ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL OF SCIENCE ENGINEERING AND TECHNOLOGY

INVESTIGATION OF SCALE EFFECTS IN SAILING YACHT HYDRODYNAMIC TESTING

Ph.D. THESIS

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Department of Naval Architecture and Marine Engineering

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İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

YELKENLİ TEKNE HİDRODİNAMİK DENEYLERİNDE ÖLÇEK ETKİLERİNİN İNCELENMESİ

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June 2013 A. Ziya SAYDAM
Naval Architect

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ABBREVIATIONS

A	**:** Planform Area
AC	**:** America's Cup
Are	**:** Effective Aspect Ratio
AIAA	**:** American Institute of Aeronautics and Astronautics
ASME	**:** American Society of Mechanical Engineers
B	: Beam of Hull
Bdia	: Bulb Diameter
Blen	: Bulb Length
c	: Foil Chord Length
CAD	: Computer Aided Design
$\mathbf{C_B}$	: Block Coefficient
$\mathbf{C_D}$	:Drag Coefficient
$\mathbf{C_F}$	: Frictional Resistance Coefficient
$\mathbf{C_{FM}}$	: Model Frictional Resistance Coefficient
$\mathbf{C_L}$	**:** Lift Coefficient
$\mathbf{C_M}$	:Midship Coefficient
$\mathbf{C_T}$	**:** Total Resistance Coefficient
$\mathbf{C_W}$	: Wave Resistance Coefficient
$\mathbf{C_{WP}}$	: Wave Pattern Resistance Coefficient
$\mathbf{C_{TM}}$	: Model Total Resistance Coefficient
CFD	**:** Computational Fluid Dynamics
CLR	**:** Center of Lateral Resistance
CNC	**:** Computer Numeric Control
DNS	**:** Direct Numerical Simulation
DTNSRDC	**:** David Taylor Naval Ships Research & Development Center
Fn,Fr	**:** Froude Number
g	**:** Gravitational Acceleration
H	**:** Water Depth
IACC	**:** International America's Cup Class
IEEE	: Institute of Electrical and Electronics Engineers
ITTC	**:** International Towing Tank Conference
IMS	**:** International Measurement System
k	**:** Form Factor
$\mathbf{k}_H$, $\mathbf{k}_K$, $\mathbf{k}_B$, $\mathbf{k}_R$	: Form Factors for Hull, Keel, Bulb and Rudder Respectively
Km	: Wave Number
Loa	**:** Overall Length
LCF	**:** Longitudinal Center of Floatation
LES	: Large Eddy Simulation
LWL	**:** Water Line Length
LPT	**:** Linear Position Transducer
m	**:** Harmonic Number
MIT	**:** Massachusetts Institute of Technology

P	**:** Pressure
PACT	**:** Partnership for America's Cup Technology
PIV	**:** Particle Image Velocimetry
RANS	**:** Reynolds Averaged Navier Stokes
Re	**:** Reynolds Number
$\mathbf{R_F}$	**:** Frictional Resistance
$\mathbf{R_H}$	**:** Heel Resistance
$\mathbf{R_I}$	**:** Induced Resistance
RMP	**:** Race Model Program
$\mathbf{R_V}$	**:** Viscous Resistance
$\mathbf{R_W}$	**:** Wave Resistance
$\mathbf{R_T}$	**:** Total Resistance
S	**:** Wetted Surface Area
$\mathbf{S_M}$	: Model Wetted Surface Area
t	: Foil Thickness
Te	**:** Effective Draft
TP52	: Transpac 52
V	: Velocity
VPP	**:** Velocity Prediction Program
W	**:** Tank Width
WS	**:** Wetted Surface Area
$\boldsymbol{\alpha}$	**:** Induced Drag Curve Slope
$\boldsymbol{\rho}$	**:** Density
$\boldsymbol{\mu}$	**:** Kinematic Viscosity
$\boldsymbol{\Theta_M}$	**:** Wave Number

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INVESTIGATION OF SCALE EFFECTS IN SAILING YACHT HYDRODYNAMIC TESTING

SUMMARY

Towing tank experiments on geometrically similar models of a prototype have been used by naval architects to predict the performance of yachts to mitigate the risk of performance penalties. Although this technique has been utilized for over a century, the data obtained from the towing tank tests are still treated as “engineering estimates” rather than “exact scientific results”. This may be sufficient for an engineering approach for certain types of vessels, however it has never been acceptable in the case of sailing yacht testing.

Performance of sailing yachts have been estimated by balancing hydrodynamic and aerodynamic forces and moments with the aid of a Velocity Prediction Programme (VPP). There have been major progresses in terms of VPP software development and estimation of forces with Computational Fluid Dynamics (CFD) methods, but no tool has managed to replace experiments upto now. CFD analyses are widely used for local flow investigation, flow visualisations, comparison of design alternatives in short time but sufficient confidence has not been gained to eliminate the need of experimentation.

Under these circumstances, it is of vital importance for the yacht designers to solely rely on experiments and have confidence on the results so that the model results may be extrapolated to full scale with suitable tools. Procedures like ITTC recommendations, which are developed mainly for conventional ships, give reasonably accurate results in terms of powering with the large margins. Although they are suitable for ship powering purposes, they are not so suitable for sailing yacht testing owing to differences in expectations of accuracy and existance of scale effects experienced by the hull and the relatively larger appendages. Therefore, it is required to explore the trends in the variation of sideforce, viscous and wave resistances with scale and assess the suitability of the currently available extrapolation methods for sailing yacht experiments. It is worth noting that studies in the field are no longer up-to-date and contemporary experimental and numerical studies in the field are required which would make best use of currently available experimental methods and state-of-the-art CFD tools in order to have a better understanding of the scale effects experienced during tank testing of sailing yachts.

Another vital consideration in towing tank testing campaigns of sailing yachts is the cost versus accuracy trade off issue. As the advancements in terms of testing and extrapolation techniques force experimenters to test large models –especially for appendage development studies- leading to extreme budgets and towing tank run times. A systematic study of scale effects with emphasis on hull and appendage development studies would give a better understanding of cost versus scale trade off in this field.

The study has been initiated with a broad survey on the literature in the field, going back to Watson's tests in 1901. The efforts in improving the testing and extrapolation techniques associated with yacht testing since then has been analyzed. It's been seen that the methods have improved over the years, certain assumptions were replaced with more accurate estimates, model manufacturing, measurement tools and testing tehniques were improved upto a certain extent.

As the techniques were changing, the hull form and appendage considerations were also differing from era to era, neccesitating the renewal of the resarch in the field every now and then. Therefore, it has been decided to conduct the research on modern hull and appendage configurations.

The testing program was developed with consideration of deficiencies of the current state of the art and the capabilities of the facility. All testing were performed at the large towing tank at Ata Nutku Ship Model Testing Laboratory of Istanbul Technical University. As a single post dynamometer system was available for force measurement, the maximum model size was limited to 4 meters. The decision was made to test a TP52 type yacht, hence the maximum scale to be tested was 1/4 at a length of 3.96 meters. The smallest model scale was chosen to be unconventionally small in order to track the differences at a length of 1.58 meters corresponding to 1/10 scale. As a result, four models of a TP52 yacht has been tested, with scales 1/10, 1/8, 1/6 and 1/4.

During the tests, resistance and side force were measured by the 6-component dynamometer. The system enabled the models to freely heave and trim, these were measured by the aid of a linear displacement transducer and a potentiometer respectively.

As the main concern of the tests was to investigate the differences in the measured parametes with respect to scale, it was decided that total force measurement in terms of resistance would not be sufficient to draw the conclusions. Therefore, it was decided to measure the wave pattern of the models. The wave patterns were analyzed with the longitudinal wave cut technique, they were measured by the aid of three probes on each side of the tank.

The tests were conducted in upright condition for naked and appended configurations. For the heeled cases, only appended models were tested.

CFD analysis were also performed during the research. A Reynolds Averaged Navier Stokes (RANS) code has been used for the calculations. The main objective of the CFD analysis was to assess the similarity of the variation of trends in investigated paramaters compared to the experiments.

In the upright condition, variation of hull and appendage resistance and wave profile were investigated at scales 1/8 and 1/4. In the heeled condition, variation of heel drag and lift generation efficiency (induced drag generation characteristics) were assessed at scales 1/8, 1/4 and full scale.

The results of the study were found to agree with the available literature. Even though this topic has been the subject of research for many in the past, it has been reconsidered with a genuine methodology in a systematic manner with combining experiments and numerical methods and the results were found to be benefical for consideration in future testing campaigns.

YELKENLİ TEKNE HİDRODİNAMİK DENEYLERİNDE ÖLÇEK ETKİSİNİN İNCELENMESİ

ÖZET

Hidrodinamik model deneyleri gemi ve yat tasarımcıları tarafından, test edilen model ile geometrik olarak benzer büyük prototiplerin performansının tahmin edilmesi maksadıyla kullanılmaktadır. Bu deneylerin geçmişi yüz yıldan fazla olsa da, günümüzde hala deneylerden elde edilen verilen "gerçek bilimsel sonuçlar" olmaktan çok "mühendislik tahmini" olarak nitelendirilmektedir. Bu durum, birçok gemi ve deniz aracı tipi için mühendislik açıdan yeterli bulunmakta olup yelkenli tekneler için kabul edilmesi mümkün olamamaktadır.

Yelkenli tekne performans tahmini hidrodinamik ve aerodinamik kuvvetlerin deneylerle hesaplanıp Hız Tahmin Programı (Velocity Prediction Program, VPP) vasıtasıyla dengelenmesi esasına dayanmaktadır. Bu alanda gerek VPP yazılımlarında gerek kuvvetlerin deneyler ile ölçülmesi yerine hesaplamalı akışkanlar dinamiği ile hesaplanabilmesi gibi gelişmeler olsa da henüz deneylerin tamamen yerini alacak bir metod geliştirilebilmiş değildir. Yapılan hesaplamalı akışkan dinamiği analizleri akım hatları görüntüleme, birden çok modeli kısa zamanda karşılaştırmalı olarak deneme gibi avantajlar sunsa da henüz deneylerin yerini alabilecek güven telkin etmemektedirler.

Bu şartlarda tasarımcıların deney sonuçlarına güvenebilmeleri ve doğru metodlarla model üzerinde ölçülen kuvvetleri tam ölçeğe ekstrapolasyonunu yapabilmeleri önem kazanmaktadır. Gemi direnci için Uluslararası Deney Havuzu Konferansı (International Towing Tank Conference, ITTC) gibi prosedürlerle yapılan direnç tahminleri ekstrapolasyon sırasında bırakılan büyük marjlarla makine seçimi ve dizayn hızının sağlanması için yeterli sonuçlar vermektedir. Fakat, aynı yöntemlerin hiçbir değişiklik olmadan yelkenli teknelere uygulanması performans tahmini açısından sıkıntılı sonuçlar doğurmaktadır. Performans tahmininde gereken hassasiyet, gemilerden farklı olarak salma ve fin gibi büyük takıntıların maruz kaldığı ölçek etkisi bu sıkıntıların başında gelmektedir.

Yanal kuvvet, viskoz ve dalga direnci komponentlerinin model ve tam ölçek arasında nasıl farklılıklar gösterdiğinin analiz edilerek mevcut ekstrapolasyon yöntemlerinin irdelenerek yelkenli tekneleri uyarlanması gerekmektedir. Bu alandaki çalışmalar güncelliğini yitirmiş olup, modern tekne formları ve takıntılar gözönüne alınarak, son yıllardaki test tekniklerindeki gelişmeler ışığında ve mevcut son teknoloji hesaplamalı akışkanlar dinamiği yazılımları yardımıyla yapılacak deneysel ve hesaplamalı bir çalışmanın ölçek etkilerinin anlaşılması konusuna fayda sağlayacağı öngörülmektedir.

Yelkenli tekne hidrodinamik deneylerinde göz önüne alınması gereken önemli bir diğer nokta maliyet ve doğruluk dengesinin sağlanmasıdır. Deney ve ekstrapolasyon tekniklerindeki gelişmeler büyük model kullanımını zorunlu kıldığından (özellikle takıntı geliştirme için) oluşan maliyetler artmakta olup deney sürelerinde uzamalar

meydana gelmektedir. Tekne ve takıntı geliştirme odaklı sistematik bir ölçek etkisi araştırma çalışmasının, tasarımcılar ve deney yapıcılar için ölçek, maliyet ve doğruluk arasındaki ilişkinin daha iyi anlaşılması ve beklentiler ile örtüşen bir test programının bütçe kısıtları doğrultusunda oluşturulabilmesi için faydalı olacağı görülmektedir.

Çalışmanın başlangıç aşamasını yelkenli tekne deneylerinde kronolojik gelişmelerin araştırılması oluşturmaktadır. 1901 yılında Watson tarafından yapılan deneylerden bu yana, deney teknikleri ve ekstrapolasyon tekniklerinde elde edilen tüm gelişmeler bu kapsamda incelenmiş ve metodların zaman içerisindeki gelişimi irdelenmiştir. Geçen süre boyunca model üretimi, kalite kontrol, deney ekipmanları, deney teknikleri, ekstrapolasyon tekniklerinin belirli bir noktaya kadar ilerlediği görülmüş, fakat temel problemlerin genele uygulanabilir bilimsel bir metodoloji ile çözülememiş olduğu belirlenmiştir.

Tekniklerdeki gelişmelere paralel olarak, tekne formları ve kullanılan takıntıların karakteristikleri de zaman içinde değişim göstermiştir. Özellikle Amerika Kupası yarışları için yapılan araştırma ve geliştirme çalışmalarına paralel olarak bu yeniliklerin ortaya çıktığı görülmektedir. Tekne ve takıntı karakteristiklerindeki değişiklikler dolayısıyla ölçek etkisi incelemeleri zaman içinde demode kalmakta ve özellikle yenilenen takıntılar için araştırmaların yenilenmesi gereksinimi doğabilmektedir.

Deney programı güncel metotların eksiklikleri ve kullanılacak deney ortamının kapasite ve kabiliyetleri göz önüne alınarak oluşturulmuştur. Tüm deneyler İstanbul Teknik Üniversitesi Gemi İnşa ve Deniz Bilimleri Fakültesi Ata Nutku Gemi Model Deney Laboratuarı'nda bulunan büyük deney havuzunda yapılmıştır. 160 metre uzunluğundaki havuzda, testler esnasında 5 m/s hıza kadar çıkılabilmektedir. Deney havuzunda kuvvet ölçümleri için tek nokta bağlantılı 6 bileşenli dinamometre bulunmaktadır.

Model ölçeğinde ulaşılması gerekilen hızlar, blokaj etkileri, tek noktadan bağlantılı ölçüm sistemi sınırlandırmaları ve havuzda yapılmış olan geçmiş deney tecrübeleri göz önüne alındığında test edilebilecek azami yelkenli tekne boyunun 4 metre mertebesinde olması gerektiği tespit edilmiştir. Deneylerde TP 52 tipi yelkenli tekne modelleri kullanılmasına karar verildiğinden (Loa=15.85 metre), test edilebilecek en büyük tekne modelinin 1/4 ölçekte 3.96 metre boyunda olmasına karar verilmiştir. Test edilecek en küçük modelin boyutuna karar verilirken geçmiş tecrübelerde kullanılmış olan en küçük model boyutunun altına inilmesi ve bu boyutlarda oluşan farklılıkların tespit edilmesi amaçlanmıştır. Sonuç olarak en küçük modelin 1/10 ölçekte 1.58 metre boyunda olmasına karar verilmiştir. Toplam 4 adet, ölçekleri 1/10, 1/8, 1/6 ve 1/4 olan TP 52 tipi yelkenli tekne modelleri test edilmiştir.

Deneylerde direnç ve yanal kuvvet ölçümleri için 6 bileşenli dinamometre kullanılmıştır. Bu sistem, modellerin dalıp çıkma ve baş kıç vurma hareketlerine serbest kalmasına müsade etmektedir. Dalıp çıkma lineer yer değiştirme ölçeri, baş kıç vurma ise açı potansiyometresi yardımıyla deneyler esnasında ölçülmüştür.

Deneylerin ana amacı ölçülen parametrelerin ölçek ile değişiminin incelenmesi olduğundan, direnç parametresinin toplam olarak ölçülmesinin istenilen analizlerin ve beklenen çıkarımların yapılabilmesi için yetersiz olabileceği öngörüldüğünden teknelerin dalga formlarının ölçülmesi ve ilgili dirençlerinin hesap edilmesine karar verilmiştir. Dalga formları, modellerin her iki tarafında üçer adet dalga probu yardımıyla deneyler esnasında ölçülmüş ve analizler için “boyuna dalga kesi” metodu kullanılmıştır.

Deneyler, meyilsiz durumda çıplak tekne ve takıntılı koşullarda, meyilli durumda ise sadece takıntılı koşulda yapılmıştır. Tüm deneyler tam yüklü ağırlık durumu için yapılmıştır.

Çalışmada, deneylere elde edilen sonuçları tamamlayıcı olması maksadıyla hesaplamalı akışkanlar dinamiği analizlerine de yer verilmiştir. Yapılan hesaplamalı çalışmalarda, incelenen parametrelerdeki ölçeğe bağlı değişim eğilimlerinin deneylerde elde edilen sonuçlar ile karşılaştırılması amaçlanmıştır. Ayrıca, deneylerde incelenemeyen tam ölçek durumu için analizlere yer verilmiştir.

Hesaplamalı çalışmalarda, Reynolds Averajlı Navier Stokes (Reynolds Averaged Navier Stokes, RANS) metodu kullanılmış olup meyilsiz durumda tekne ve takıntı direnç ve dalga profili değişimleri 1/8 ve 1/4 ölçekler için incelenmiştir. Meyilli durumda ise, meyil direnci ve taşıma kuvveti üretim verimlerinin ölçek ile değişimleri 1/8, 1/4 ve tam ölçek için incelenmiştir.

Deneysel çalışmalar sonucunda, yelkenli tekne modellerinin hidrodinamik deneylerinde maruz kalınan ölçek etkileri, TP 52 tipi bir tekne için tespit edilebilmiştir. Teknenin, meyilsiz durumda test edilen farklı ölçekteki modellerinin dalga direnci, dalga formu ve dinamik davranışlarındaki farklılıklar ortaya çıkartılmıştır. Özellikle takınıtlı koşulda, modeller arası farklılıkların olası sebepleri incelenmiş olup, mevcut ölçüm sistemi ile elde edilen veriler ışığında bu sebeplerin bir kısmının tespit edilmesi mümkün olmuştur. Yapılan analiz sonucunda, yelkenli tekne model deneylerinde –özellikle yüksek süratlerde- maruz kalınan dinamik hareketler dolayısıyla oluşan ıslak alan değişiminin direnç ayırıştırma ve ekstrapolasyon sürecinde göz önüne alınması gerekliliği ortaya çıkarılmıştır.

Yapılan meyilli durumdaki testlerde ise, yanal kuvvet ve indüklenmiş direnç üretimi eğiliminin ölçek ile değişimi ile ilgili farklılılar tespit edilmiştir.

Deneysel olarak varlığı tespit edilen fakat kaynakları mevcut deneysel veriler ışığında tanımlanamayan ölçek etkilerinin belirlenebilmesi maksadıyla yapılan hesaplamalı çalışmalarda, özellikle takıntı dirençlerinin hesap edilmesi için kullanılan ampirik formülasyonların, yüksek süratlerde oluşan trime, takıntı-tekne arası ve takıntılar arası etkileşimlere hassas olmadığı ve bu sebeplerden dolayı direnç ayırıştırma ve ekstrapolasyon işlemlerinde hatalar oluştuğu tespit edilmiştir.

Tamamlanan çalışma sonucunda, yelkenli tekne model deneylerinde direnç ayırıştırma ve ekstrapolasyon için kullanılan metodolojinin eksiklikleri, kullanılan ampirik formülasyonların yetersizlikleri anlaşılmış ve bu eksiklik ve yetersizliklerin giderilmesi için öneriler sunulmuştur. Meyilsiz ve meyilli durumdaki testler için ölçek belirleme sürecinde model ölçeği, doğruluk ve bütçe arasındaki ilişki tanımlanmış ve proje beklentileri göz önüne alınarak ölçek belirleme süreci için tavsiyeler verilmiştir.

Yapılan çalışmada elde edilen sonuçların literatür ile örtüşmekte olduğu görülmüştür. Araştırma konusunun, geçen yıllar boyunca birçok araştırmacı tarafından irdelenmiş olmasına rağmen, konu özgün bir metodoloji ile deneysel ve hesaplamalı çalışmaları sistematik bir biçimde birleştirmek suretiyle ele alınmış ve sonuçların gelecekte yapılacak test çalışmaları için dikkate alınacak nitelikte olduğu tespit edilmiştir.

1. INTRODUCTION

Realistic physical model testing of sailing yachts is a difficult process due to simulation of sailing conditions with the heeled hull advancing with an angle of attack. Due to the large number of tests required to cover the practical heel and leeway angles in addition to upright tests, it is a time consuming process. However towing tank testing is still the most reliable method in order to select the best performing boat amongst the alternatives.

Small geosim models are utilized to minimize the cost of such testing procedure, even though scaling of sailing yacht test results presents problems in comparison to those of conventional ship tank testing. Existence of keel-bulb-rudder configurations and generation of side force in small models present difficulties on the viscous resistance similarity assumptions, especially in higher leeway angles. Hence, scaling procedure is one of the primary concerns for sailing yacht testing campaigns.

A substantial amount of research has been carried out in the past to enhance the testing techniques and to increase the accuracy associated with tank testing of sailing yachts. The majority of this work was associated with high cost campaigns; large models, long waiting times and high budgets became standard practice in the field. This led to lack of accessibility for lower budget campaigns and for designers of ordinary sailing yachts to these tests.

