center gravity(m) =',cg
print*,'Volume (m)          =',Vol
print*,'Min Ship Speed (knots) =',Vmin

```

```

print*,'Max Ship Speed (knots) =',Vmax
print*,'Gravity (m/s2)    =',g
print*,'Sea Water density(kg/m3) =',rho
print*,'Kinematic Velocity(m2/s) =',eu
print*,' '
print*,'                    Foil   Strut'
print*,'"
print*,'Height (m)    =',hs,ht
print*,'Chord Length (m) =',cs,ct
print*,'Span      (m) =',ss,st
print*,'Thickness (m) =',ts,th
print*,'Angle of Attack =',angs,angt
print*,'Maximum Camber =',fs,ft
print*,'-----'

```

! Calculation Procedure

```

write(12,*)'<<<<<<<<<<<<<CALCULATIONRESULT>>>>>>>>>>>>>>>>>>'
    write(12,*) "
    if (yes.EQ."y".or.yes.EQ."Y") then
200    Vs=Vmin
        print 18,Vs
        write(12,18) Vs

18    format(',', Vs='4X,F7.2)

    Cv=(Vmin*0.5144)/(g*Bwl)**0.5
    print 5,Cv
    write(12,5) cv

```

```
5  format(' ' Cv=',5X,F7.4)
```

```
      Clb=(rho*g*Vol)/(0.5*rho*(Vmin*0.5144)**2*(Bwl)**2)
```

```
print 6,Clb
```

```
      write(12,6) Clb
```

```
6  format(' ' Clb=',3X,F7.4)
```

```
      w=0.1
```

```
120 call f(X,w,Bwl,Cv,cg)
```

```
      call fdot(Y,w,Bwl,Cv,cg)
```

```
      w1=w-X/Y
```

```
      H=w1-w
```

```
      if(ABS(H).LE.1.0E-06) then
```

```
          w=w1
```

```
          print 30,w
```

```
          write(12,30) w
```

```
30  format (' ' w=',6X,F7.4)
```

```
      else
```

```
          w=w1
```

```
          goto 120
```

```
      end if
```

```
      clo=0.1
```

```
130 call k(A,clo,Beta,Clb)
```

```
      call kdot(B,Beta,clo)
```

```

clo1=clo-A/B
C=clo1-clo
if (ABS(C).LE.1.E-07) then

clo=clo1
print 31,clo
write(12,31) clo

31  format (',' clo=',3X,F7.4)

else
clo=clo1
goto 130
end if

Trim=(clo*Cv**2/(0.012*Cv**2*(w)**0.5+0.0055*(w)**2.5))**0.91

print 7,Trim
write(12,7) Trim
7  format(',' Trim=',3X,F7.4)

call Rads(Radian,PI)
corw=w-
0.03+(0.5*(0.57+(Radian*Beta)/1000)*(Tand(Beta)/2*Tand(Trim)-
(Radian*Beta)/167))

print 8,corw

write(12,8) corw
8  format(',' corw=',3X,F7.4)

```


$$Rn = ((Vmin * 0.5144) * Bwl * w) / eu$$

$$CF = (0.075) / (\log_{10}(Rn) - 2)^{**2}$$

$$Df = 6.9810E-04$$

$$RF = (\rho * (Vmin * 0.5144)^{**2} * (w) * Bwl^{**2} * (CF + Df)) / (2 * \text{Cosd}(\text{Trim}) * \text{Cosd}(\text{Beta}))$$

!print*, 'RF=', RF

$$RW = \rho * g * Vol * \text{Tand}(\text{Trim})$$

!print*, 'RW=', RW

$$Fnhs = (Vmin * 0.5144) / (g * hs)^{**0.5}$$

!print*, 'Fnhs=', Fnhs

$$Rncs = (Vmin * 0.5144) * cs / eu$$

!print*, 'Rncs=', Rncs

$$Cfs = 0.075 / (\log_{10}(Rncs) - 2)^{**2}$$

!print*, 'Cfs=', Cfs

$$Fncs = (Vmin * 0.5144) / (g * cs)^{**0.5}$$

!print*, 'Fncs=', Fncs

$$AR = (4 * ss) / 3.1416 * cs$$

!print*, 'AR=', AR

$$A = ss^{**2} / AR$$

!print*, 'A=', A

$$Cvs = 2 * Cfs * (1 + 2 * (ts / cs) + 60 * (ts / cs)^{**4})$$

!print*, 'Cvs=', Cvs

$$Rvs = Cvs * 0.5 * \rho * (Vmin * 0.5144)^{**2} * cs$$

!print*, 'Rvs=', Rvs

$$\text{rad} = 3.1416 / 180$$

$$Cis=4*3.1416*rad*angs*AR/(AR+2)**2$$

$$Ris=0.5*Cis*rho*(Vmin*0.5144)**2*A$$

!print*, 'Ris=', Ris

$$Cir=((Vmin*0.5144)*cs*rad*angs+2*fs*(Vmin*0.5144))*(8*3.1416**2*(hs)**2+0.5*3.1416**2*(cs)**2)/(1-0.5*cs)$$

$$Rws=(rho*g*Cir**2/(Vmin*0.5144)**2)*exp(-2/Fnhs**2)$$

!print*, 'Rws=', Rws

$$Rst=0.12*rho*(Vmin*0.5144)**2*(ts)**2$$

!print*, 'Rst=', Rst

$$Rtfoil=Rvs+Ris+Rws+Rst$$

!print*, 'Rtfoil=', Rtfoil

$$Fnht=(Vmin*0.5144)/(g*ht)**0.5$$

!print*, 'Fnht=', Fnht

$$Rnct=(Vmin*0.5144)*ct/eu$$

!print*, 'Rnct=', Rnct

$$Cft=0.075/(log10(Rnct)-2)**2$$

!print*, 'Cft=', Cft

$$Fnct=(Vmin*0.5144)/(g*ct)**0.5$$

!print*, 'Fnct=', Fnct

$$AR=(4*ht)/(3.1416*ct)$$

!print*, 'AR=', AR

$$A=ht**2/AR$$

!print*, 'A=', A

$$Cvt=2*Cft*(1+2*(th/ct)+60*(th/ct)**4)$$

!print*, 'Cvt=', Cvt

```
Rvt=Cvt*0.5*rho*(Vmin*0.5144)**2*ct
```

```
!print*,Rvt=', Rvt
```

```
rad=pi/180
```

```
Cit=4*3.1416*rad*angt*AR/(AR+2)**2
```

```
Rit=0.5*Cit*rho*(Vmin*0.5144)**2*A
```

```
!print*,Rit=', Rit
```

```
Rstr=0.12*rho*(Vmin*0.5144)**2*(th)**2
```

```
!print*,Rst=', Rst
```

```
Rtstrut=2*(Rvt+Rit+Rstr)
```

```
!print*,Rtstrut=', Rtstrut
```

```
Rthull=RW+RF+RtFoil+RtStrut
```

```
print 32,Rthull
```

```
write(12,32) Rthull
```

```
32 format(' Rthull=',1X,F10.2)
```

```
write(12,*)
```

```
write(12,*)
```

```
print*,"
```

```
print*,"
```

```
if (Vmin.lt.Vmax) then
```

```
Vmin=Vmin+1
```

```
goto 200
```

```
end if
```

```
end if
```



```

return
else
B=1-0.0039*Beta*clo**(-0.4)
end if
return
end

```

```
!=====
```

```

Subroutine Rads(Radian,PI)
Radian=PI/180
return
end

```

2.0 Input Data (File Name : *q.txt*)