New sailing yacht forms, new appendage configurations are developed for various purposes. Tank testing for sailing yacht racing campaigns such as America's Cup caused dramatic increases in the model sizes and testing costs, as the effects of different appendage configurations on the performance of the full scale yachts with small models has not produced acceptable results (Campbell & Claughton, 1987). Also, hull forms and most importantly appendage configurations differ from the past when majority of the scale effect investigations have taken place.

DeBord, Kirkman and Savitsky recommended that given these advancements and hence the increased cost of the tests, renewed research on scale effects with modern hull forms equipped with highly loaded appendage configurations would be beneficial in terms of making best use of tank test data and assessment of cost versus scale trade off issues (Debord et al. , 2004).

The current work attempts to advance the knowledge on scale effects experienced in sailing yacht testing by using a series of 4 geosim models ranging from 1/10 scale to 1/4 scale of a TP52 yacht. Variation of resistance components due to viscous and wave sources, heel resistance, induced resistance and side force as well as running attitude within the model scale range have been demonstrated systematically. CFD analyses were used to verify the findings from the experiments.

1.1 Aims and Objectives

The aim of this thesis is to investigate the possible reasons of scale effects in sailing yacht tank testing with particular emphasis on:

- Variation of appendage lift and drag with scale,
- Variation of form factor & viscous resistance with scale,
- Variation of wave resistance with scale,
- Suitability of current testing and extrapolation methods for use in sailing yacht testing.

All these goals which are crucial to the performance analysis of a sailing yacht will be investigated in different model scales in order to obtain sound relations in the variation of these parameters with scale.

During the phases of the project, it is intended to achieve the aforementioned aims in accordance with the below objectives:

- To make best use of state of the art CFD tools in understanding the scale effect phenomenon,
- To conduct the experiments on modern yacht hull & appendage combinations to capture peculiar features associated with them,
- To aim at reducing tank testing costs with proper assessment of accuracy of small scale experiments,

- To enhance testing & data extrapolation techniques in order to achieve better accuracy.

1.2 Methodology

A research study has been initiated to investigate the scale effects associated with tank testing of sailing yachts. The intention has been to make best use of modern experimental and computational methods to understand the scale effects in conjunction with systematic tank tests.

This study is comprised of two parts, which are designed to support each other.

In order to divide the total resistance into components and to understand the effects of scale on each component, a test program consisting of total resistance, wave pattern, running trim and sinkage measurements across the speed, heel angle and leeway angle was utilized.

The experimental part of the study covers scales 1/10, 1/8, 1/6 and 1/4. The models were tested at fully loaded sailing trim condition both in upright and heeled conditions.

The upright tests were conducted for both naked and appended configurations whereas the heeled tests were conducted only in the appended configuration. A single post, six-component dynamometer was used for resistance and side force measurement enabling the models to freely heave and pitch. The sinkage, trim and the wave pattern were measured additionally. Longitudinal wave cut technique was used for wave pattern analysis; the wave pattern was measured by the aid of three probes on each side of the models.

During the upright tests, changes in:

- Total resistance,
- Wave resistance,
- Wave pattern resistance,
- Wave breaking resistance,
- Wave profile,
- Sinkage,

- Trim were investigated. In the heeled condition, the main parameters of interest were:
- Heel drag,
- Induced drag,
- Side force.

The CFD analyses are performed for two main purposes. Firstly, CFD has been utilized to investigate the detailed flow characteristics such as pressure, velocity distributions, hull-appendage interference, appendage-appendage interactions that could not be performed easily in the experimental programme due to difficulties of physical models. Secondly, larger scaled models up to full scale could not be tested; CFD has been utilized to understand the scale effects at these model scales, which could not be tested physically.

The CFD study was conducted at scales 1/8, 1/4 and 1/1. With the RANS code utilized, fully turbulent numerical studies could be conducted with consideration of the free surface where applicable. With the CFD methodology, it was possible to explore the variation of the above-mentioned parameters on a broader scale range, even up to full scale.

In the upright condition, resistance and wave profile was investigated for scales 1/8 and 1/4 for Froude Number 0.5.

The lift generation characteristics and the lift generation efficiencies (induced drag production characteristics) were investigated for scales 1/8, 1/4 and full scale for 9 knots of full-scale speed. These analyses were conducted for the underwater part of the hull and appendages only.

2. PREVIOUS DEVELOPMENTS AND CURRENT STATE-OF-THE -ART

2.1 Developments in Hydrodynamic Testing of Sailing Yachts

Since the beginning of the past century, starting with the upright resistance tests of America's Cup challenger Shamrock II in 1901 conducted by Watson, towing tank tests have been widely utilized for the design of sailing vessels. It is accepted by the yacht designers that tank testing is a valuable tool for design performance assessment. Since then, a large number of advancements have been achieved in the understanding of the physics behind hydrodynamics, test procedures, model manufacturing methods, model scales, measuring equipment, interpretation of the results and data extrapolation, but major problems are still the same and they are still not sufficiently understood. Even today, after 110 years of development, tank testing cannot give exact answers to designers; it rather gives an insight to the problem being investigated. In other words, rather than obtaining the exact performance of each prototype, "towing tanks are valuable means for selecting between alternative design configurations for sailing yacht hulls" (Kirkman & Pedrick, 1974).

One of the main reasons preventing tank testing from being an exact science is the unsolved issue of scale effects. The existence of the so-called scale effects raises concerns about the inadequacy of the methods in predicting full-scale performance. According to Kirkman, in order to accurately predict full scale performance from towing tank experimental data, the model flow conditions either has to be exactly similar to the prototype's or it should vary in a systematic and evident fashion so that the variations may be precisely accounted for and model measurements should exactly be converted to prototype's performance predictions. He defines scales effects as "any shortcoming in the ability of the experiment to conform to these requirements" (Kirkman & Pedrick, 1974). He adds that results from towing tank experiments should be treated as engineering estimates rather than accurate scientific predictions.

Tank testing methods have advanced especially in the last 40 years. All these advances were efforts to find ways to make best use of tank test data in the existence of scale effects and experimental inaccuracies. Many of the developments in the field of tank testing have been related to America's Cup campaigns. After Watson's upright resistance tests for Shamrock II America's Cup campaign, in 1932 Davidson has stated that only upright testing was not sufficient to assess windward performance. He is the first researcher to consider the effect of heel on hydrodynamic drag in his experiments (Davidson, 1936). From that day on, until early 70's, nearly all the campaigns including defenders and challengers used tank testing in their design evaluations (DeBord & Teeters, 1990). This situation changed after questionable performance of several competitors raised doubts on the accuracy of towing tank experiments in the 1970 campaign. Prior practice amongst competitors was then to test models at 1/13 scale (DeBord, et al. , 2004).

In 1974, Kirkman et al have published a comprehensive work on scale effects in tank testing consisting of compilation of tank tests results from various facilities of different sized models of 5.5 meter yacht Antiope, One Tonner Yacht Bullet, 12 meter yacht Intrepid, 12 meter yacht Gleam, J Class yacht Ranger, 6 meter yacht Jack and 12 meter yachts Valiant, Stormy Weather, Gimcrack and 5.5 meter yacht Bingo. Their findings led to a better understanding of scaling and resistance components of sailing yacht hulls. They have concluded that models with water line lengths of less than 12-15 feet possess poor correlation with full scale results due to tremendous sensitivity to turbulence stimulation which lacks existing correlation methods to accurately predict full scale resistance (Kirkman, 1974). Even though this effort was not sufficient to boost the popularity of towing tanks in the yachting world, this benchmark study ended the era of 1/13 scale testing in America's Cup campaigns. In 1977 Enterprise AC campaign models of 1/3 and 1/8 scale have been tested.

In 1985, Kirkman published another paper to emphasize the power of towing tanks and their important role in yacht design. He has stated that "the yachtsman's affair with tank testing has run hot and cold based upon misunderstandings of the correct role of model testing and economic considerations" (Kirkman K. , 1985). He notes that current studies improved the use of towing tanks for design assistance and proper use of the towing tank makes it a necessary and convenient tool that should be incorporated in the design loop. His efforts were aiming to highlight the changing role of the towing tank within the light shed by the contemporary understanding of the scale effects.

Unfortunately, efforts of Kirkman were not sufficient to overcome conservatism. The situation deteriorated until 1983 when no tank testing was made for the Liberty campaign.

The towing tanks have regained popularity amongst the America's Cup campaigns after Australia II; a scientifically developed yacht has won the America's Cup in 1983. In 1985, van Oossanen has published the design process and technology involved in the development of the winning yacht with particular emphasis on towing tank testing procedures and data extrapolation methods (van Oossanen, 1985). From then on, all candidates were tested with 1/3 – 1/4 scale models in the America's Cup.

In 1987, Campbell and Claughton have published Wolfson Unit's testing techniques and showed results from 12-meter class yacht experiments at 1/4 and 1/10 scale. They compared drag breakdown for different scale models with different keel configurations. They also discussed scaling issues and concluded, "although the extrapolation of model data for conventional yachts has been shown to be a tolerably reliable exercise, the problem of adequately scaling the performance of small, heavily loaded appendages, such as keel wings, is not straightforward" (Campbell & Claughton, 1987). They propose that flow visualization and measurements for establishment of information on how these appendages perform at full scale would allow more realistic viscous resistance estimation while extrapolating. They also conclude that 1/10 scale testing may be sufficient up to the extent of accuracy limitations and depending on the level of expectations; and 1/4 scale model testing was necessary in developing the challenging wing keel. They state that the designers need better understand the limitations and extents of the data that may be obtained from the experiments and the experimenters should have a better idea of the design problem so that they know the expectations of the designer and do not mislead them with the data provided.

Same year, Seragg et al published their work on the design development of the 12 meter Stars & Stripes where they aimed at minimizing all forms of hydrodynamic drag. They discuss procedures to compute, estimate and reduce wave resistance of the America's Cup yacht Stars & Stripes (Seragg, et al., 1987). They have adopted an optimization procedure for the constraints of the campaign with the use of VPP12 (12 meter velocity prediction programme), ACUPS (America's Cup Racing Simulation Program) and OPTIMIZE (hull optimization code). The output was in the form of the best suitable sectional area curve that would guide the designers while drawing the hull lines. The hulls derived from the sectional area curves were then analyzed for wave resistance by the WAVE DRAG code. They have verified their findings in both model and full scale.

Letcher et al, from the same Stars & Stripes campaign have developed a full-scale data collection system, "around Ockam on board instruments with a link to two DEC Microvax II computers, one aboard the tender and one on shore". Their aim was "validation and improvement of computer models and towing tank tests used in design; discrimination of performance differences associated with hull, keel and sail changes; and development of on board computer systems to assist decision making during races" (Letcher & McCurdy, 1987).

Testing techniques and extrapolation methods used in sailing yacht hydrodynamic testing have been based on methods developed by Davidson and more recent ITTC procedures mainly designed for use in ship testing. Suitability of these procedures has always been questionable due to difference of the natures of design and the flow phenomenon between sailing yachts and ships. In the 1987 ITTC conference, a session has been held to discuss the advances in yacht testing techniques.

One of the papers presented in ITTC 87 was of Murdey, et al., where experience from the 1983 Canadian America's Cup campaign is summarized with emphasis of different model towing configurations. They adopted the free sailing system –which was thought to be more suitable to 1/3 scale models that were to be used for the campaign- where the model is unconstrained and towed from the point of the top of a mast, which is located at the center of effort of the sails. They found out that both towing methods might be used to obtain cup-winning results. However, the conclusion is that the semi captive system first used by Davidson, where the model is free to heave and pitch but fixed in heel, yaw and sway gives the best results in the form which is most convenient to the designers in order to see the effects of modifications to the hull or appendages (Murdey, et al., 1987).

In the same conference, DeBord (1987) has presented his work, which summarizes the currently available state of the art for sailing yacht model testing, and addresses the possible future challenges that the testing facilities might be facing. He mainly describes methods used for preparations to the 1987 America's Cup campaign, experimental and numerical as well as addresses full scale testing efforts for validation purposes. He states that methods and techniques vary from facility to facility and there has not been a consensus achieved in crucial aspects in model sizing, testing techniques, turbulence stimulation and so on. He also describes different scaling and performance prediction techniques used amongst different facilities. He states that majority of the facilities neglect viscous effects on induced drag and drag due to heel. Viscous drag calculation methods vary tremendously. Some facilities use the Davidson's approach where 0.7 times water line length is used to calculate Reynolds number for use in Schoenherr's friction line whereas some facilities use the ITTC friction line. He also states that some facilities have been using Prohaska's methods to derive form factors. He notes that previous studies indicate that these different scaling methods may result in differences in prototype resistance as much as 8% at 5 knots speed and 4% at 8 knots of speed.

Rest of his work addresses the issue that designer's requirements in terms of accuracy may well be beyond the facilities' capabilities. The issues regarding quality assurance and repeatability of the results for different size models are addressed. His suggestions for further research were emphasizing the encouragement in the turbulence stimulation area, scaling viscous resistance, performance prediction procedures and seakeeping.

Brown and Savitsky (1987) have also published their work consisting of comparisons of towing tank tests at different scales. They have provided results from 1/10 and 1/3 scale testing that they have conducted. They also present other work conducted at the University of Southampton, MIT and DTNSRDC. Their conclusion is that results make it evident that the models smaller than 1/3 scale may be useful in design development especially when there is a budget constraint and lack of access to large facilities. They suggest that ITTC undertake a study for model scale options and set up a benchmark model prototype extrapolation practice peculiar to sailing yachts. They also point out that computational fluid dynamics methods may be useful tools in the understanding and analyzing of the results of such a study.

Claughton (1989) published towing tank test results from experiments with a 2. 4-meter model where performance of a 10-meter LWL racing yacht was being investigated. Full-scale predictions for 5 different configurations were compared with regression analysis procedures of estimating drag based on hull geometry. They utilized IMS handicapping rules and Standfast series for this purpose to assess the capabilities of mathematical models in resistance calculations compared to computational fluid dynamics methods. They have found a rather broad agreement with the experimental data owing to the differences in the geometry limitations of the regression methods and inability of the methods to take into account the bow down trim moment due to sail forces which was accounted for during the experiments.

Gerritsma, et al. (1991) shared the extended results of the Delft Systematic Yacht Hull Series experiments where upright and heeled resistance, side force and stability have been analyzed for different speeds. Their extension to the series was deemed necessary due to the new design trends of light displacement high beam vessels. They also utilized a new regression method for VPP analysis and compared performances of light, medium and heavy displacement vessels. Details of Delft's experimental setup and resistance breakdown used in regression analysis are shown and experimental results are compared to predictions.

Caponetto (1993) showed the overall picture of the design process for a cup challenger, Il Moro di Venezia by publishing the hydrodynamic design methodologies they have been using. He has briefly showed some peculiar aspects of the use of the towing tank, the wind tunnel, the CFD codes and VPP's in his paper. They have conducted the experiments in the 470-meter long INSEAN tank where three measurements could be done in a single run. His conclusion was that "towing tanks and wind tunnels are still the best tools for design off sailing yachts and CFD methods are greatly useful in the early stages of the design as well as the comprehension of special phenomena".

As the America's Cup class rule was changed to IACC in 1992, Japanese challengers were busy with systematic wind tunnel and tank tests. Nagami (1993) states that, results from these systematic experiments were used to build full-scale boats for comprehensive tests. Results of these tests were then input for simulations in order to fine-tune the resultant design. Nagami finally concludes that the towing tank model has to be approximately 3.0 meter in length to accurately predict full-scale forces and added that it was impossible to evaluate appendage design in the towing tank even with half scale models due to scale effects. They believe that actual full scale testing is necessary for appendage design evaluation.

While the Japanese were testing two prototypes at full scale, American's full scale testing program was in danger due to time and budget constraints. Todter et al. (1993) from Team Dennis Conner published their design process for the 1992 Stars & Stripes campaign. Where they explained the methodologies used in towing tank testing, wind tunnel, Splash Wave Drag Code, Non free surface CFD codes, VPPs and RMPs (Race Model Program), Sea keeping codes, structural codes, scaled sailing and full size data analysis. They have found the overall design management successful in the sense that only one boat was build that could be tested and the utmost potential was extracted from the original design.

The Partnership for America's Cup Technology (PACT) had a testing program aiming at defending the cup in 1992 races. Teeters (1993) explains the focuses of the program as "establishing baseline data, in both calm water and waves, for the American defense syndicates; addressing the tankery issues of test reliability and accuracy and expansion to full scale; developing the test program so that the tank serves as a more capable partner with CFD and improving the techniques of processing test data". His paper addresses solutions to peculiar problems associated with yacht testing. Comparisons are included in the paper between conventional viscous drag predictions and methods developed by PACT that incorporate dynamic wetted area and wetted length calculation as well as enhancements in appendage drag estimations and multiple canoe body form factors. Teeters (1993) concludes that the enhanced methods they utilized may in fact eliminate some of the errors associated with human input and shows better robustness. Also, achievements in viscous drag estimations are addressed resulting in better residuary resistance stripping that has been validated with analytical and computational methods.

In the same year, published work of Parsons & Pallard (1997) shows that The National Research Council of Canada's Institute for Marine Dynamics was in the effort of designing and building a new dynamometer that would be capable of testing 1/3 scale models of AC yachts in waves and 1/2 scale models in calm water. Their paper shows the design process of the yacht dynamometer, error analysis, calibration and repeatability of the results. They claim that this new design increased the quality of the results obtained in the towing tanks and the results are now closer in quality to the ones obtained in wind tunnels. They also claim that evaluations of lifting surfaces in the presence of the free surface is now possible as well as experiments on devices like strakes to reduce interference drag due to viscous effects (Parsons & Pallard, 1997).

Coming to the year 2000, it was evident that there was lack of published full-scale hydrodynamic data and towing tank data comparison. Such comprehensive work has numerous times been undertaken for America's Cup campaigns and due to confidentiality reasons; these have not been extensively published. In 1999, Hochkirch and Brandt (1999) et al have published results of a study in the field where they have done measurements on a 33 feet sailing yacht equipped with a full scale sail force dynamometer. The paper mainly explains the design and construction of the dynamometer system, calibration and data acquisition procedures. Authors' aim was to build a database of full-scale data along with the tank test results that would lead to a better understanding of the performance prediction methods and hydrodynamics flow aspects and the keel hull interactions.

There is restricted amount of work published in the field of scale effects on foils. van Walree and Luth (2000) have published the results of their experiments and calculations for investigation of the scale effects in hydrofoil experiments. Their work was aiming to investigate the scale effects on foils used for stabilizing vessels and for foils used for lifting hulls in hydrofoils in unsteady flow conditions. They state that models scales are smaller in sea keeping experiments compared to ship testing. The Reynolds numbers associated with these tests are close to values experienced in sailing yacht testing and hence results of these scale effect experiments may be useful in understanding the scale effects experienced by yacht appendages and setting the methodology of possible experiments for investigating the scale effects on yacht appendages.

van Walree and Luth (2000) have tested a Naca 16 section foil with 8% thickness submerged at four chord lengths. They have measured lift and drag forces in steady flow and in waves. For comparison, they have calculated the full-scale forces by Vortex Lattice method. They have concluded that in steady flow, turbulence stimulation is necessary to compensate for laminar flow effects. Also, at low Reynolds numbers, where it is impractical to stimulate turbulence, scale effects may be so large that the lift force may be impossible to extrapolate. If this is the case, reliable results may only be obtained at small angles of attack and the range should be determined prior to experiments via calculations. The results of the unsteady experiments showed that the scale effect on lift in unsteady flow conditions is much smaller than for steady flow conditions for typical hydrofoil type appendages.

Gomez (2000) has summarized ITTC's efforts of investigating scale effects on form factors in his study where CFD codes were utilized for axisymmetric body analysis for comparison with experiments on four geosim families that were already conducted. He states that normal assumption in ship tank testing to use same form factor for model and prototype. This is questionable since the form factor is part of the viscous resistance, which depends on the Reynolds number differences between the model and the prototype. He has found out that the form factor obtained by Prohaska's method increases with model size in all the case analyzed.

His studies are part of the program initiated by ITTC in 1993 where four geosim model families' resistance tests have been reanalyzed. Form factors of each geosim families' models at four different scales have been obtained. Regression lines have been drawn by the aid of the least squares method with correlation coefficients ranging between 0.927 and 0.984. These analyses showed that there is considerable variation of form factor with increasing scale. He notes though that the form factor dependency is associated with the friction line utilized for calculations and it should be kept in mind that in case of different friction line usage, trends in the variation of form factor will differentiate from the current study.

Graf and Böhm (2005) have presented their work in the Yacht Research Unit of University of Applied Science Kiel regarding the AVPP, which they name as a towing tank postprocessor. They use this new methodology to model the hydrodynamic forces on the hull and appendages and these forces can be derived from a limited set of towing tank data. Examples of the application of the new methodology are shown on the effects of changes of rules V4/V5 on the design of AC yachts and comparison between a conventional and a canting keel yacht.

By surveying the developments in tank testing methods of sailing yachts, especially in the last 40 years, certain developments resulted in increased accuracy of results. Matrix testing, turbulence stimulation, CNC machined models, application of blockage corrections, 3-dimensional extrapolation in viscous scaling are amongst the major developments. Also, due to trends in America's Cup tank testing in the last 40 years, model scales have increased and normal practice became testing models in 1/3 size in America's Cup campaigns. This resulted in increased budgets and testing of numerous design candidates brought the subject to a point where alternatives are being sought. Design teams are in the study of trying to integrate CFD methods into their design loop; but these methods have not gained sufficient confidence as yet amongst the sailing yacht design community in order to replace experiments. Instead, these methods are being used as supplementary tools to experiments, for validation, flow mapping, pressure measurements, boundary layer surveys, wake surveys and wave measurements. Also, CFD methods have the potential for use in better understanding of scale effects.

Considering that the high budgeted projects are seeking for alternatives, relatively lower budgeted projects are bound to testing with smaller models or even neglecting tank tests. Considering that the developments in sailing yacht tank testing and a reasonable accuracy level was achieved for big models; understanding the phenomenon behind small model testing remains as an important research area currently since it would reduce costs of tank testing programs and would enable numerous candidate testing for some campaigns.

Even though the aforementioned developments increased accuracy of the experiments, main unresolved issues are still the same; scale effects are not very well understood, experimental accuracy and repeatability and quality assurance remains areas of further improvements. DeBord et al. (2004) have recommended that if Kirkman's scale effect experiments on the Yacht Antiope may be repeated for two different hulls with modern appendages, this would lead to a better understanding of the scale effects associated with tank tests of sailing yachts.

2.2 Current State of the Art

2.2.1 ITTC procedures for ship testing

The scaling procedure of Froude has been modified and standardized by ITTC 1957 and ITTC 1978 methods. Although the division of resistance into viscous resistance governed and scaled by Reynolds Number and wave resistance governed and scaled by Froude Number is a hypothetical approach, it is widely used for practical purposes. ITTC 1978 method suggests the total resistance of a ship model can be separated into viscous and wave components and interaction between them can be neglected. This assumption is satisfactory for most ship forms without extensive spray or wave breaking.

The viscous component of resistance can also be expressed as summation of frictional resistance and form drag, modelled as a fraction of frictional drag, both are assumed to be independent from Froude number.

$$\begin{aligned} R_T &= R_V(Re) + R_W(Fn) \\ &= (1+k)R_F + R_W \end{aligned} \quad \textbf{(2.1)}$$

The frictional resistance (R_F) in coefficient form (C_F) is derived from resistance coefficient of a flat plate with the same length and wetted surface area of the hull such as the ITTC-1957 correlation line. The formulation is:

$$C_F = \frac{0.075}{(log(Re) - 2)^2} \quad \textbf{(2.2)}$$

(1+k) is called the form factor and has been utilized to consider the three dimensional effects on the viscous resistance due to hull form. It is normally assumed to be independent of the speed and ITTC procedures recommend Prohaska's method for form factor determination. In this method, the wave resistance portion of the total resistance between $0.1<Fr<0.2$ is assumed to be a function of Fr^4. This results in a straight-line plot of C_{TM}/C_{FM} vs. Fr^4/C_{FM} intersecting the y-axis at 1+k, which in turn is the form factor of the hull (ITTC, 2002).

While extrapolating to different scales, the frictional resistance is recalculated according to the correlation line. The form factor and the wave making resistance coefficient are assumed to be independent of model scale.

2.2.2 Testing techniques for sailing yachts

Although the extrapolation technique for ship testing is relatively straightforward and generically applicable, scaling of sailing yacht model test results possesses two problems:

First of all, the appendages of sailing yachts, namely keel-bulb combinations and rudder, are in considerable size of wetted area and effective Reynolds numbers for these appendages are very different comparing to the main hull. Hence, a scaling procedure has to be adopted to scale the resistance of hull, keel-bulb and rudder separately for the upright models.

$$\begin{aligned} R_T &= R_V\left(Re_{H,}Re_{K,}Re_B, Re_R\right) + R_W(Fn) \\ &= (1+k_H)R_{F-H} + (1+k_K)R_{F-K} + (1+k_B)R_{F-B} \\ &+(1+k_R)R_{F-R} + R_W \end{aligned} \tag{2.3}$$

Secondly, the effect of leeway and heel angle on resistance may result in a highly separated flow mainly at higher speeds and higher leeway angles. The resistance components of heel resistance and induced drag must be investigated for the scaling purposes.

$$\begin{aligned} R_T &= R_V\left(Re_{H,}Re_{K,}Re_B, Re_R\right) + R_W(Fn) + R_H(Fn) + R_I(Re, Fn) \\ &= (1+k_H)R_{F-H} + (1+k_K)R_{F-K} + (1+k_B)R_{F-B} \\ &+(1+k_R)R_{F-R} + R_W + R_H + R_I \end{aligned} \tag{2.4}$$

. In recent methodology of sailing yacht tank testing, total resistance (R_T) is measured, form factor of the hull (k_H) is estimated from low speed tests, form factors for keel (k_K), bulb (k_B) and rudder (k_R) are estimated from empirical formulae. Heel resistance (R_H) and induced resistance (R_I) can be determined from total resistance tests in heel and leeway. Wave resistance cannot be obtained independently, it is normally obtained from the upright total resistance tests.

2.3 Deficiencies of the Current State of the Art

Several advancements to the ITTC extrapolation method developed for ships have been proposed over the years, mainly owing to the research efforts for America's Cup campaigns. Although modifications to the ITTC method developed for ships, like multiple canoe body form factors and utilization of dynamic wetted area may eliminate some of the errors and lead to better robustness (Teeters, 1993), there are still areas for further investigations and a requirement for a generically applicable extrapolation method.

Naturally, as majority of these advancements were results of America's Cup campaigns without budgetary issues, they evidently led to dramatic increases in model sizes and testing costs. This problem was mainly due to the inability to assess the influence of different appendage configurations on the performance of the full scale yachts with small models (Campbell & Claughton, 1987). Also, hull forms and most importantly appendage configurations differ from the past when majority of the scale effect investigations have taken place. DeBord et al. (2004) recommended that given these advancements and hence the increased cost of the tests, renewed research on scale effects with modern hull forms equipped with highly loaded appendage configurations would be beneficial in terms of making best use of tank test data and assessment of cost versus scale trade off issues.

Below, the deficiencies of the currently available methods that lay the base of this investigation are summarized:

- There is, as yet, no consensus within facilities in terms of testing techniques and extrapolation methods.
- Uncertainties associated with the experimental lift and resistance data in the presence of heel and leeway are not clearly known.
- Only 1/3 scale experiments give satisfactory results for use in grand prix racing campaigns as stated by experimenters. Some find even this scale unsatisfactory for appendage development.
- Extensive budgets and time is required for 1/3 scale model testing which leads to search for alternative methods.
- Extrapolation methods are adopted from ship approaches and they are still questionable in terms of suitability for sailing yachts.
- Appendage development is not possible with small scale testing.
- CFD methods have not gained sufficient confidence within the sailing yacht design community for being able to eliminate the need of experiments.

3. HYDRODYNAMIC TESTING

The intended testing program consisted of experiments at Ata Nutku Ship Model Testing Laboratory on four different scale versions of a TP52 class yacht. The towing tank is 160 metres long with a width of 6 metres and a water depth of 3.4 metres (nutkulab.itu.edu.tr)). The yacht was designed by Rob Humphreys Design according to the 2010 rules of the TP52 class association. This class gained popularity in the recent years around the globe and the yachts of this class, built with modern technology and materials, are good performers in the overall range of wind speed and heading combinations.

Table 3.1 : Main Parameters of the tested models and the full-scale prototype.

Scale	1/10	1/8	1/6	1/4	Full Scale
Length (m)	1.58	1.975	2.633	3.95	15.8
Beam (m)	0.442	0.553	0.737	1.105	4.42
Draught (m)	0.038	0.048	0.064	0.096	0.384
WS (m^2)	0.363	0.567	1.008	2.267	36.273
Displacement (kg)	8.66	16.91	40.09	135.31	8660
C_B	0.44	0.44	0.44	0.44	0.44
C_M	0.807	0.807	0.807	0.807	0.807
WS Total (m^2)	0.452	0.706	1.255	2.823	45.168

Figure 3.1 : Four different scales models tested.

3.1 Experimental Procedure

3.1.1 Manufacture of models

One of the main challenges in yacht testing is the fact that the models need to be of lightweight construction, owing to the low displacements in model scale and the need for ballasting. This enforces the utilization of composite model construction that is time consuming and expensive. As scale is increased (model size is decreased), the displacement values may even become unachievable even with composite materials. As an example, the 1/10 scale model has a displacement of 8.66 kilograms including the model weight, appendages, ballast and dynamometer connection. Even though it is not a preferred condition, counter weight arrangements may be required if small models are utilized.

The CAD version of the model hull lines was received from the designer in IGES file format. The model manufacturing facility in the Ata Nutku Ship Model Testing Laboratory was used for the model construction process. Male moulds of the hulls were CNC cut at the laboratory from wood. These were then used to manufacture the polyester female moulds. Then the polyester hulls were laid on the female moulds with the aid of gel coat material. The hulls were stiffened with aluminum honeycomb in way of center keelson and two transverse beams. The result was smooth surfaced very lightweight polyester hulls. The keel and rudder moulds for the hulls were manufactured from composite materials, which were used for aluminum casting. The casted appendages were then faired in the laboratory and mounted to the hulls with the aid of studs. All appendages were fixed permanently to the hulls.

The accuracy of the manufactured models was assessed by measurements of 5 random points as specified in Appendix A. The maximum average dimensional error was seen to be encountered at 1/10 scale with a value of 1.05% in comparison to the full-scale dimensions.

3.1.2 Experimental setup

3.1.2.1 Measurement of forces and motions

The 6 component dynamometer of Kempf & Remmers was used for resistance and side force measurements. The dynamometer was connected to the models via a single post which enabled the model to heave and pitch but restricted the model in all other degrees of freedom. Heave was measured with the aid of a linear position transducer, LPT. Pitch was measured with the angle potentiometer connected to the towing post-model connection flange. All equipment were rigged to the National Instruments Digital-Analogue Converter and then to the computer on the carriage. National Instrument’s LabView software was used for data acquisition of drag, sideforce, moments, pitch and heave. A software designated for this setup was developed within LabView for data gathering.

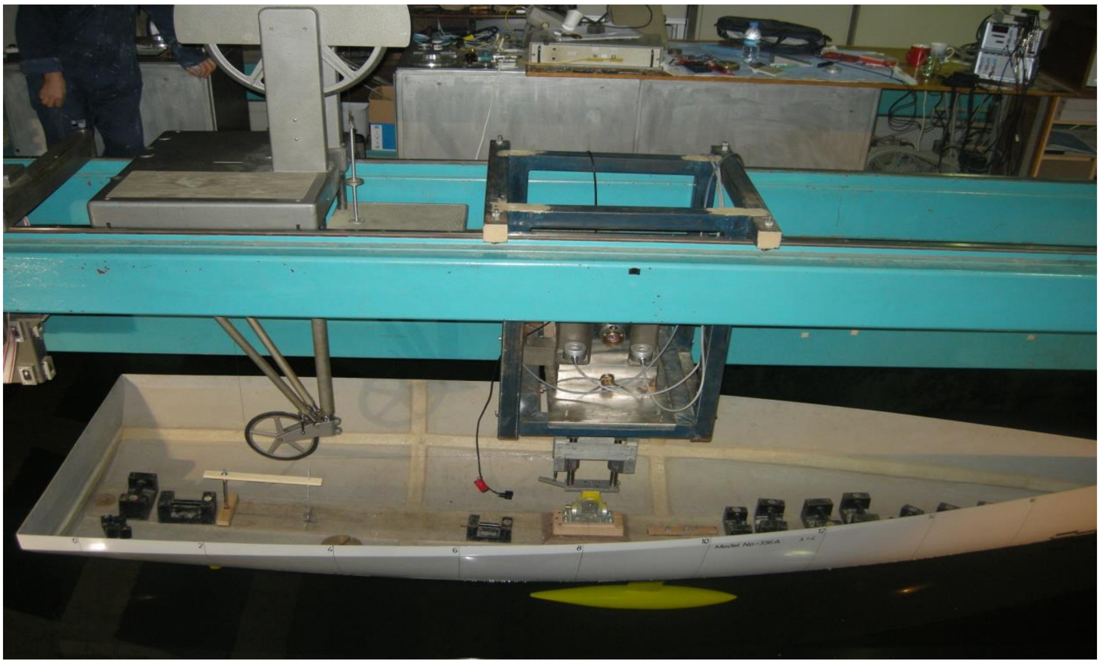

Figure 3.2 : Quarter scale model, connected to the six component dynamometer.

3.1.2.2 Measurement of the wave pattern

The wave pattern measurement system consisted of three wave probes located on the 100th meter of the tank on each side, which were placed at 30%, 33% and 40% tank widths from centerline respectively. These were connected to wave probe monitors, and via National Instruments Digital Analogue Converter, to a PC with Lab View software.

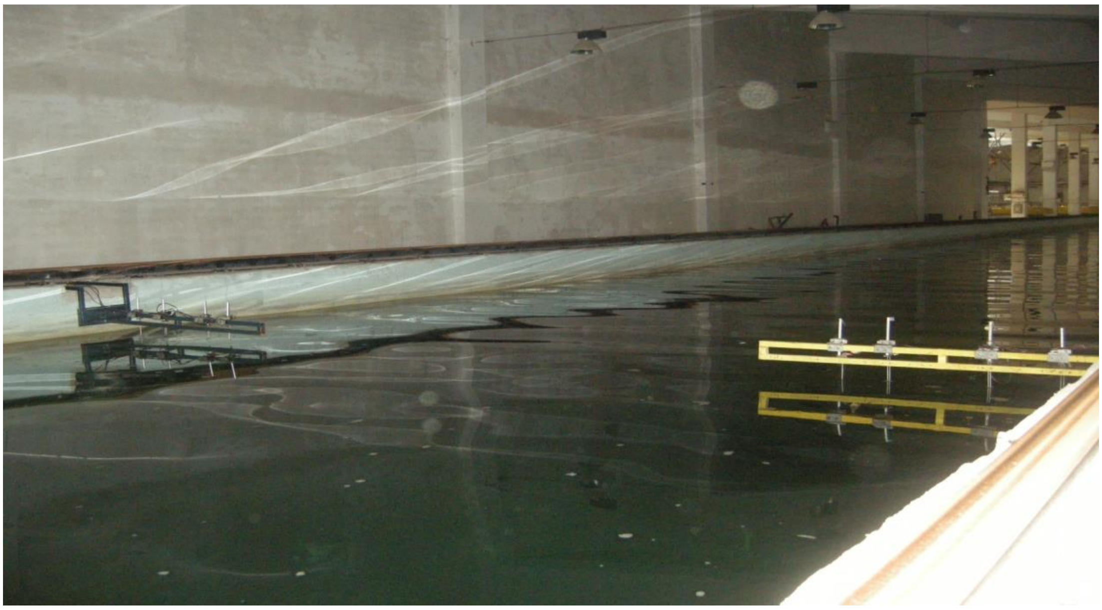

Figure 3.3 : The wave probes.

3.1.3 Calibration

Wave probes were calibrated prior to the experiments in an isolated water tank one by one. This was not sufficient due to differences in water composition, orientation of the probes in the towing tank and the calibration tank, and the sensitivity of the probes to variations in grounding. Hence, as they were installed to the tank, they were calibrated, in-situ prior to the experiments. It has been detected that the probes may deviate from the calibration with time; the slopes of the calibration lines may vary. Hence, the calibration of the wave probes was checked before a new set of experiments.

Calibration of the dynamometer was carried out prior to a set of experiments; checked before each experiment. The dynamometer was found to be robust and reliable, with repeatability of forces within a maximum of 30 grams difference.

The heave and pitch potentiometers were also calibrated prior to the experiments; for pitch potentiometer, calibrated aluminum wedges were used whereas for the heave potentiometer, a caliper was used.

During the experiments, it has been seen that the repeatability of the trim measurements was at the order 0.1 degrees. The repeatability of the sinkage measurements was at the order of 1 millimeter.

3.1.4 Scope of experiments

For all scales, the experimental investigation aims at achieving a better understanding of variations the following parameters with scale:

- Total drag,
- Viscous drag,
- Wave making drag,
- Generated wave profile,
- Side force,
- Sinkage,
- Trim.

To achieve this understanding, wave profiles, side force and drag values, sinkage and trim values were measured in upright and heeled & yawed configurations for all scales.

3.1.5 Definition of test matrix

The TP 52 class yacht was tested in 4 different scales (1/4, 1/6, 1/8 and 1/10). The test matrix for each model consists of following runs (speeds as corresponding to full scale):

- Upright runs without appendages (5-20 knots),
- Upright runs with appendages (5-20 knots),
- Heeled runs in 10 (5-13 knots), 15 (8-16 knots) and 25 degrees of heel (8-16 knots) with leeway angles of 0, 2, 4 and 6 degrees in starboard tack.

Table 3.2 : Heeled test matrix as supplied by the designer.

Heel Angle (°)	-10				-15				-25			
Yaw Angle (°)	*-6*	*-4*	*-2*	*0*	*-6*	*-4*	*-2*	*0*	*-6*	*-4*	*-2*	*0*
Upwind Speeds in knots (full scale)	5	5	5	5	8	8	8	8	8	8	8	8
	7	7	7	7	9	9	9	9	9	9	9	9
	9	9	9	9	10	10	10	10	10	10	10	10
Downwind Speeds in knots (full scale)	-	-	11	11	-	-	13	13	-	-	13	13
	-	-	13	13	-	-	16	16	-	-	16	16

For heeled & yawed cases, the models were only tested on starboard tack due to time constraints and geometric limitations of the carriage. It was not possible to heel the model in port tack due to high beam of the model at 1/4 and 1/6 scales. Since the model was manufactured with CNC and initial runs showed negligible side force during upright runs, sufficient confidence was gained about the symmetry of the model and hence it was decided to test on a single tack.

3.1.6 Experimental methodology

To achieve similarity between prototypes and small scale models during the experiment, certain non-dimensional parameters must be kept constant. For viscous forces associated with the fluid flow, Reynolds number (Re) must be similar between the model and the prototype. Same applies for Froude number (Fr) in order to keep gravitational forces similar. In towing tank testing, it is impossible to keep both parameters constant for the model and the prototype.

$$Re = \left(\frac{VL}{\gamma}\right) \tag{3.1}$$

$$Fn = \left(\frac{V}{\sqrt{gL}}\right) \tag{3.2}$$

In conventional tank testing methodology, Froude similarity is achieved by testing the models at slower speeds which would in turn lead to same Froude numbers for the model and the prototype.

However, this procedure makes achieving Reynolds Number similarity impossible. Hence, the model may be operating in laminar or transitional flow where as –in majority of the cases- the prototype would be operating in fully turbulent flow. Also, compliance with the skin friction formulae suggests that flow around the hull needs to be fully turbulent. Owing to the the low Reynolds numbers of the hulls and appendages –especially at small scales-, turbulence studs were utilized for the compliance with the skin friction formulae and to achieve Reynolds similarity.

Turbulence stimulation was achieved by 3 mm diameter, 5 mm long studs spaced 25 mm apart and located at 1/20 model length behind the stem profile of the models, at quarter chord behind the leading edge of the fin keels, at quarter chord behind the leading edge of the rudders and at quarter length behind of the bulb tip point. Model resistance test results were corrected for the resistance of the studs.

As stated in previous sections, the semi-captive testing approach has been utilized where the yacht model is fixed to the dynamometer through a single post at the LCF, free to heave and trim. Ballast weights were used to adjust the static trim of the models and to achieve the required displacement in upright and heeled conditions.

While utilizing this approach, care must be taken in heeled condition testing. As the yacht is towed from the center of floatation rather than the center of effort of the sails, the bow down trim which would be caused by the presence of the sails will not be considered with this approach. This needs to be corrected for by placing weights in the bow depending on the sailing atitude (upwind versus downwind), scale and speed of the test in consideration. In this approach, the moment casued by the resistance of the hull and the appendages are calculated for each speed, scale and sailing condition and respective weights are added to the model. It has been ensured that all the models tend to trim equally as the full scale prototype. Details of the calculations may be found in Appendix B.

3.2 Interpretation of Raw Tank Data

3.2.1 Corrections

As suggested by ITTC Recommended Procedures, the dimensions of the towing tank should be large enough to avoid wall and blockage effects. As this was the case for 1/10, 1/8 and 1/6 scale models, no corrections were necessary. For the 1/4 scale model, Tamura's method was utilized. (ITTC, 2002).

$$\frac{\Delta V}{V} = 0.67m \left[\frac{L}{B}\right]^{\frac{3}{4}} \frac{1}{(1 - Fr^2)} \tag{3.3}$$

$$m = \frac{A_x}{A} \tag{3.4}$$

In this equation, A_x denotes maximum sectional area of the model and A is the sectional area of the towing tank in square meters. The maximum correction applied was at the order of 3% for the 1/4 scale model.

3.2.2 Force breakdown

The measured resistance R_{TM} is expressed in non-dimensional form:

$$C_{TM} = \frac{R_{TM}}{\frac{1}{2}\rho_M S_M V_M^2} \tag{3.5}$$

The density is calculated as per ITTC Recommended Procedure 7.5-02-01-03 according to the measured temperature of the water in the towing tank. Figure 3.5 shows the variation of density with temperature. Static wetted area was utilized during force breakdown. Averaged velocity throughout the data acquisition was corrected for blockage.