```

34.000
31.667
7.4168
3.300
1.365
14.000
13.5755
127.098
23
40
9.81
1025.00
9.71718E-07

```

1.1005	1.1005	
1.00	0.3255	
7.00	1.1005	
0.1194		0.093
0.0	0.0	
0.0	0.0	

3.0 Output Data (File Name : **output.txt**)

#####< INPUT DATA >#####

Length Overall (m) = 34.000000
 Length Waterline (m) = 31.667000
 Breadth Waterline (m) = 7.416800
 Depth (m) = 3.300000
 Draught (m) = 1.365000
 Deadrise Angle (deg) = 14.000000
 Long.Center gravity(m) = 13.575500
 Volume (m) = 127.098000
 Min Ship Speed (knots) = 23.000000
 Max Ship Speed (knots) = 40.000000
 Gravity (m/s²) = 9.810000
 Sea Water density(kg/m³) = 1025.000000
 Kinematic Velocity(m²/s) = 9.717180E-07

Foil Strut

Height (m) = 1.100500 1.100500
 Chord Length (m) = 1.000000 3.255000E-01

Trim= 2.3921
corw= 4.3595
Rthull= 106089.30

Vs= 26.00
Cv= 1.5679
Clb= .2534
w= 4.3025
clo= .2974
Trim= 2.4559
corw= 4.2740
Rthull= 110642.30

Vs= 27.00
Cv= 1.6283
Clb= .2350
w= 4.2162
clo= .2771
Trim= 2.5202
corw= 4.1877
Rthull= 115194.60

Vs= 28.00
Cv= 1.6886
Clb= .2185
w= 4.1297
clo= .2590
Trim= 2.5840

corw= 4.1013
Rthull= 119725.00

Vs= 29.00
Cv= 1.7489
Clb= .2037
w= 4.0438
clo= .2426
Trim= 2.6464
corw= 4.0154
Rthull= 124212.50

Vs= 30.00
Cv= 1.8092
Clb= .1904
w= 3.9591
clo= .2278
Trim= 2.7064
corw= 3.9307
Rthull= 128636.90

Vs= 31.00
Cv= 1.8695
Clb= .1783
w= 3.8762
clo= .2144
Trim= 2.7628
corw= 3.8479

Rthull= 132980.30

Vs= 32.00

Cv= 1.9298

Clb= .1673

w= 3.7958

clo= .2022

Trim= 2.8148

corw= 3.7676

Rthull= 137227.50

Vs= 33.00

Cv= 1.9901

Clb= .1573

w= 3.7185

clo= .1910

Trim= 2.8613

corw= 3.6903

Rthull= 141367.60

Vs= 34.00

Cv= 2.0504

Clb= .1482

w= 3.6446

clo= .1808

Trim= 2.9016

corw= 3.6164

Rthull= 145393.90

Vs= 35.00
Cv= 2.1107
Clb= .1399
w= 3.5744
clo= .1714
Trim= 2.9351
corw= 3.5462
Rthull= 149304.70

Vs= 36.00
Cv= 2.1710
Clb= .1322
w= 3.5082
clo= .1628
Trim= 2.9617
corw= 3.4800
Rthull= 153102.80

Vs= 37.00
Cv= 2.2313
Clb= .1251
w= 3.4460
clo= .1549
Trim= 2.9811
corw= 3.4178
Rthull= 156795.00

Vs= 38.00
Cv= 2.2916
Clb= .1186
w= 3.3878
clo= .1475
Trim= 2.9935
corw= 3.3596
Rthull= 160391.60

Vs= 39.00
Cv= 2.3519
Clb= .1126
w= 3.3335
clo= .1407
Trim= 2.9991
corw= 3.3053
Rthull= 163905.40

Vs= 40.00
Cv= 2.4122
Clb= .1071
w= 3.2830
clo= .1344
Trim= 2.9984
corw= 3.2548
Rthull= 167350.80

APPENDIX E

Offset Table 130.275 Tonnes High Speed Craft

LONGI	STN	0	0.417	0.83	1.242	1.365	1.655	2.067	2.48	2.892	3.293
		BL	WL1	WL2	WL3	WL4	WL5	WL6	WL7	WL8	WL9
-15.297	1(AP)			3.454	3.52	3.54	3.579	3.627	3.674		
-14.425	2			3.456	3.522	3.542	3.58	3.628	3.675		
-13.553	3			3.456	3.524	3.544	3.582	3.63	3.675		
-12.681	4			3.432	3.521	3.545	3.584	3.63	3.675		
-11.81	5		0.601	3.432	3.519	3.544	3.583	3.629	3.675		
-10.938	6		0.701	3.432	3.518	3.542	3.581	3.626	3.674		
-10.066	7		0.766	3.432	3.515	3.535	3.572	3.619	3.672		
-9.194	8		0.824	3.432	3.512	3.528	3.564	3.613	3.671		
-8.322	9		0.877	3.432	3.511	3.531	3.567	3.615	3.671		
-7.451	10		0.925	3.432	3.51	3.533	3.57	3.616	3.671		
-6.579	11		0.967	3.432	3.509	3.529	3.564	3.609	3.669		
-5.707	12		1.005	3.432	3.508	3.526	3.559	3.602	3.668		
-4.835	13		1.059	3.432	3.505	3.524	3.562	3.61	3.67		
-3.963	14		1.107	3.432	3.502	3.523	3.564	3.618	3.672		
-3.092	15		1.128	3.432	3.507	3.529	3.57	3.62	3.673		
-2.22	16		1.146	3.432	3.512	3.536	3.575	3.622	3.673		
-1.348	17		1.165	3.326	3.511	3.534	3.573	3.617	3.671		
-0.476	18		1.182	3.316	3.509	3.533	3.57	3.611	3.669		
0.396	19		1.192	3.236	3.482	3.506	3.549	3.602	3.667		
1.267	20		1.193	3.172	3.46	3.486	3.534	3.596	3.666		
2.139	21		1.187	3.124	3.444	3.471	3.526	3.596	3.667		
3.011	22		1.186	2.918	3.367	3.404	3.48	3.579	3.667		
3.883	23		1.185	2.735	3.286	3.334	3.433	3.56	3.667		
4.755	24		1.152	2.547	3.17	3.225	3.344	3.498	3.644		
5.626	25		1.106	2.364	3.03	3.091	3.227	3.405	3.607		
6.498	26		1.059	2.195	2.885	2.952	3.105	3.309	3.565		
7.37	27		0.986	1.991	2.7	2.777	2.957	3.198	3.496		
8.242	28		0.919	1.812	2.508	2.595	2.802	3.082	3.426		
9.114	29		0.81	1.601	2.279	2.378	2.612	2.944	3.331		
9.985	30		0.698	1.392	2.03	2.142	2.405	2.793	3.227		
10.857	31		0.593	1.187	1.668	1.87	2.142	2.557	3.036		
11.729	32		0.494	0.979	1.383	1.511	1.826	2.227	2.75		
12.601	33		0.398	0.781	1.112	1.211	1.508	1.874	2.422	2.744	2.645
13.472	34		0.291	0.576	0.829	0.904	1.158	1.501	2.011	2.444	2.324
14.344	35		0.149	0.327	0.518	0.576	0.77	1.075	1.53	2.021	1.853
15.216	36(FP)			0.081	0.229	0.276	0.39	0.67	1.018	1.548	1.382
16.088	37					0.003	0.104	0.306	0.549	0.97	0.867
16.96	38								0.147	0.415	0.407