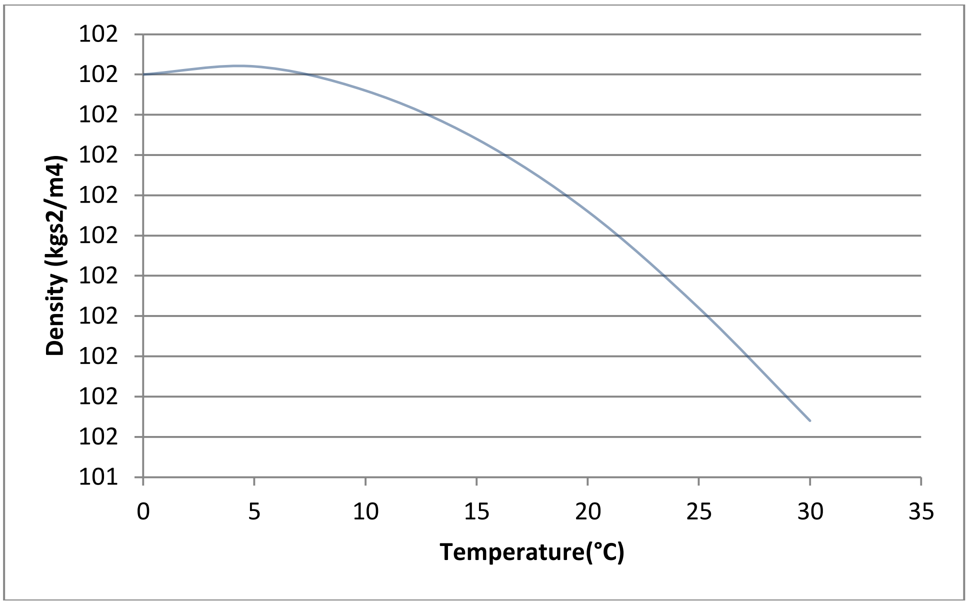

Figure 3.4 : Variation of density with temperature.

The upright resistance breakdown is as stated in chapter 2.2.2. The resistance in the naked hull condition is stripped as:

$$R_T = R_V(Re) + R_W(Fn) \tag{3.6}$$
$$= (1+k)R_F + R_W$$

The frictional resistance is calculated from emprical formulae, assuming fully turbulent flow over the whole model. As this usually is not the case, especially for small models, turbulence stimulation devices are utilized, forcing transition from laminar to turbulent flow. As transition is forced, the viscous flow condition similarity is assumed to be achieved between the model and the prototype.

The drag of the stimulators were deduced from the total resistance. A drag coefficient of Cd=1.00 was utilized for the stimulators (Hoerner, 1965).

One of the major differences from the mentioned ITTC force breakdown methodology in the utilized force breakdown is the friction line used for frictional drag calculations. ITTC procedures recommend the use of ITTC Ship- Model Correlation line, whereas in the computations, Grigson's Friction line has been used. This line has better characteristics in 3-D extrapolation and is less prone to scale effects when form factors are utilized (ittc.sname.org, 2007). The comparison of Grigson's Friction Line and ITTC Ship Model Correlation line may be found in

Figure 3.6. It is worth noting that the ITTC line has been designed for 2-D extrapolation. The Grigson formulation is expressed as :

$$C_F = [0.93 + 0.1377(logRn - 6.3)^2 - 0.06334(logRn - 6.3)^4] \times \left(0.075/(logRn - 2)\right)^2 \quad \textbf{(3.7)}$$

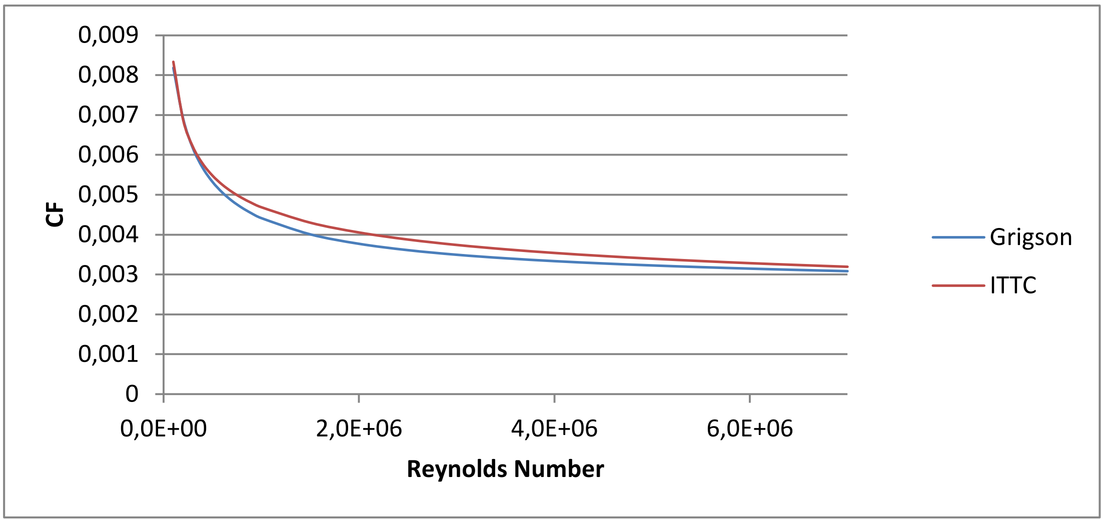

Figure 3.5 : Variation of Cf with Reynolds Number.

The viscous resistance component is expressed as a percentage increase in frictional resistance due to 3-D form effects.

$$C_{V}=(1 + k)C_F \quad \textbf{(3.8)}$$

In this formulation, the form factor is represented by the coefficient k. ITTC procedures recommend Prohaska's method for the evaluation of the form factor. According to this method, at low speeds (0.1<Fr<0.2), the total resistance is assumed to be a function of Fr^4 and the C_{TM}/C_{FM} versus Fr^4/C_{FM} plot would intersect the ordinate (zero speed) at (1+k) which would yield the form factor (ITTC, 2002).

As far as extrapolation is concerned, the form factor is assumed to be independent of scale, speed, Reynolds number, Froude number, trim and sinkage. Prohaska's methodology, which works well for ship models, posesses difficulties for yacht testing due to several reasons. As the models are smaller compared to ships, testing at 0.1<Fr<0.2 speed range causes accuracy and repeatability issues. At this speed range, the scatter in the data would prevent accurate form factor determination, especially for the case of 1/6, 1/8 and 1/10 scale models. Another important difference between ships and sailing yachts that would cause skepticism regarding the validity of the form factor assumption is concerned with the differences in the running attitude. In case of ships, the trim variation across the tested speed range is not substantial and the underwater form of the ship does not vary substantially. However, a sailing yacht may trim up to 4 degrees across the tested speed range and the underwater form changes substantially. As this would only cause extrapolation problems and lead to misleading results in the full scale performance estimation but would not effect the comparison between different scale models, the investigation of the variation in the form factor with trim has been left out of scope of the test program. Due to the mentioned difficulties in obtaining the form factor from low speed tests, the hull form factor has been obtained by combining low speed test results with CFD calculations and designer's recommendations. A fixed form factor of 1+k=1.09 is utilized for the hull at all scales.

As the total upright resistance of a sailing yacht in naked condition is expressed as the summation of viscous and wave resistance components; when the calculated viscous resistance and the resistance of the studs are deduced from the total measured resistance, the wave resistance may be obtained. The wave resistance, when non-dimensionalized to coefficient form by the corrected speed, density at measurement temperature and static wetted area, is assumed to be identical for geometrically similar models at different scales that are operating at the same Froude numbers.

In the upright appended condition, the total resistance may be stripped to its components as:

$$\begin{aligned} R_T &= R_V\left(Re_{H,}Re_{K,}Re_B, Re_R\right) + R_W(Fn) \\ &= (1+k_H)R_{F-H} + (1+k_K)R_{F-K} + (1+k_B)R_{F-B} \\ &\quad +(1+k_R)R_{F-R} + R_W \end{aligned} \tag{3.9}$$

The resistance of the appendages in turbulent flow conditions may be estimated empirically as the summation of frictional resistance and form resistance by taking into account the augmented velocity over the foil and of the effects of pressure resistance. This methodology, suggested by Hoerner, is valid for foils having their maximum thickness at approximately 30% of the chord from the leading edge

(Hoerner, 1965). The formulation accounts for the pressure form resistance as a fraction of the friction resistance and is expressed as:

$$C_V = C_F \times \left[1 + 2\left({}^{t}/_{c}\right) + 60\left({}^{t}/_{c}\right)^4\right] \quad \textbf{(3.10)}$$

In the formulation, the ratio of thickness of the foil to the chord length is approximated to the resistance increase due to the form of the foil. Therefore, this formulation could be valid as the form factor for the keel and the rudder.

As the bulb may not be treated as a foil, its form resistance needs to be calculated by other means. The methodology proposed by Debord and Teeters (1990) has been followed with the formula given as:

$$C_V = C_F \times \left[1 + 1.5 \times (\frac{Bdia}{Blen})^{1.5} + 7 \times (\frac{Bdia}{Blen})^3\right] \quad \textbf{(3.11)}$$

In this formulation, Bdia stands for the diameter of the bulb and Blen is the bulb length.The utilized form factors for the keel, bulb and the rudder were 1.33, 1.13 and 1.33 respectively. As these formulations are valid for fully turbulent flows, any deficiency in the forced laminar to turbulent transition might lead to difficulties in compliance with the formulations. Attached laminar flow might lead to less resistance but in case of laminar separation, the measured resistance might be higher than that of a fully turbulent flow. Van den Bosch and Pinskter have reported that lifting surfaces operating at Reynolds numbers less than about 100000 to 150000, when artifically tripped to turbulent flow may not always produce the desired results (Oosanen, 1985). Also, any interaction component between the hull and the appendages can not be considered within this formulation.

In the above mentioned upright appended resistance breakdown methodology, the viscous resistance of each individual component (hull, keel, bulb and rudder) is empirically calculated and deduced from the total measured resistance to identify the wave resistance component. This methodology is known to work well for ship types in which the interaction between the viscous and wave resistance components are negligibly small, the appendages are smaller in size compared to sailing yachts and the interactions between the hull and appendages are also negligable. As this is not the case for a sailing yacht with appendages comprising a subtantial portion of the wetted area, a transom stern running dry at almost half the test speed range; there are some additional resistance components that may not readily be calculated with empirical formulations. These components are due to transom stern effects, interactions between the hull and the appendages, variation of appendage resistance with respect to trim and interaction between the appendages individually. Therefore, the computed wave resistance component in the appended condition could include non-wave related resistance components. As majority of these components are of viscous nature, assumption of identicality of the calculated wave resistance coefficient at same Froude numbers needs careful consideration in the appended condition.

The total resistance in the heeled condition is composed of three main components as stated in 2.2.2. Recalling the same equation:

$$\begin{aligned} R_T &= R_V\left(Re_{H,}Re_{K,}Re_B, Re_R\right) + R_W(Fn) + R_H(Fn) + R_I(Re, Fn) \\ &= (1 + k_H)R_{F-H} + (1 + k_K)R_{F-K} + (1 + k_B)R_{F-B} \\ &+ (1 + k_R)R_{F-R} + R_W + R_H + R_I \end{aligned} \quad \textbf{(3.12)}$$

The heel drag (R_H) is obtained from tests with heel but no leeway. The total measured resistance in this condition is deduced from the upright resistance at same speed to obtain the heel resistance.

Induced drag (R_I) is then obtained by repeating the heeled tests under a range of leeway angles as specified in the test matrix. Induced drag is also a beneficial parameter for sailing yacht performance estimation as it is related to the generation of side force. It is used to express the drag penalty associated with side force generation. The induced drag, in coefficient form C_{DI} is expressed as:

$$C_{DI} = \frac{{C_L}^2}{\pi(ARe)} \quad \textbf{(3.13)}$$

$$ARe = \frac{(2Te)^2}{2A} \quad \textbf{(3.14)}$$

The effective aspect ratio (ARe) differs from the geometric aspect ratio (span/chord) as it is proportional to the ratio of the square of the effective draft and planform area. The effective draft (Te) is the span of the total lifting surface system (hull, keel, bulb) taking into account the effects of the mirror image of the lifting surface due to the symmetry condition at the free surface. Therefore, a lifting surface gets more efficient, i.e. less induced drag penalty is paid for lift generation as the effective draft increases over the actual draft. Effective aspect ratio and hence the effective draft are cruical parameters as they reflect the efficiency of the whole yacht (hull, keel, bulb, rudder and the mirror image) in acting as a single lifting surface. For simplicity purposes, the induced drag expression will be used as in equation 3.15 further in the analyses.

$$C_{DI} = \propto C_L^{\,2} \tag{3.15}$$

As there is a linear relationship between the two parameters, the slope of the C_L^2 versus C_D curve (α) will be utilized as the efficiency factor, as shown in equation 3.15.

3.2.3 Wave pattern resistance calculations

Several attempts were made in the past for the investigation of wave pattern resistance of sailing yachts. Influence of appendage configuration and hull form variations on the performance of the yacht may be assessed with the longitudinal wave cut technique.

The wave pattern of a model which is asymmetrical according to the centre plane of a towing tank can be mathematically expressed as (Insel, 1990);

$$\zeta = \sum_{m=0}^{M} \left[\begin{matrix} \xi_m \\ \alpha_m \end{matrix} \cos(\omega_m x) + \begin{matrix} \eta_m \\ \beta_m \end{matrix} \sin(\omega_m x) \right] \begin{matrix} \cos \\ \sin \end{matrix} \left(\frac{m\pi y}{W} \right) \tag{3.16}$$

Where upper terms apply for even m, lower terms apply for odd m, and

$$\omega_m = K_m \, cos(\theta_m) \tag{3.17}$$

$$K_0 = \frac{g}{V^2} \tag{3.18}$$

Where m is the harmonic number, the wave number (K_m) and wave angle (θ_m)can be obtained from the solution of:

$$K_m - \frac{g}{V^2}\sec^2(\theta_m)\tanh(K_m H) = 0 \tag{3.19}$$

Where *W* is the width of the tank, H is the water depth of the tank. The wave pattern resistance can be written from the law of the conservation of momentum as follows:

$$R_{WP} = \frac{\rho g W}{2} * \left\{ \begin{matrix} \left(\xi_0^2 + \eta_0^2\right) \left(1 - \dfrac{2K_0 H}{Sinh(2K_0 H)}\right) \\ + \displaystyle\sum_{m=1}^{M} \left(\begin{matrix} \xi_m^2 \\ \alpha_m^2 \end{matrix} + \begin{matrix} \eta_m^2 \\ \beta_m^2 \end{matrix} \right) \left[1 - \dfrac{\cos^2(\theta_m)}{2} \left(1 + \dfrac{2K_m H}{Sinh(2K_m H)}\right) \right] \end{matrix} \right\} \tag{3.20}$$

Where ξ_m, η_m applies for even m and α_m, β_m applies for odd m.

Wave pattern experiments with all four models were conducted both in naked hull and appended hull configurations with the keel and rudder. The waves generated by the models were acquired through three wave probes at both sides and a computer aided data acquisition system in order to make comparisons of wave pattern resistance for the scale effects.

The wave pattern analysis was conducted with multiple longitudinal cut method with matrix solver (Insel, 1990). The wave pattern traces were separated into symmetrical and asymmetrical components by using the waves probes on both sides.

$$\zeta_{SYM} = \frac{\zeta_+ + \zeta_-}{2} \tag{3.21}$$

and

$$\zeta_{ASYM} = \zeta_+ - \zeta_- \tag{3.22}$$

There is clear difference between wave resistance defined by ITTC 1978 method and wave pattern resistance. Wave pattern resistance is obtained from energy spent on the created waves. Meanwhile wave resistance is obtained from substraction of frictional resistance and form resistance, hence it includes not only the energy spent on wave generating but also all the other physical causes such as separation, wave breaking, vortex forming etc.

3.3 Upright Testing

3.3.1 Upright resistance testing for the naked hull

Upright testing for the naked hull has been successfully completed for all scales as per proposed test matrix. As these were the initial set of experiments to be conducted, reliability of the experimental setup was of a bigger concern compared to latter experiments. The dynamometer was observed to be reliable, owing to minor differences in the results of repeated experiments. Regardless of the scale of the model and speed in consideration, the repeatability was observed to be at the order of 30 grams of resistance force.

The models were tested for speeds corresponding from 5 knots to 20 knots in full scale. Experimental total resistance plotted against Reynolds number in the naked condition is given in Figure 3.6. It may be seen from the mentioned plot that the Reynolds number range lies in the turbulent region even for the smallest model; theoretically any effect of laminar flow is not expected. Total resistance coefficient and wave resistance coefficient results across the Froude number range are given in Figures 3.7 and 3.8 respectively.

Total resistance measurements at all scales show similar resistance curves across the tested speed range. However the total resistance and wave resistance magnitude is higher for 1/10 scale model. Small size models are prone to errors originating both from measurement uncertainties and laminar flow regime. Other three scale models, namely 1/8, 1/6 and 1/4 scales are in very good agreement across the Froude number range.

The methodology of obtaining the wave resistance component is based on application of a form factor of 0.09 to the frictional resistance to calculate the viscous resistance. The viscous resistance component and the drag of the turbulence stimulators are then deduced from the total resistance to obtain the wave resistance component. According to the obtained data, scale effects on the naked hull are negligible with Froude scaling using ITTC 1978 method satisfactorily except for the smallest 1/10 scale model.

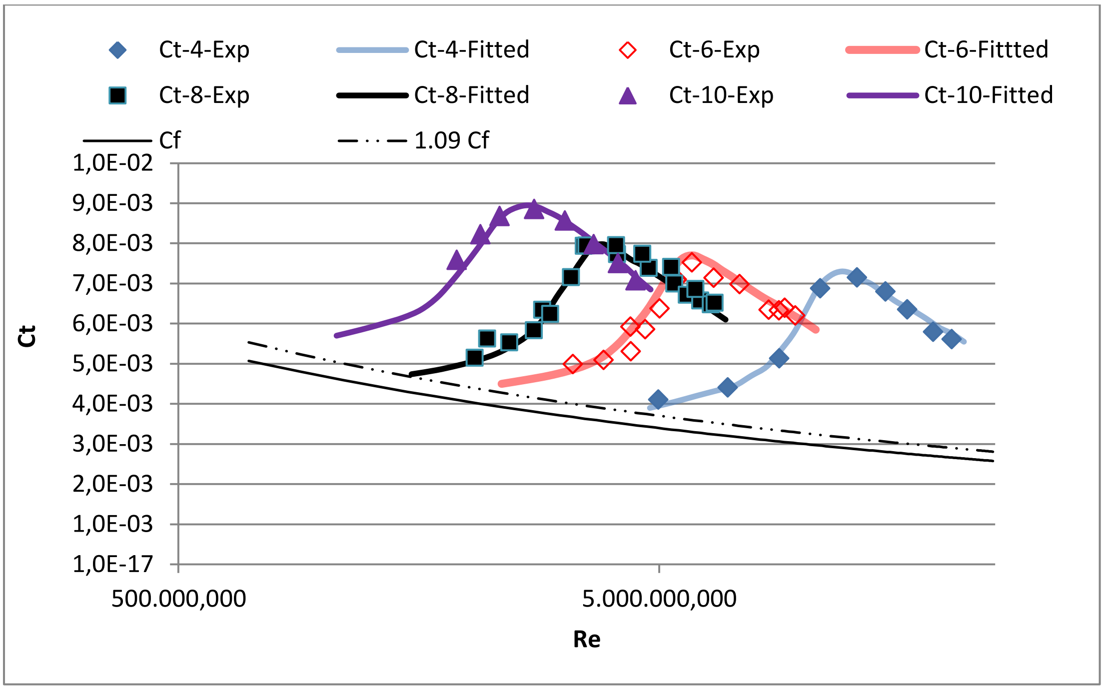

Figure 3.6 : Total resistance coefficent versus Reynold Number for the naked hulls.

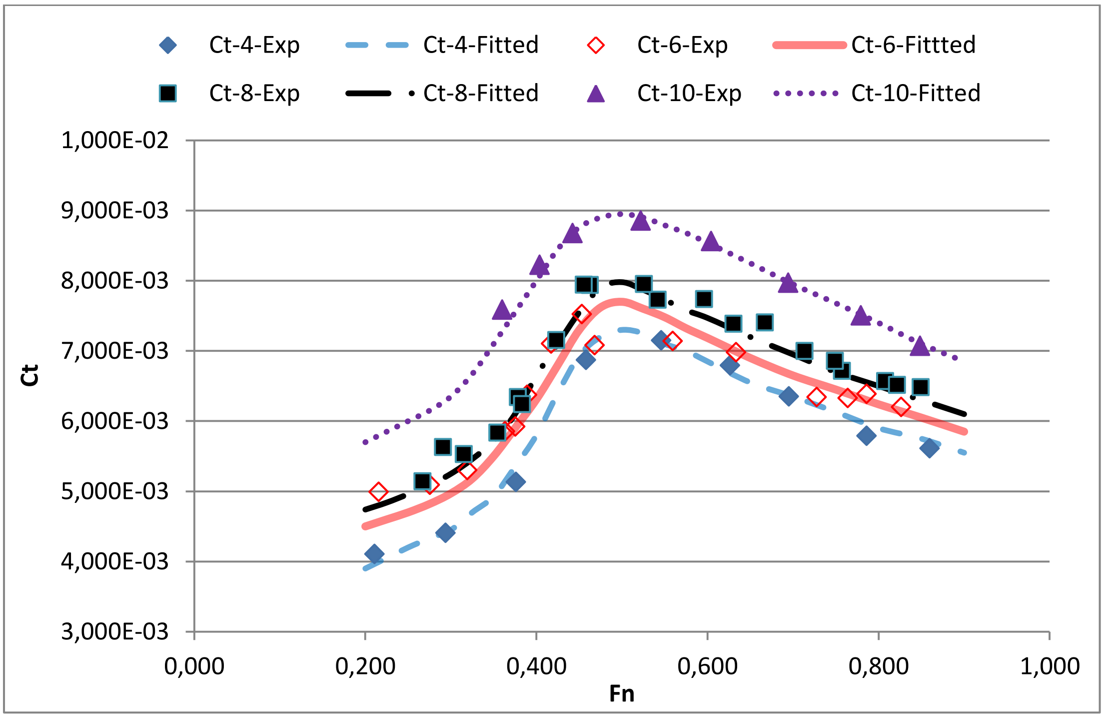

Figure 3.7 : Total resistance coefficient versus Froude Number for the naked hulls.

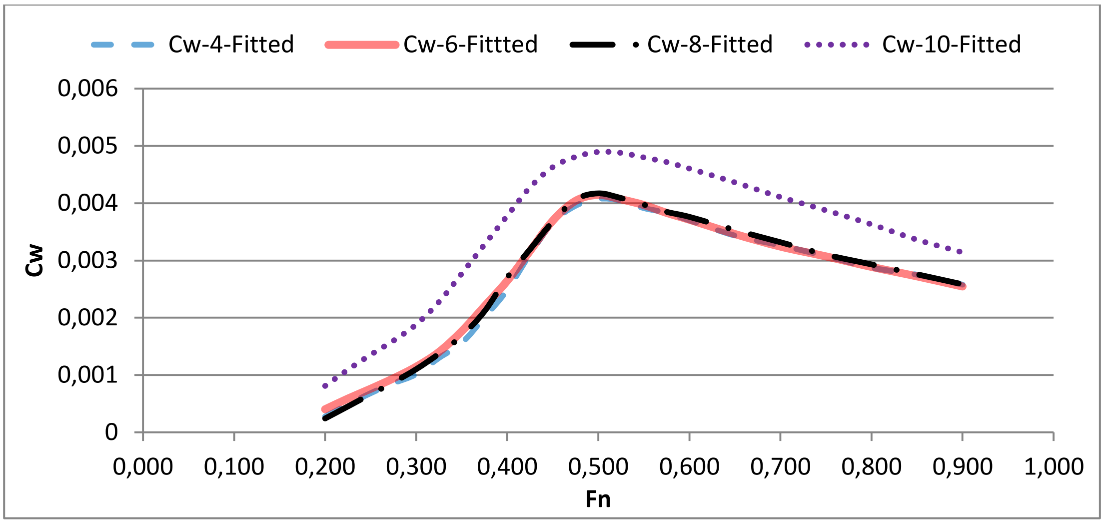

Figure 3.8 : Wave resistance coefficient versus Froude Number for the naked hulls.

The dynamic behaviors of the models are shown in Figures 3.9 and 3.10 respectively. The trimming attitude is consistent across the scale range. Below Froude number of 0.5, there is an excellent agreement in the trim angles. As Froude number increases towards planing regime, certain amount of scatter in the data has been observed.

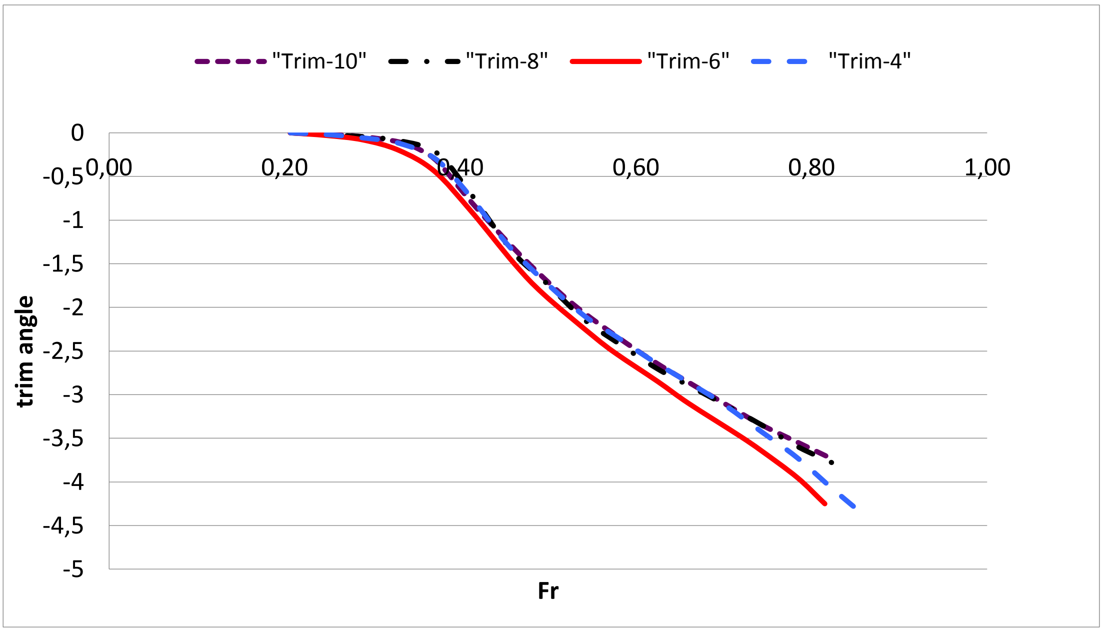

Figure 3.9 : Trim angle versus Froude Number for the naked hulls.

The sinkage attitude is as expected at all scales. All the models tend to sink into their wave system across the Froude number range. Maximum sinkage is at Froude number of 0.5 at all scales when the wave generated is expected to be at its maximum. After Fr>0.5 the models start gaining positive lift. At about a Froude number of 0.8, the models return to their static water line level.

The magnitude of sinkage is less in 1/10 scale compared to other models. This could be associated with surface tension effects as the model size and hence tests speeds are small, the forces in the naked condition at this scale are expected to be insufficient in order to overcome the inertia of the system.

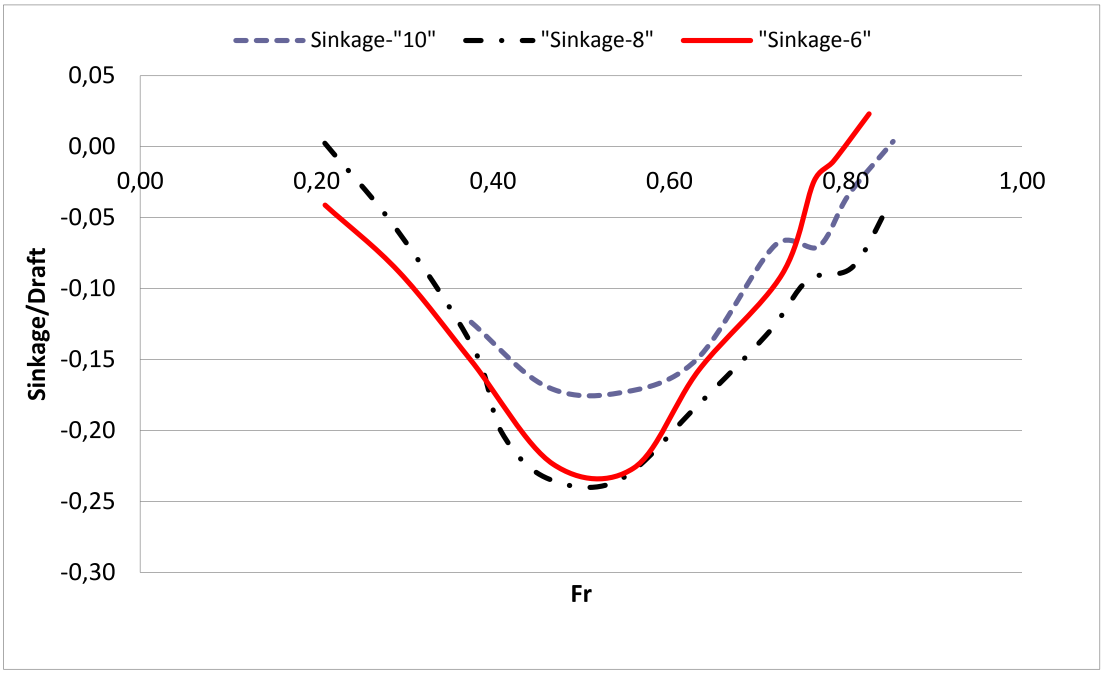

Figure 3.10 : Sinkage/Draft versus Froude Number for the naked hulls.

3.3.2 Upright resistance testing with appendages

Total resistance test results for the appended models are given in Figures 3.11 and 3.12 across Reynolds number and Froude number respectively. Total resistance tests for 1/4 scale model could only be conducted up to a Froude number of 0.5 due to excessive forces generated by the hull which should be avoided on a single post system.

The wave resistance of the appended models are given in Figure 3.13, derived using individual form factors for each appendage, namely, fin keel, bulb and rudder. As in the naked test extrapolation, Grigson’s friction line has been utilized.

Total resistance curve in the appended condition has the same tendencies as the total resistance curve for the naked hull, meanwhile wave resistance curves reveal that the scale effect is further attenuated in case of appended model for 1/10 scale. There are also differences between 1/8, 1/6 and 1/4 scale model results in the higher speed range.

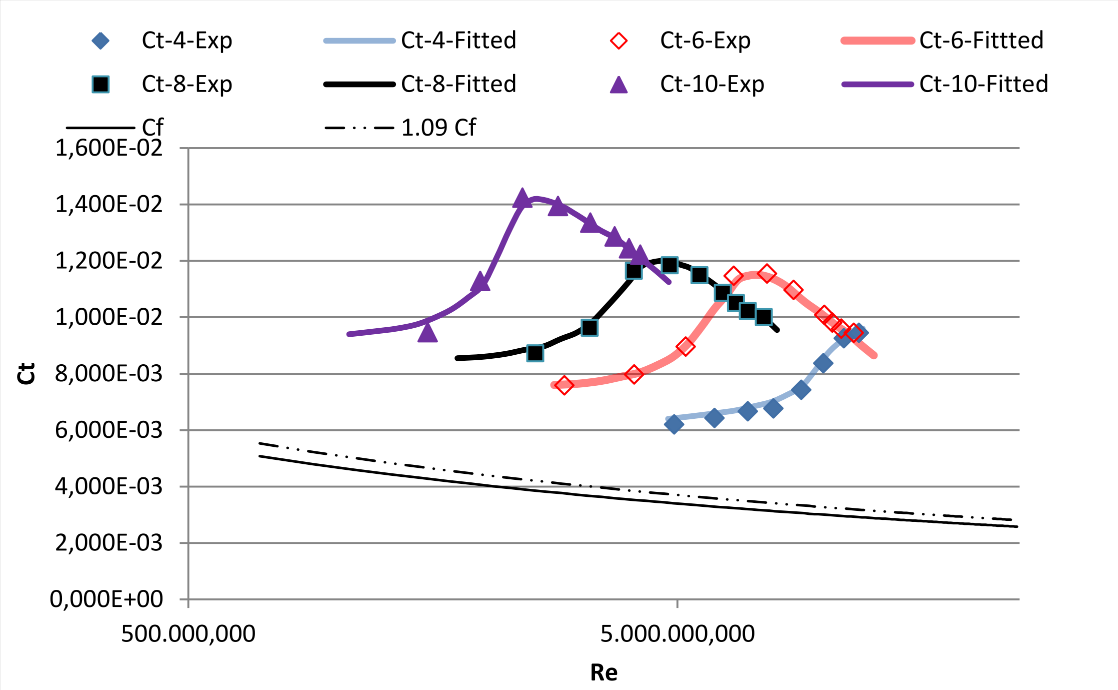

Figure 3.11 : Total resistance coefficient versus Reynolds Number for the appended hulls.

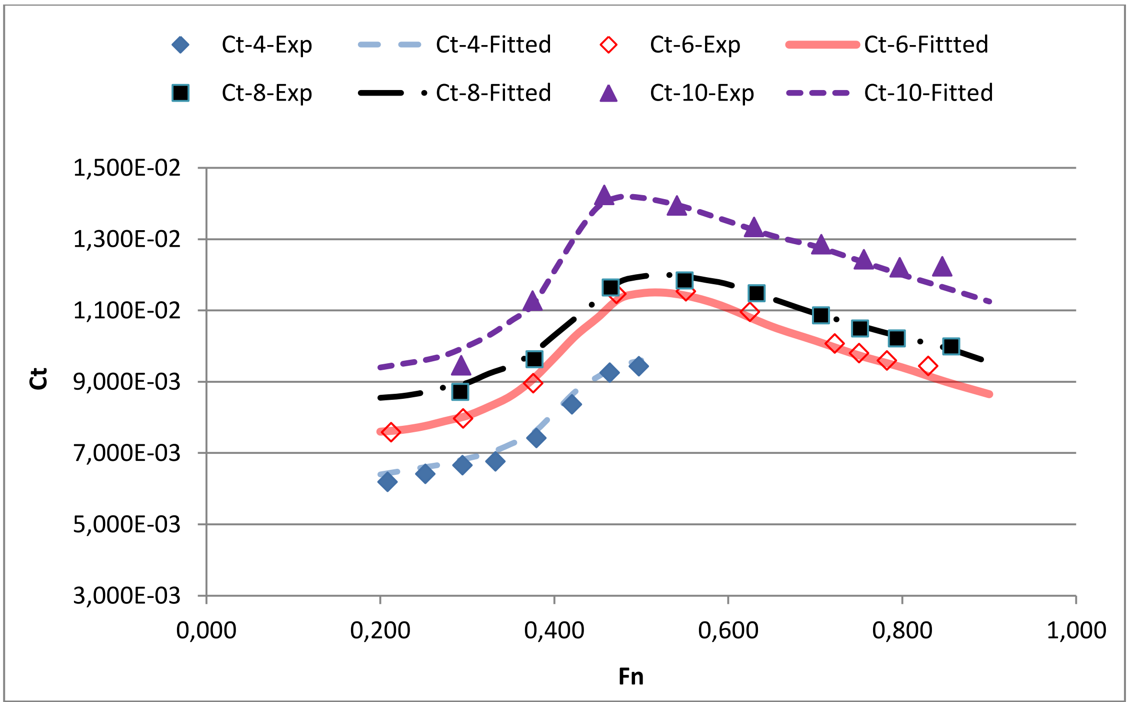

Figure 3.12 : Total resistance coefficient versus Froude Number for the appended hulls.

As the viscous resistance and the resistance of studs were deducted from the total resistance to identify the wave resistance component, it has been seen that an agreement between the wave resistance coefficients in the appended configuration could not be achieved among the tested models. The distribution of the obtained wave resistance curves for scales of 1/10, 1/8, 1/6 and 1/4 suggests ITTC 1978 method with individual form factors may not be sufficient for scaling in the appended condition. The obtained wave resistance values -by subtracting the viscous resistance of the hull and the appendages and turbulence stimulators from the total resistance- may also contain portions of the total drag which has not been broken down according to the current methodology, most possibly from non-wave originated sources. This issue is a deficiency of the utilized force breakdown methodology and it should not be interpreted as differences in the actual wave resistance components. This portion of the drag may be referred to as the drag due to interference of the appendages, the hull and the free surface; and quantification of this drag component requires further investigations.

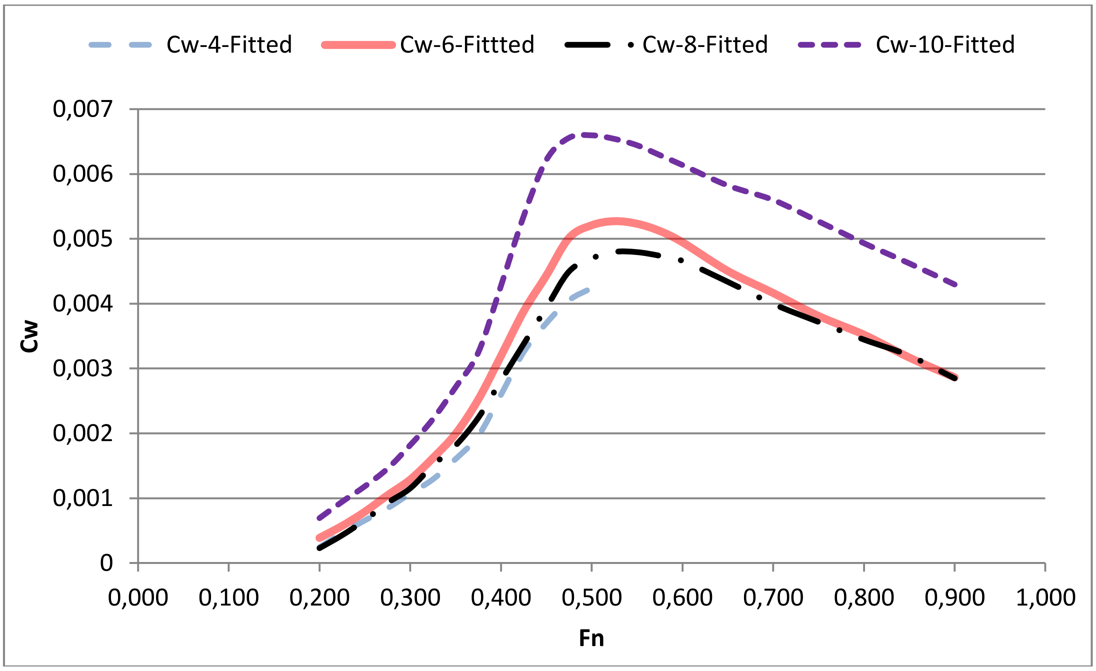

Figure 3.13 : Wave resistance coefficient versus Froude Number for the appended hulls.

The effects of appendages on the wave resistance of the hull is expected to be negligible as they are deeply submerged in water. A comparison between wave resistance of the hull obtained from appended model total resistance tests by subtracting the viscous resistance of hull, fin-keel, keel-bulb, rudder and wave resistance obtained from naked model hull total resistance tests by subtracting viscous resistance of hull is given in Figure 3.14.

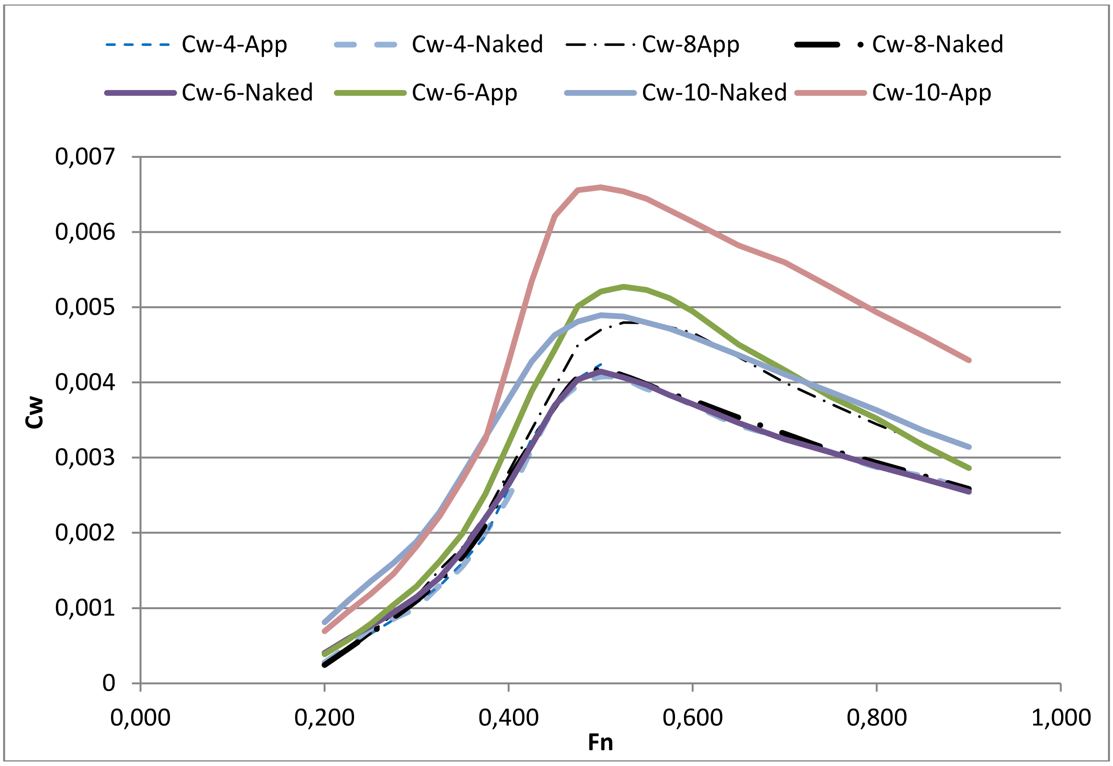

Figure 3.14 : Comparison of naked and appended wave resistance for 1/10, 1/8, 1/6 and 1/4 scale models.

As seen in Figure 3.14, the wave resistance components of the 1/4, 1/6 and 1/8 scale naked hulls are in excellent agreement with the 1/4 scale appended model upto Froude number 0.40. After his speed (0.40<Fn<0.50), there are minor differences between 1/4 scale appended wave resistance coefficients and 1/8, 1/6, 1/4 scale naked wave resistance coefficients. The wave resistance coefficients of the 1/6 and 1/8 scale appended models show similar characteristics –similar values upto Froude number 0.40-, however differences after Froude number 0.40 are substantial between these two models and 1/4, 1/6, 1/8 scale naked model wave resistance coefficients. These results are indicating that individual form factors for hull and the appendages are not sufficient for use in sailing yacht testing campaigns in case of high speed consideration. Differences after Froude number 0.40 may be indicating interaction between hull and appendages, transom stern effects or variation of appendage form factors with speed.

The wave resistance of 1/6 and 1/8 scale appended models and 1/10 scale model in both conditions show different tendencies comparing to the naked condition, especially at Fn>0.4. Although this was expected for the 1/10 scale model in both conditions, the 1/6 and 1/8 scale models also suffer from scale effects in the appended condition; as appendages are accounted for, the current extrapolation methods are not satisfactory at these scales. Although there is a decisive reason for this scale effects, a number of parameters must be considered. The variation of form factors across Froude number, interaction resistance between the hull-keel, keel-bulb and hull-rudder, the wash effect of the keel wake on the rudder and free surface effects on the appendages can be mentioned here.

The dynamic behaviour of the models are similar to what's been observed in the naked condition; there is excellent aggreement between different scales in trim upto Froude number of 0.5 and a scatter of +- 0.2 degrees at higher Froude numbers.

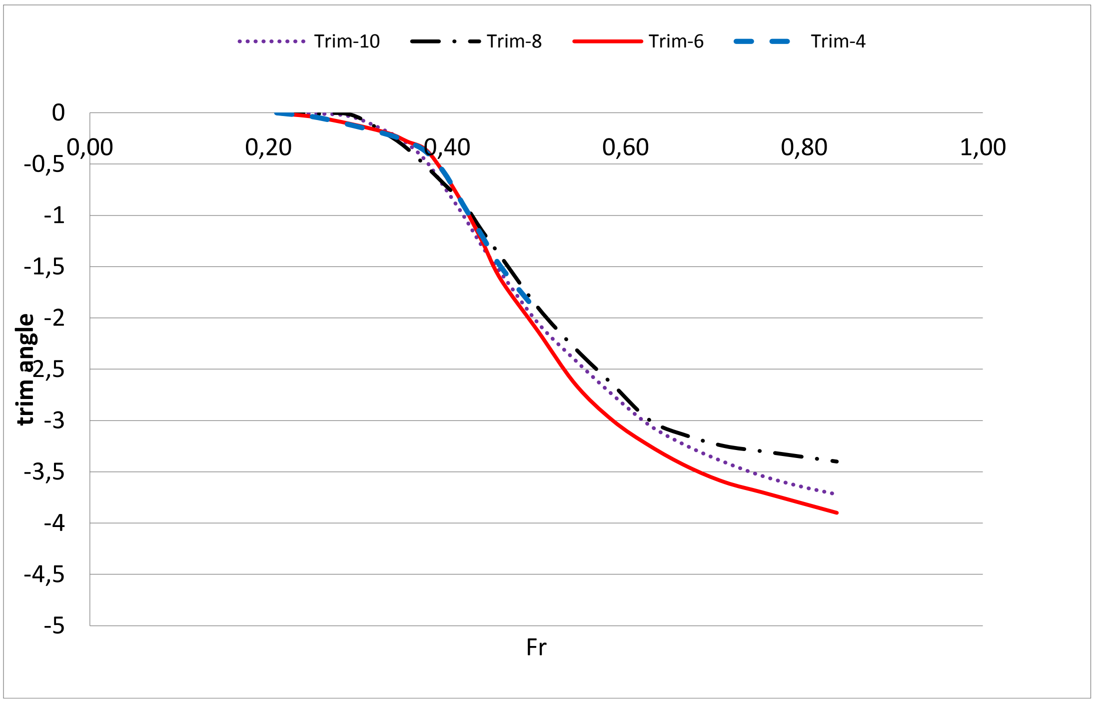

Figure 3.15 : Trim angle versus Froude Number for the appended hulls.

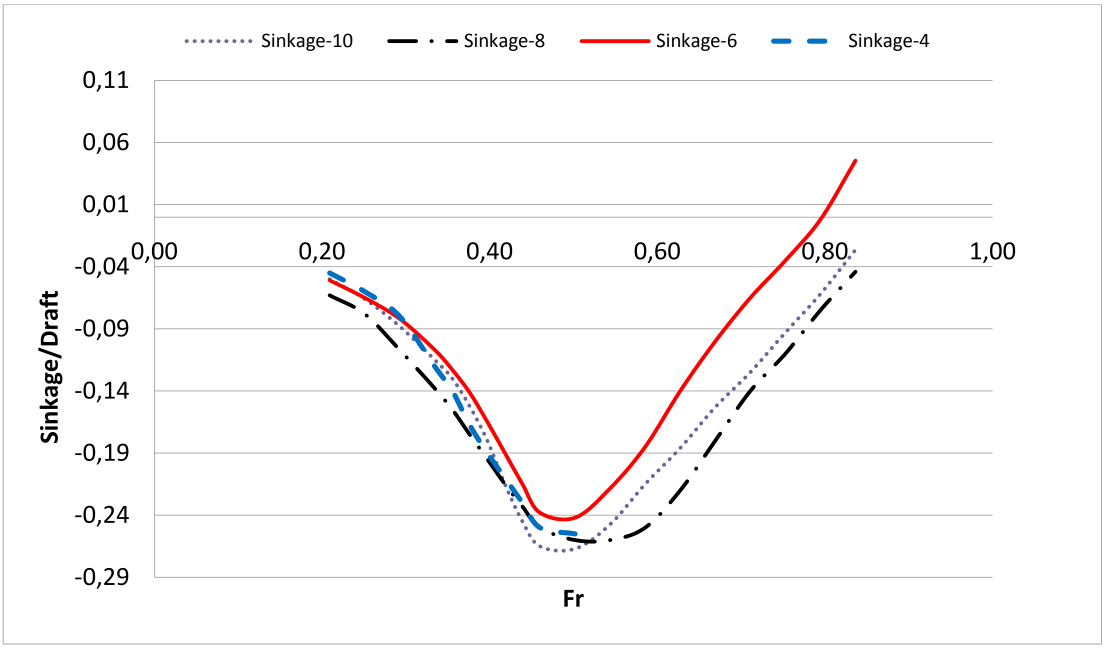

Figure 3.16 : Sinkage/Draft versus Froude Number for the appended hulls

3.3.3 Upright wave profile testing

Several attempts were made in the past for investigation of wave pattern resistance of sailing yachts. Influence of appendage configuration (Binns, 1997) and hull form variations (Scragg, 1987) on the performance of the yacht may be assessed with the longitudinal wave cut technique.

Wave pattern experiments with all four models were conducted both in naked hull and appended hull configurations with the keel, bulb and rudder. The waves generated by the models were acquired through three wave probes at both sides and a computer aided data acquisition system in order to make comparisons of wave pattern resistance for the scale effects.

The wave pattern analysis was conducted with multiple longitudinal cut method with matrix solver (Insel, 1990). The wave pattern traces were separated into symmetrical and asymmetrical components by using the wave probes on both sides as stated in section 3.2.3.

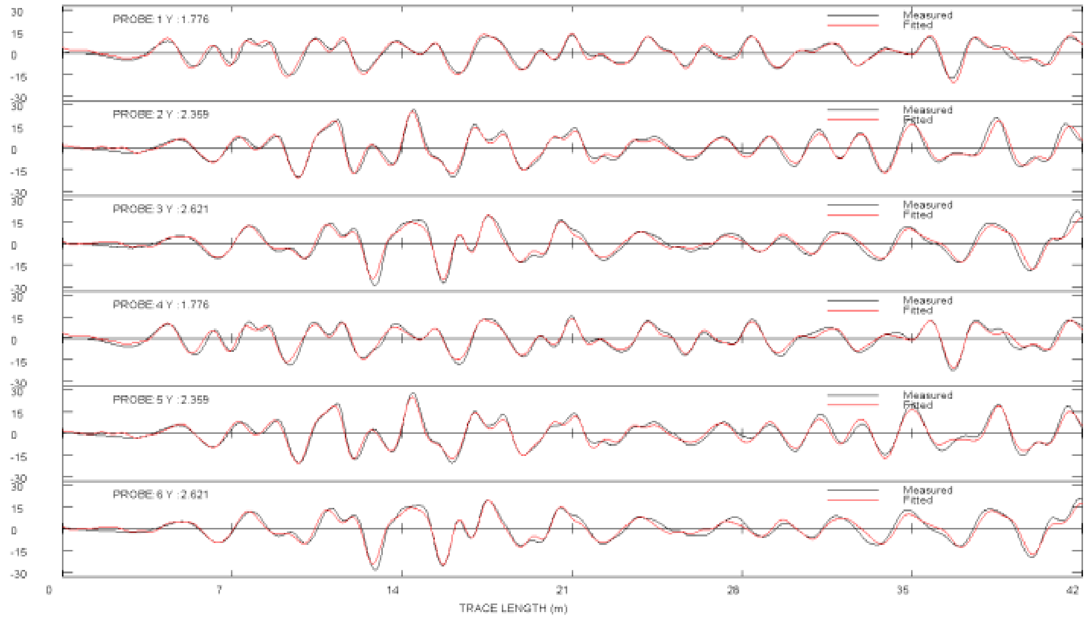

Figure 3.17 : Wave pattern typical traces for Froude number of 0.45.

The wave pattern resistance comparison among the 4 different scale hulls is given in Figure 3.18. Even though the total resistance results show significantly higher resistance coefficient for the smallest model, i.e.1/10 scale, as shown in Figure 3.18, the wave pattern results are approximately the same. The scale effect on the wave pattern resistance is confirmed to be negligible even for the 1/10 scale model.

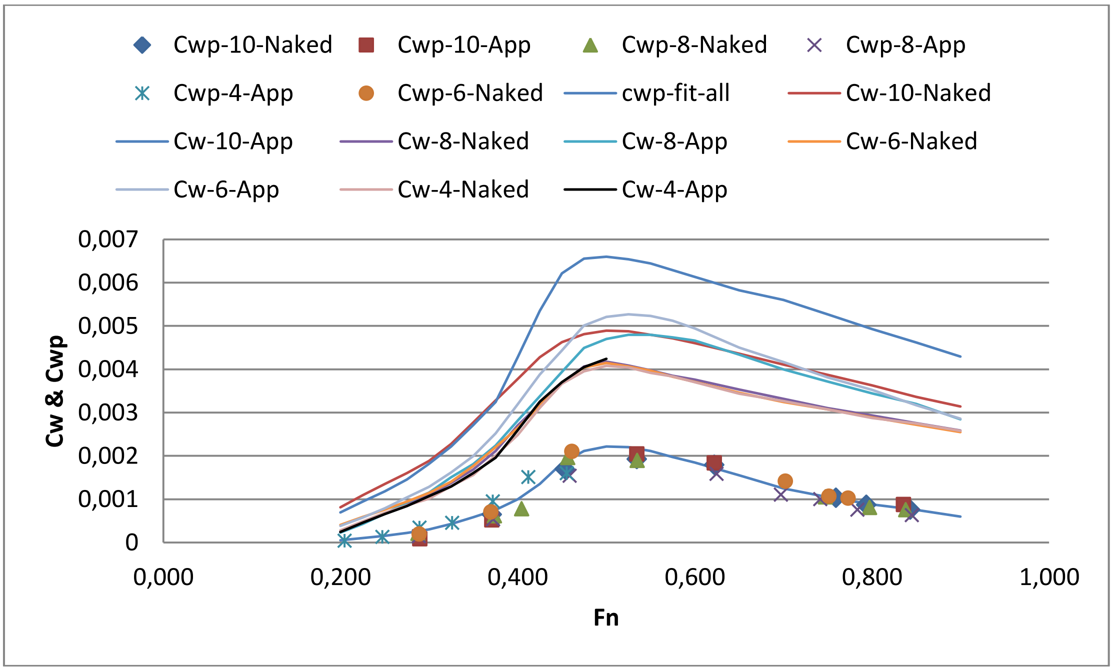

Figure 3.18 : Wave pattern for all four models.

As the yachts were tested floating at the same waterline in naked and appended conditions, the wave patterns in these two testing conditions were not variant. Figures 3.19, 3.20 and 3.21 show comparisons of wave pattern traces in naked and appended conditions for 1/8 scale model at Froude Number 0.62.

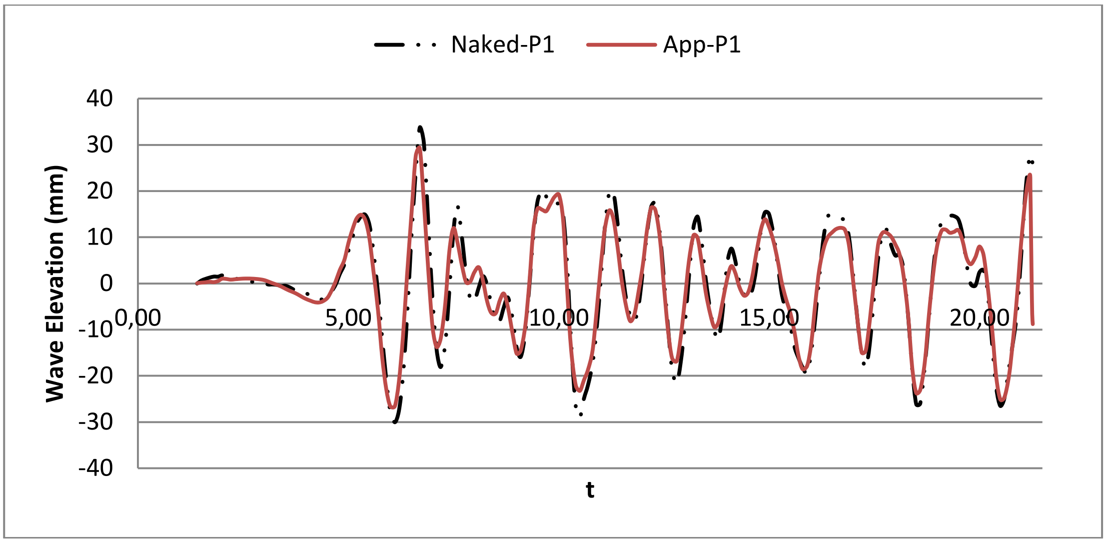

Figure 3.19 : Wave traces in naked and appended condition for 1/8 scale model at Probe 1 at Fn=0.62.

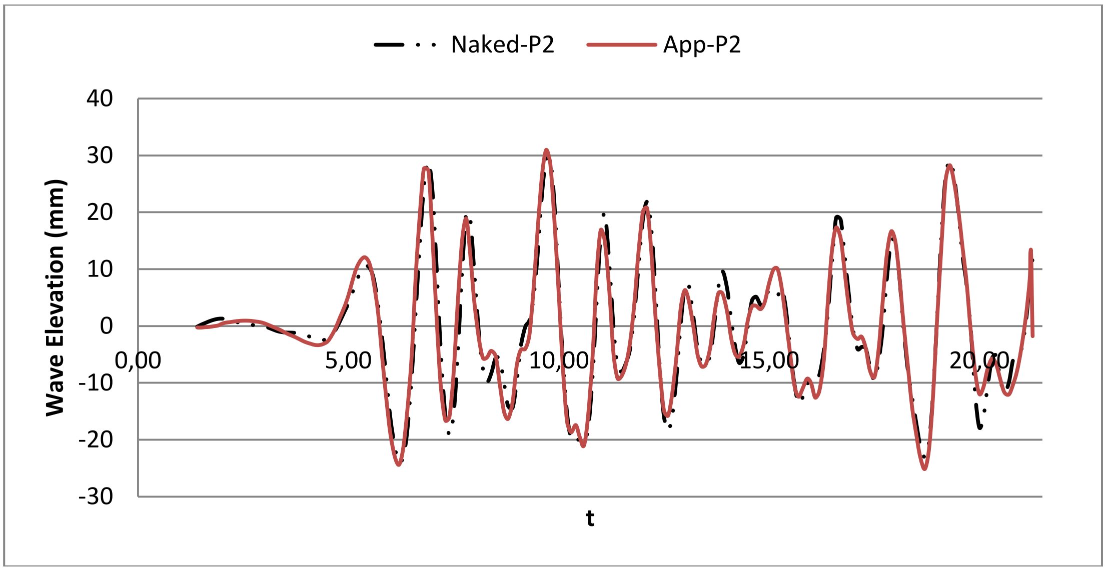

Figure 3.20 : Wave traces in naked and appended condition for 1/8 scale model at Probe 2 at Fn=0.62.

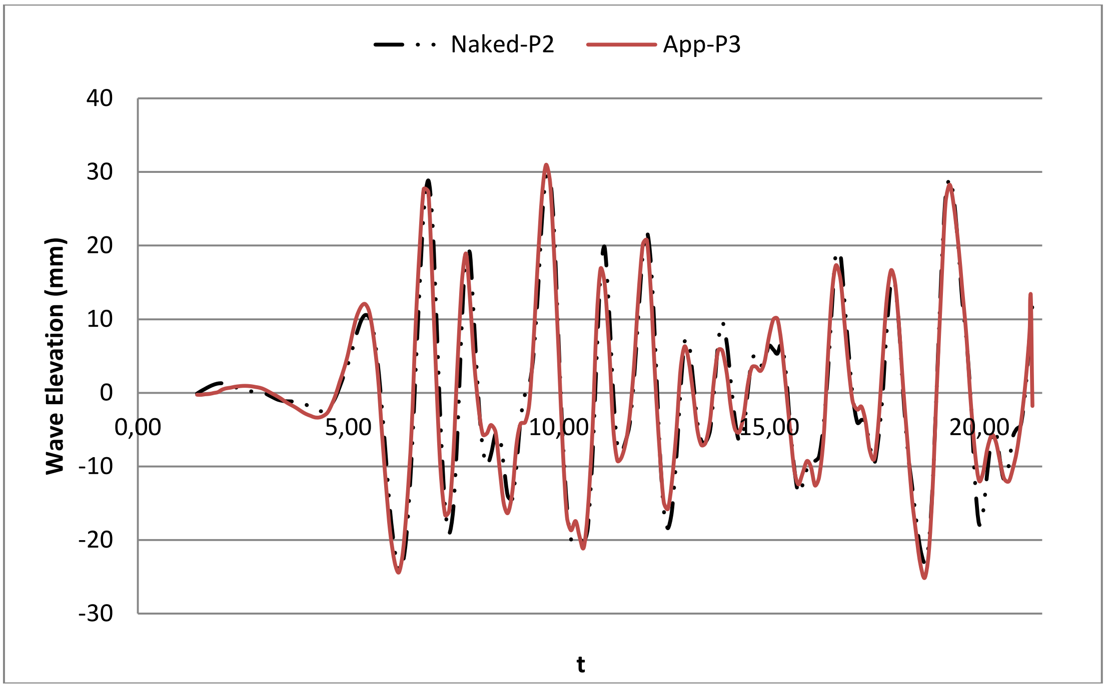

Figure 3.21 : Wave traces in naked and appended condition for 1/8 scale model at Probe 3 at Fn=0.62.

By conducting comparisons not only on the basis of wave pattern resistance but also on the harmonic bases, the change in the wave making can also be investigated. Due to the characteristic of scaled models tested in the same tank, the wave spectrum angles are different for each scale due to different test speeds to keep Froude number constant. Wave resistance distribution across the wave spectrum for two different speeds are given in Figure 3.22 and 3.23. The wave pattern resistance distribution changes are clearly visible from both Figures. The shift of wave resistance from the transverse to diagonal waves is apparent for smaller scales.

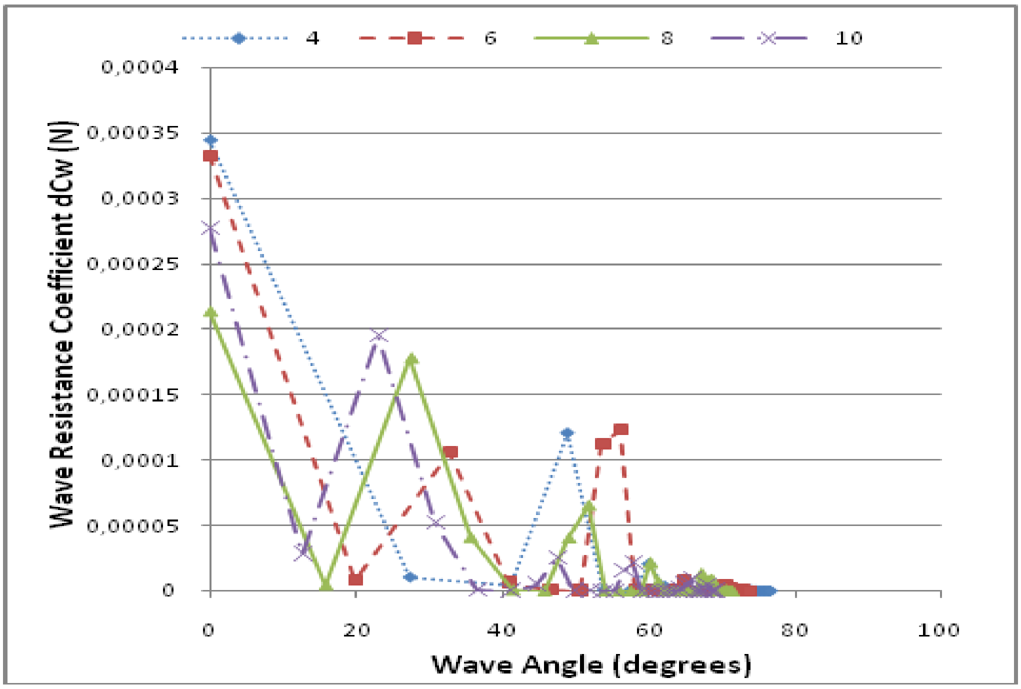

Figure 3.22 : Wave pattern resistance distribution across the wave spectrum for each model scale for Froude number of 0.45.

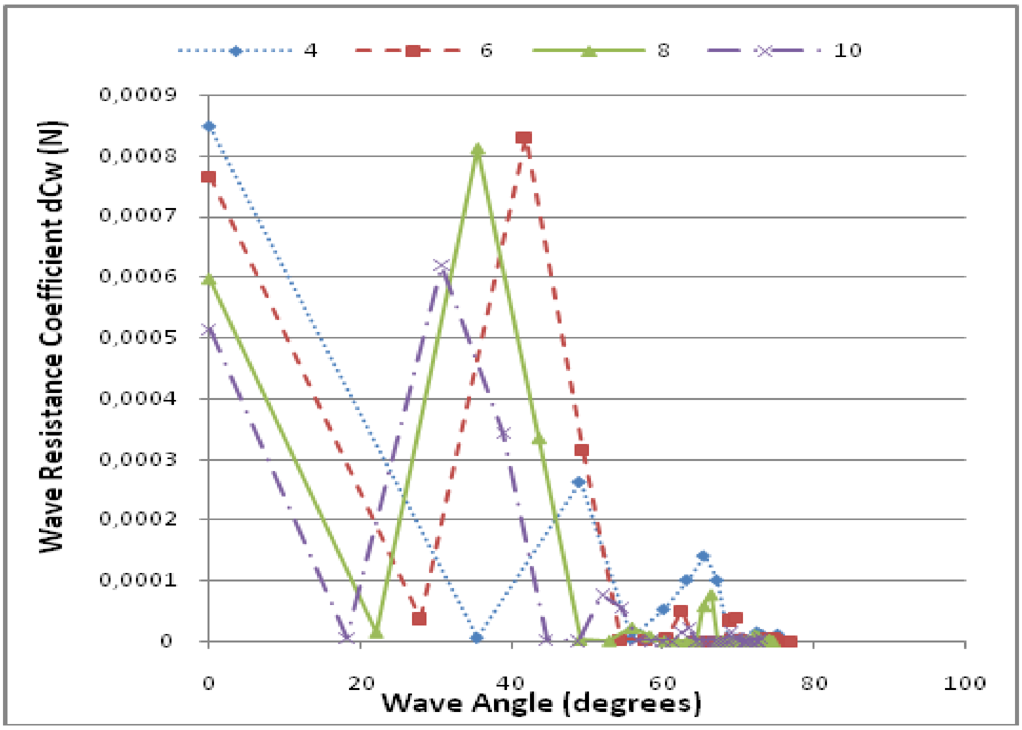

Figure 3.23 : Wave pattern resistance distribution across the wave spectrum for each model scale for Froude number of 0.75.

3.3.4 Discussions on the results of upright tests

During the upright tests, it has been concluded that the scale effects in wave pattern resistance are negligible for all four models both in naked and appended conditions. Although the wave pattern spectrums are different for each scale due to testing speed difference at equivalent Froude number, wave pattern resistance values derived from these spectrum are approximately the same. There is also negligible difference between the naked and appended cases indicating that the appendages have no influence on the wave pattern and wave pattern resistance.

However, the obtained wave resistance component, which has been derived from subtracting calculated frictional and form drag of hull and appendages from the total resistance measurement, displays scale effects. 1/10 scale model wave resistance is clearly different from other models in both naked and appended conditions. Additionally, the derived wave resistance of 1/8 and 1/6 scale models are also higher than the 1/4 scale results in the appended condition.

As the ITTC models have been developed for conventional ships for which viscous and wave based resistance interactions are negligble; the derived wave resistance normally includes wave pattern and wave breaking resistance components. For a sailing yacht with transom stern and an appendage configuration comprising a subsantial portion of the total wetted area, the wave resistance defined by the current methodology following ITTC 1978 method, includes the wave pattern resistance and other resistance components such as wave breaking resistance, viscous pressure resistance, transom stern associated resistance, appendage-hull interaction drag, appendage-appendage interaction drag, appendage form drag variations with speed etc. Figure 3.27 demonstrates the difference between the wave resistance and the wave pattern resistance in naked upright condition. Differences between 1/8, 1/6 and 1/4 scale models are negligible, indicating scale effects are not important for this drag component. However, 1/10 scaled model results are different, hence models with a length less than 2 m may not be appropriate for use in naked sailing yacht testing campaigns.

As the actual wave resistance should not vary from other models at 1/10 scale, the computed wave resistance at this scale is estimated to include the experimental errors due to testing a 1.58 metre long model.

When the sources of the errors encountered at 1/10 scale testing are investigated, the first phenomenon that needs to be questioned is the repeatability of the measurement system. As the measured forces are smaller compared to other tested models, it has been deemed necessary to investigate the effect of repeatability on the results of the 1/10 scale tests.

The maximum resistance value measured in the naked condition for the 1/10 scale model was 1.590 kilograms at Froude number of 0.90. The minimum value was 0.067 kilograms at Froude number of 0.20. The repeatability, as mentioned in the previous sections, has been found to be in the order of 30 grams of resistance force. It is clearly evident that the repeatability, altough influencial on the differences in the calculated wave resistance at low Froude numbers, may not be sufficient to explain the differences across the whole Froude number range. The influence of repeatability on the calcualted wave resistance is higher than 10% at Froude numbers less than 0.50 and drops to as low as 2% at higher Froude numbers.

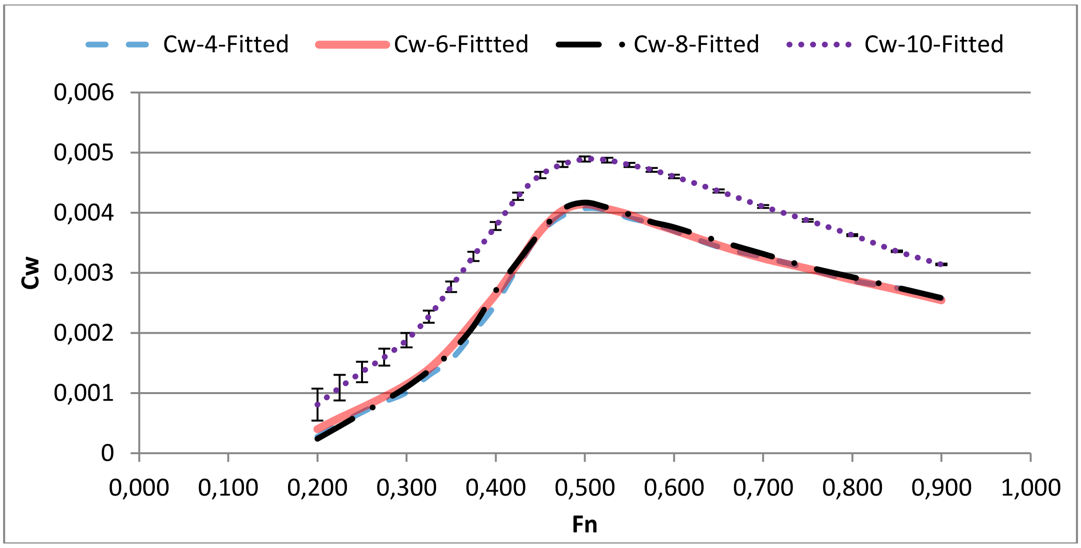

Figure 3.24 : Influence of repeatability on the wave resistance coefficient results for 1/10 scale.

As the repeatability investigation has not been conclusive, the philosophy behind the experimental procedure has been reinvestigated. The current methodology utilized during the experiments relies heavily on empirically calculated parameters such as frictional resistance, viscous resistance and form factors (for appendages). Any error in the calculation of these parameters would lead to misleading results. Therefore, these have been reconsidered with emphasis on the influence of the obtained results.

In the naked computations, the frictional resistance has been calculated with the aid of static wetted area as per ITTC's recommended procedures. As there is not sufficient dynamic behaviour difference across the speed range for ships, this procedure has been found to be producing sensible results. For a sailing yacht that operates up to Froude numbers of 0.9, the hydrodynamic regimes differ substantially across the Froude number range.As shown in the previous sections, the yachts tend to trim up to 3.5 degrees and the sinkage values differ substantially across the speed range. This dynamic attitude, especially after Froude number of 0.40 results in the variation of wetted area and hence the frictional resistance.

Smaller models are more prone to this type of error, as these operate at lower Reynolds numbers and hence have higher frictional resistance coefficients. Therefore, as static wetted area is utilized in computation of the wave resistance, an error is introduced and as the model size becomes smaller, this effect is more pronounced.

However, measurement of dynamic wetted area introduces certain difficulties. Photogrammetry techniques need to be applied during testing and these equipment are not readily available in majority of the testing facilities.

As a result, the dynamic wetted area has been approximated with the aid of sinkage and trim data acquired during the experiments. The analysis indicate that the wetted area increase is up to 17% of the static wetted area at Froude numbers between 0.45 and 0.50 for the range of models tested. The differences in the wetted area change is at the order of 1% between scales. However, the results of naked 1/10 scale model differ from other up to 4%. In addition, the trim and sinkage tendency of the 1/6 scale appended model are not consistent with the rest of the experiments. Therefore, the results have been reprocessed with the individual dynamic wetted area data for each set. In Figure 3.25, the typical wetted area variation across the Froude number range may be seen.

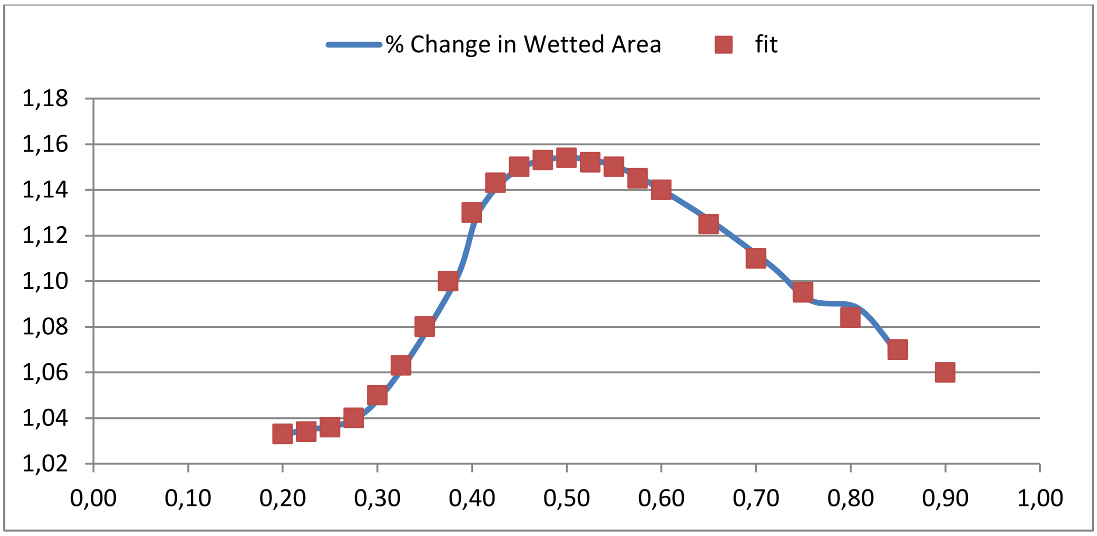

Figure 3.25 : Variation of wetted area with Froude number for 1/8 scale naked condition.

Consideration of the dynamic wetted area with respect to trim and sinkage data yields a variation in the frictional resistance; at the order of 15% for moderate speeds. The effect is more pronounced at slow speeds, differences are upto 25%. At high speeds, as the wave resistance becomes the dominant portion of the total resistance, the decrease in the computed wave resistance is at the order of 10%. As the frictional resistances values differ between models due to difference of Reynolds numbers, the difference between the computed wave resistances of the largest and smallest model due to misinterpretation of the wetted area is in the order of 3%. However, utilization of static wetted area would cause exactly specified amounts of error in terms of full scale extrapolation.

The wave resistance coefficients, calculated by using dynamic wetted area may be found in Figure 3.26.

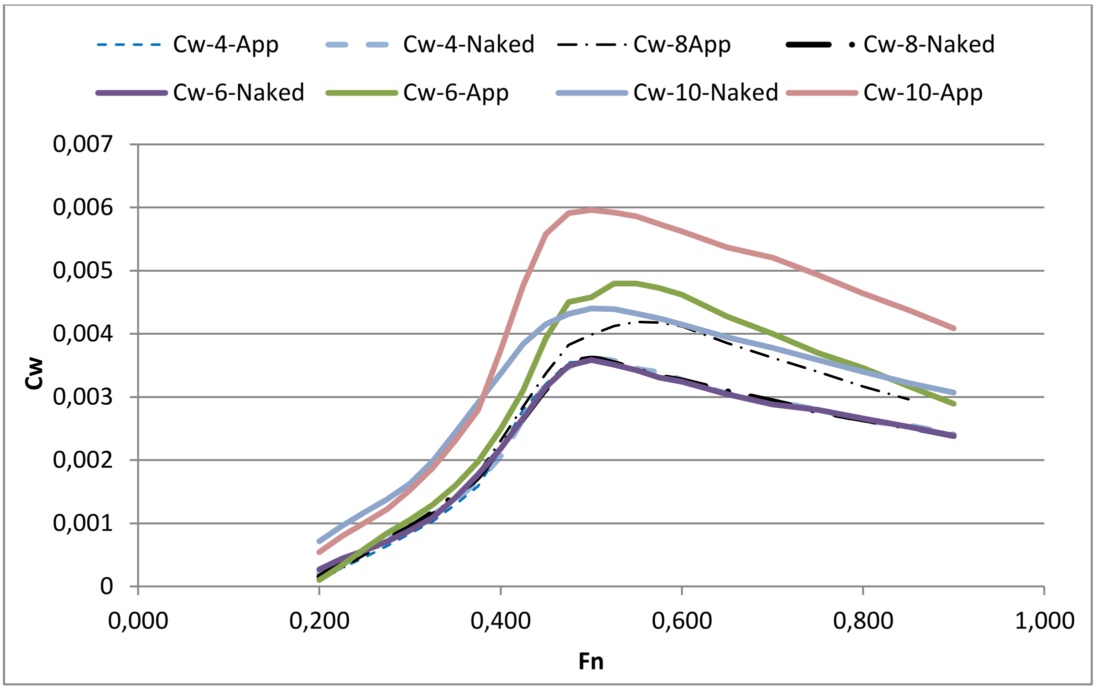

Figure 3.26 : Wave resistance coefficients calculated using dynamic wetted area for naked and appended models.

It has been found out that when the wave resistance is computed according to the utilized methodology, it includes the error that has been introduced due to the utilization of static wetted area for the whole speed range , apart from wave pattern and wave breaking resistance components. Altough this error has been found to comprise 15% of the remaining portion of the wave resistance at moderate speeds, it is not sufficient to explain the global differences in the calculated wave resistances in both naked and appended conditions.

Altough larger models are required for appended tests, small size models (between 2m and 4m) may be utilized for earlier stages of testing campaigns; in which the main emphasis of the tests are hull form optimization, wave drag minimization or similar. This would yield a more cost-effective solution with a quicker turn around time and would also enable testing in a smaller facility as well as makes it possible to utilize a larger test matrix in a given time interval. The uncertainty for models smaller than 2.0 meters is far beyond the expected accuracy and interpretation of the test data at this scale lead to misleading results with the currently available testing equipment and experimental techniques.

In the appended condition as given in Figure 3.28, the difference between the computed wave resistance and the wave pattern resistance is more influenced by the scale of the models. There are differences between 1/4, 1/6, 1/8 and 1/10 scales, indicating scale effects exits in the appended condition.

The difference between wave resistance and the wave pattern resistance above Froude number 0.45 is almost constant for all the models both in naked and appended conditions. Although this feature may not be fully understood with currently available data and analysis, it may well indicate Froude number dependancy of this portion of resistance as well as the effects of trim, sinkage, transom stern drying effects, insufficient turbulence stimulation etc.

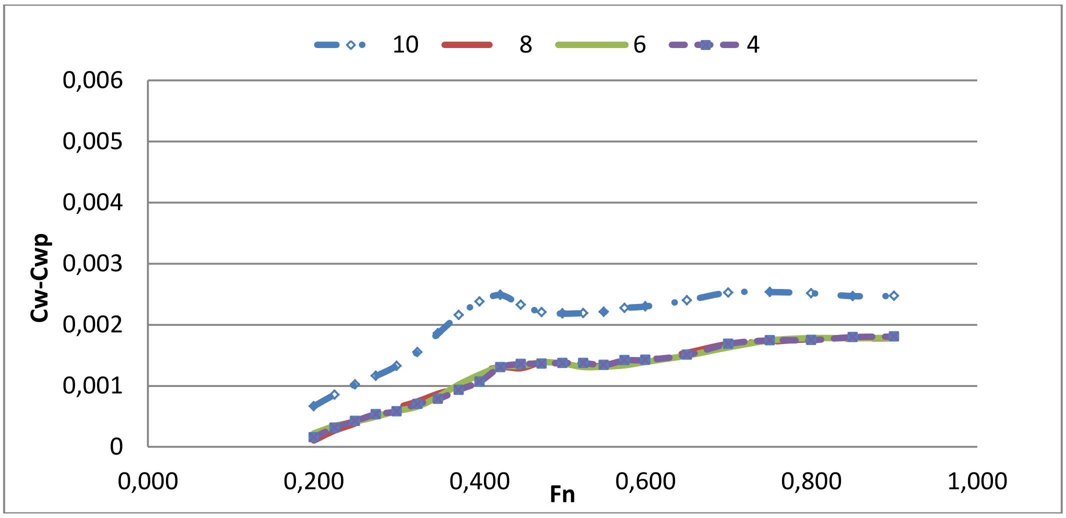

Figure 3.27 : Difference between wave resistance and wave pattern resistance in coefficient form for the naked hulls.

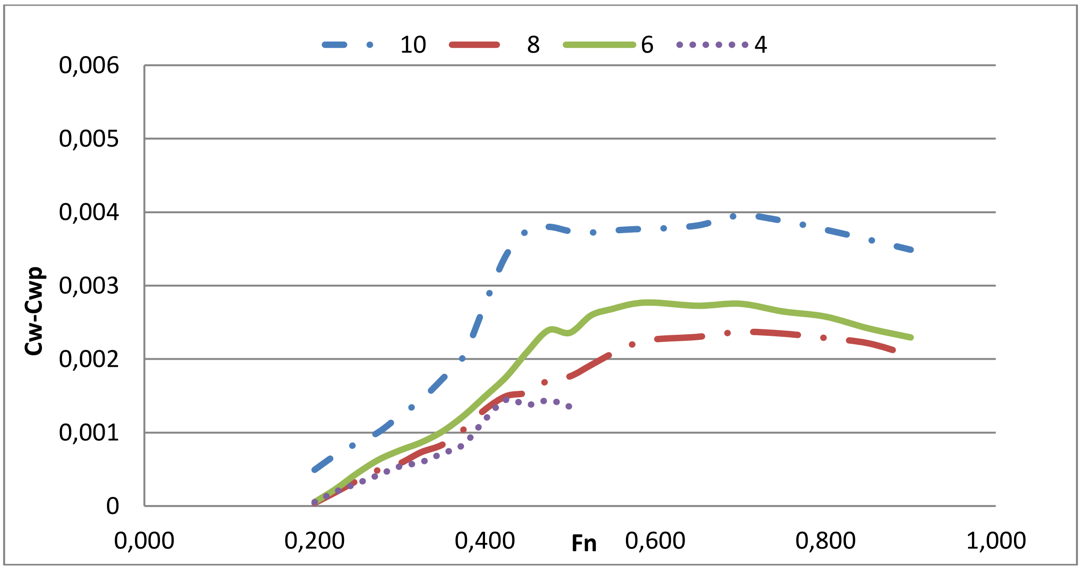

Figure 3.28 : Difference between wave resistance and wave pattern resistance in coefficient form for the appended hulls.

In Figure 3.29, the difference between wave resistance and wave pattern resistance in naked condition is subtracted from the appended condition. Altough it has not been possible to explore the trends in the variation of this parameter with just force measurement, this resistance component may be associated with appendage effects other than frictional and form drag, which may be calculated empirically. The value of this component is negligible up to Froude number 0.4, and scale effects are observed for 1/6, 1/8 and 1/10 scaled models. Meanwhile, there is minimal effect of appendages across the Froude number range for the 1/4 scaled model.

There has been certain efforts in the past in order to investigate the influence of appendages on the wave making characteristics of sailing yacht models (Keuning & Sonnenberg, 1998). Altough general applicability of these research efforts are still questionable, the normal component of appendage resistance has been found to vary with the vertical position of the center of bouyancy of the tested keels. Small scale models are expected to suffer from this phenomenon as the appendage center of buoyancies are closer to the free surface. The Froude dependent nature of Cw-Cwp component in the appended condition, as identified from Figure 3.29 and the tendency of the variation of Cw(app)-Cw(naked) plot in Figure 3.30 also raises suspicion regarding the influence of the appendages on the computed wave resistance as any change in the normal resistance component of the appendages not accounted within the empirical appendage resistance calculation would be included in the computed wave resistance component.

However, the mentioned research of Keuning and Sonnenberg has limited data in order to derive generally applicable conclusions owing to the fact that the tests were conducted on models with lengths varying between 3.15 metres and 5.73 metres. The differences associated with smaller scale testing may not readily be accounted for within the polynomial fits on the mentioned data.

The current methodology of estimating the resistance of the appendages is of aerodynamic nature (Hoerner, 1965). Any difference in the flow characteristics between foils operating in air and in close proximity to the water surface is not accounted for within the formulation.

Calculation of appendage viscous resistance with stripping method has been proposed in previous research campaigns (DeBord & Teeters, 1990). In this methodology, the appendages are stripped into veritcal sections and treated individually in evaluation of frictional and viscous resistance components. This method was mainly used for the 1987 America's Cup campaign for the Stars&Stripes team. As the appendages being utilized then were with lower aspect ratios and higher sweep angles, consideration of different Reynolds numbers along the span of the appendages were causing improvements in the viscous resistance estimation of the appendages. For high aspect ratio appendages with minimal sweep, the influence was estimated to be less compared to previous attempts. The effort of stripping the keel into 4 vertical components for the current analysis has yielded at change at the computed wave resistance coefficient less than 0.1% at all scales and speed ranges. Therefore, this methodology has been considered to be uninfluencial on the results for the current appendage system. For cruising vessels, with longer keels with lower aspect ratios, consideration may be given to stripping the appendages in viscous resistance estimation at the post-processing stage.

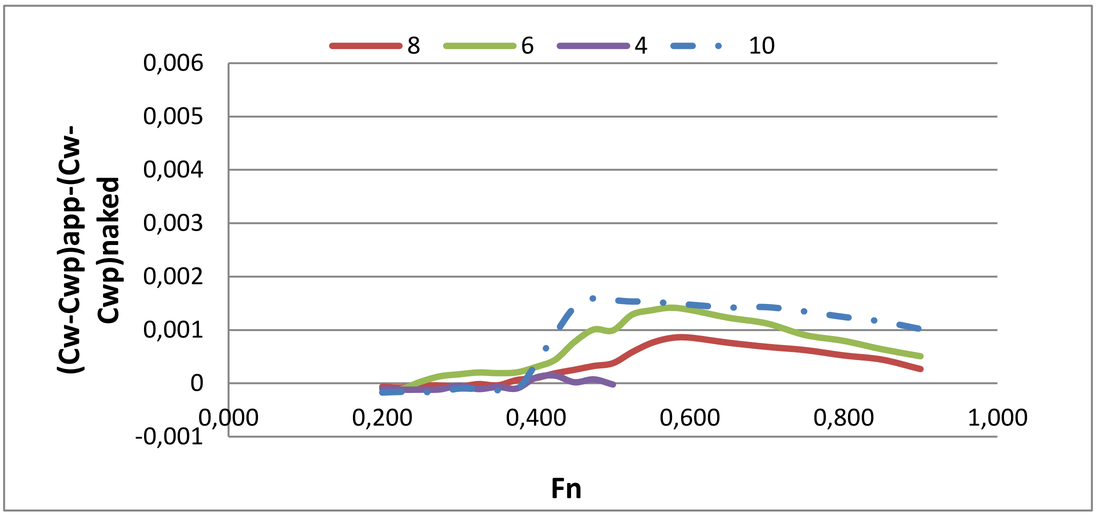

Figure 3.29 : Difference of Cw-Cwp between appended and naked condition.

As the wave resistances should be identical in coefficient form among the scale range, inclusion of this portion of resistance within the wave resistance component may lead to misconceptions. Therefore, the sources of this resistance component needs to be investigated with suitable tools like PIV measurements, CFD, or similar. For convenience purposes, this investigation has been carried out by CFD tools and presented in the relevant sections.

3.4 Heeled Testing

Following upright resistance tests, towing tests were conducted for scales 1/10, 1/8 and 1/6 with heel angle of 10, 15 and 25 degrees. It was not possible to test the largest model at 1/4 scale for all heel angles with the single post system due to high forces generated on the measurement system, hence only 15 degrees heel angle at 8 knots of full scale boat speed was tested.

3.4.1 Tests without leeway angle

Resistance change due to heel angle must be determined by total resistance tests without leeway angle. Heeled resistance tests results are presented in Figure 3.30 for scales of 1/10, 1/8 and 1/6. Resistance of the boat increases at low heel angles, i.e. 10 and 15 degrees for smaller models, meanwhile there is a resistance decrease for the larger model for 10 and 15 degrees of heel. Resistance decrease at 25 degrees of heel is encountered for all the scales.

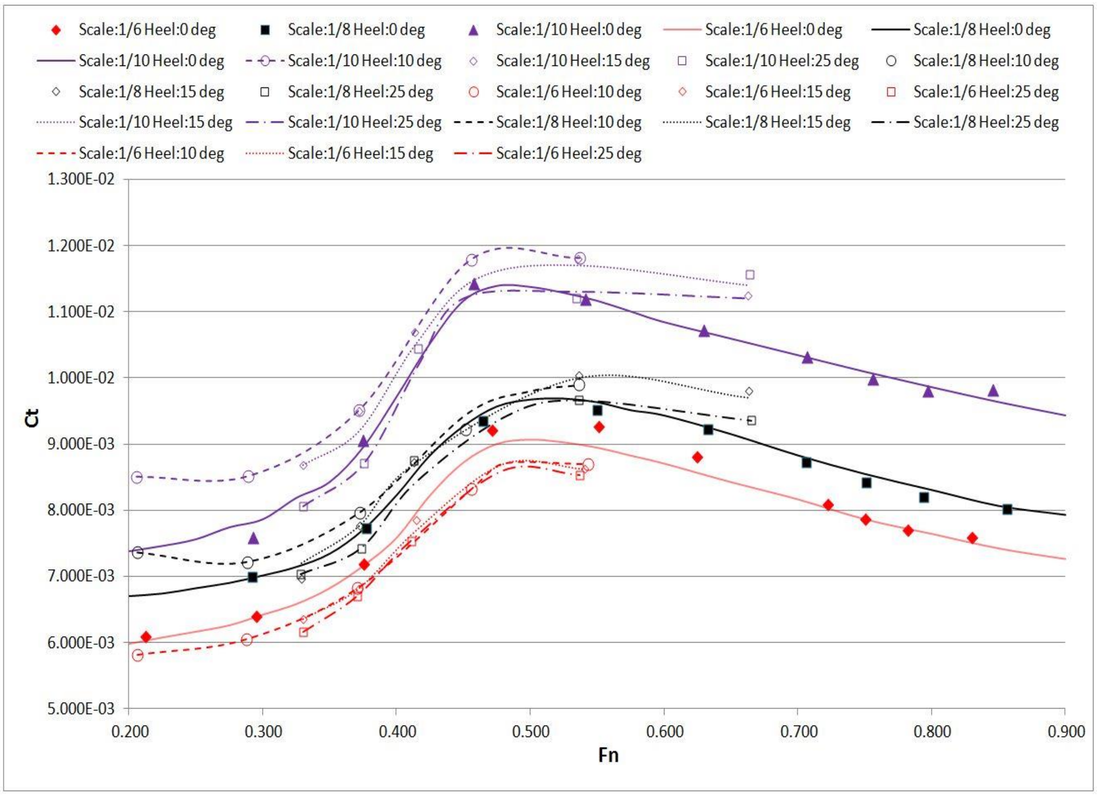

Figure 3.30 : Heeled total resistance in comparison with upright resistance.

Figure 3.31 and Figure 3.32 demonstrate the heel resistance for heel angles of 0, 10, 15 and 25 degrees for the model scales of 1/10, 1/8 and 1/6 for 9 and 13 knots of full scale boat speed. Heel resistance decreases with larger model sizes. Scale effects on the heel resistance are substantial for the range of heel angles for both speeds.

In practical sailing conditions, the heel angle is accompanied by the leeway angle as aerodynamic heeling force must be balanced by a hydrodynamic side force which can only generated by a leeway angle. Hence heel resistance is not needed to be considered individually and usually accounted together with the induced drag.

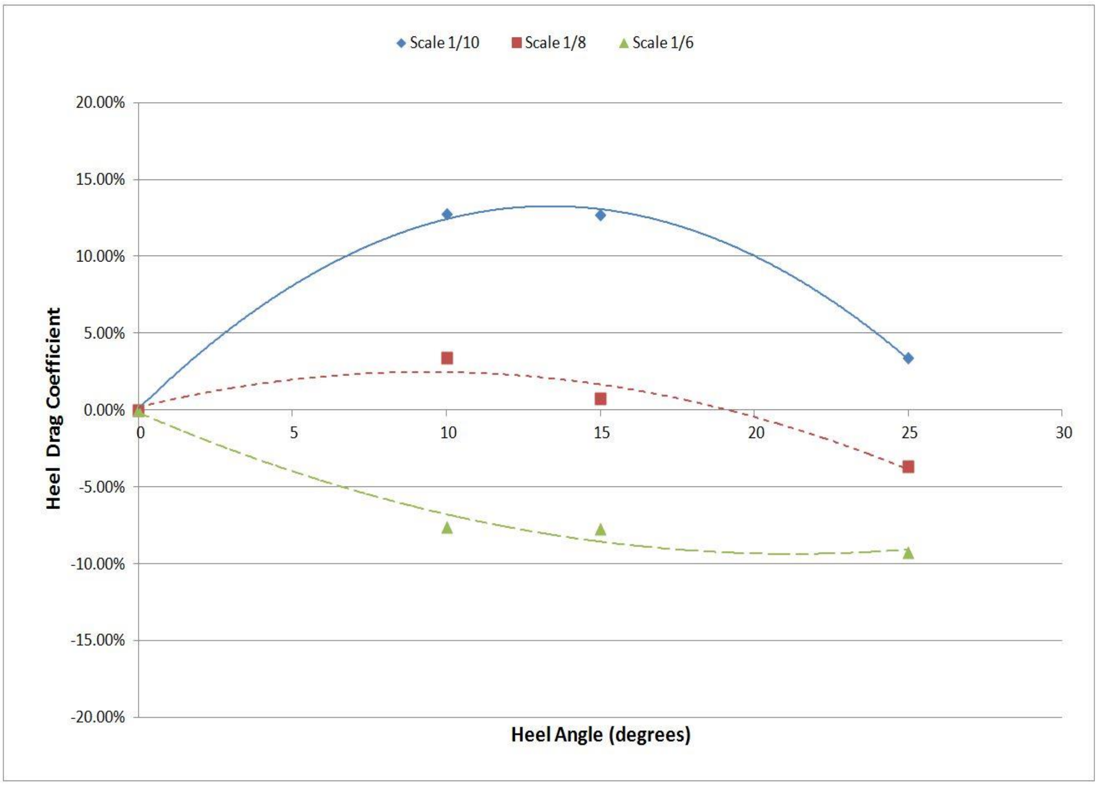

Figure 3.31 : Heel drag coefficient variation over heel angle.

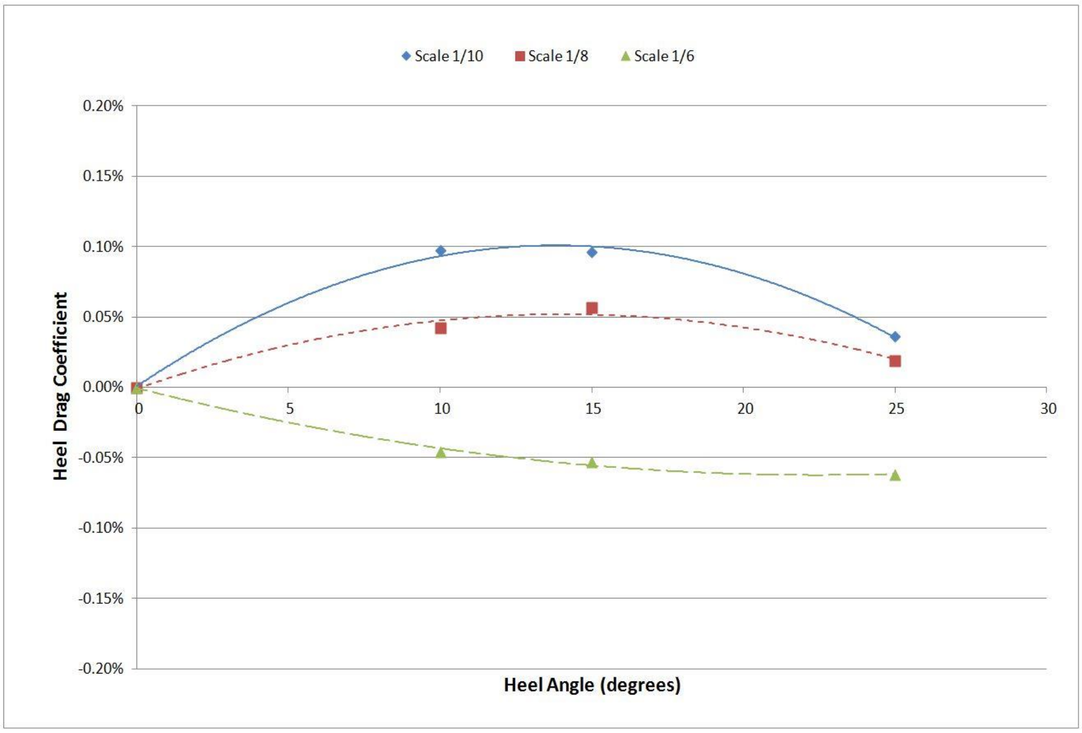

Figure 3.32 : Heel drag coefficient variation over heel angle.

3.4.2 Tests with heel angle and leeway angle

Following tests with heel but no leeway, tests with various heel angles and leeway angles were conducted for a speed range of 8 knots to 16 knots of full scale boat speed. Heel angles were varied to 0, 10, 15 and 25 degrees, meanwhile leeway angles of 0, 2, 4 and 6 were used. Both resistance and side force were measured for 1/10, 1/8, and 1/6 scale models. Side force increases with leeway angle for all the scales. Induced drag in case of lifting body can be expressed in relation to the square of side force generated.

$$C_{DI} = \alpha {C_L}^2 \tag{3.23}$$

The slope (α) is the prime feature of the resistance-side force relationship. Traditionally the slope of C_L^2-C_T plot is utilized commonly in the performance estimation as it is the sole indication of lift generation efficiency. Also, the linear relationship between the two parameters makes it more convenient for comparision purposes. Figure 3.33 demonstrates the effect of scale for a heel angle of 15 degrees at 8 knots of full scale boat speed. The slope decreases with the model size.

Change of slope with model scale change for 10, 15 and 25 degrees of heel is given in Figure 3.34, 3.35 and 3.36 respectively.

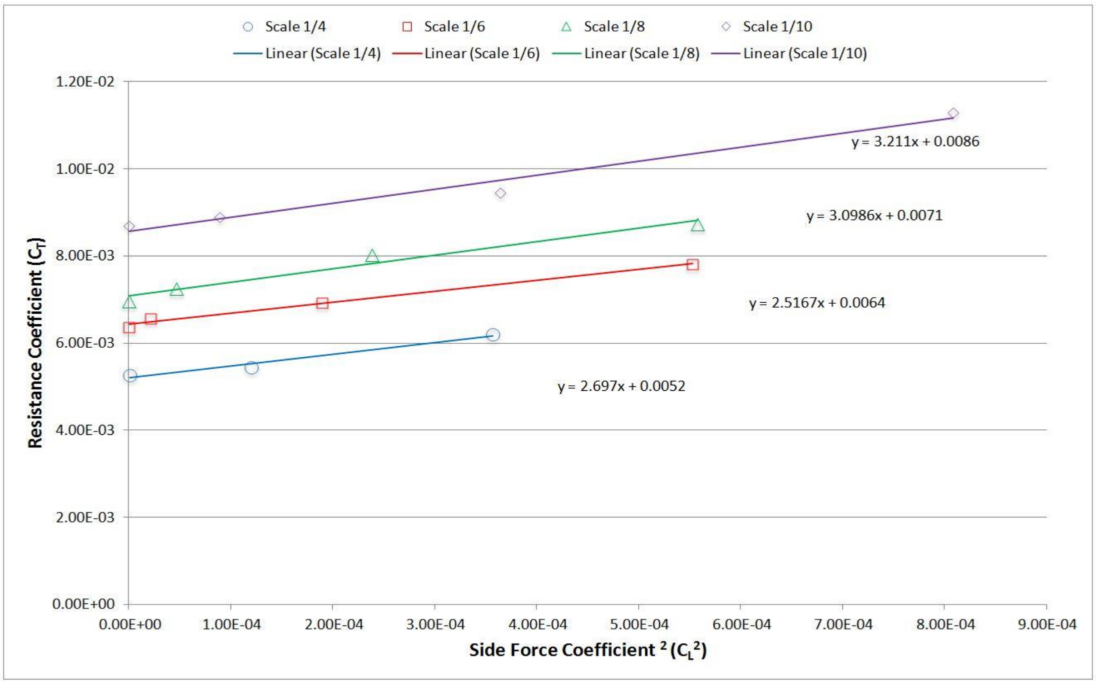

Figure 3.33 : Side force- resistance relationship for 15 degrees of heel at 8 knots of full scale boat speed.

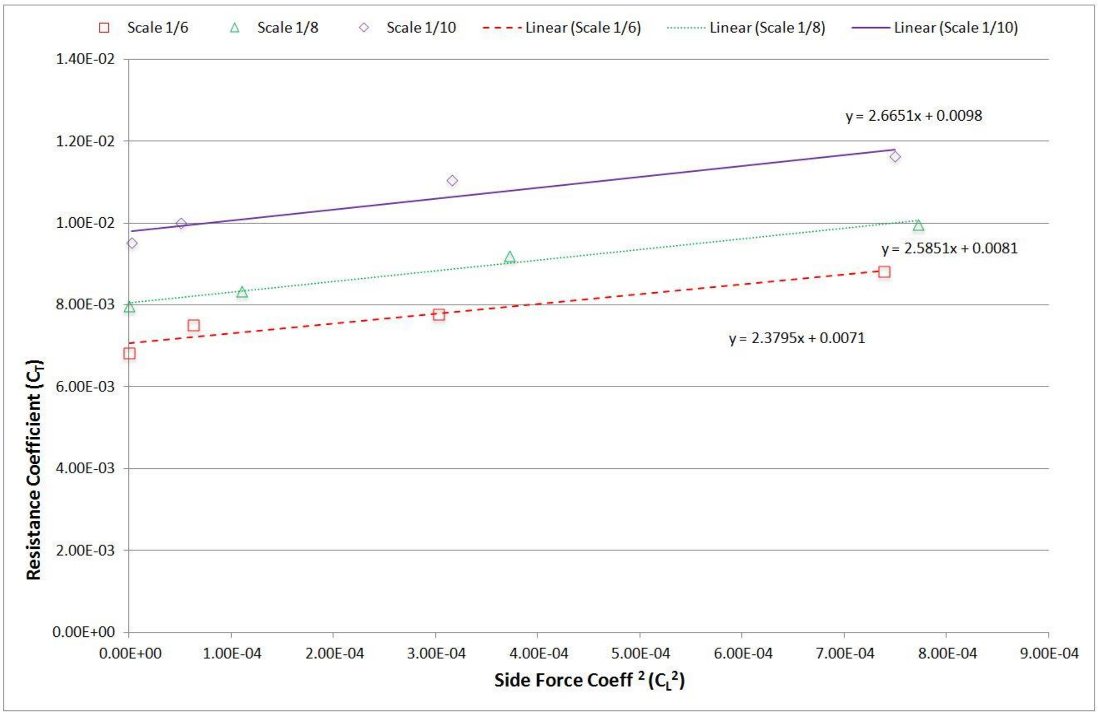

Figure 3.34 : Side force- resistance relationship for 10 degrees of heel at 9 knots of full scale boat speed.

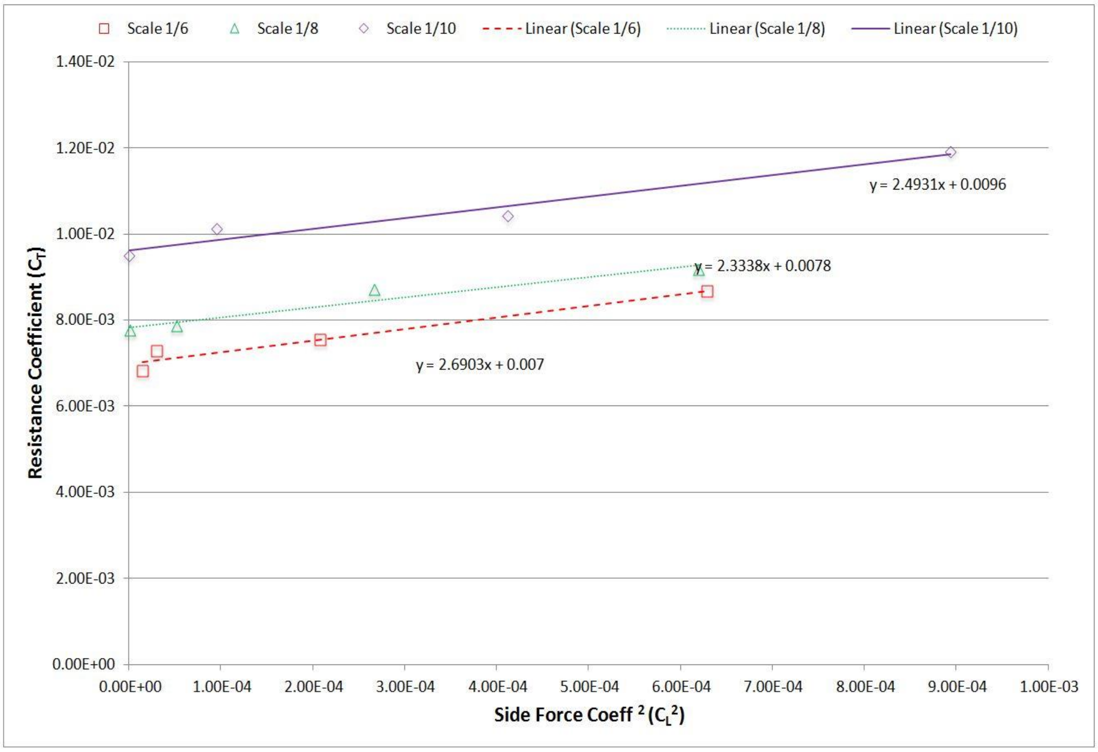

Figure 3.35 : Side force- resistance relationship for 15 degrees of heel at 9 knots of full scale boat speed.

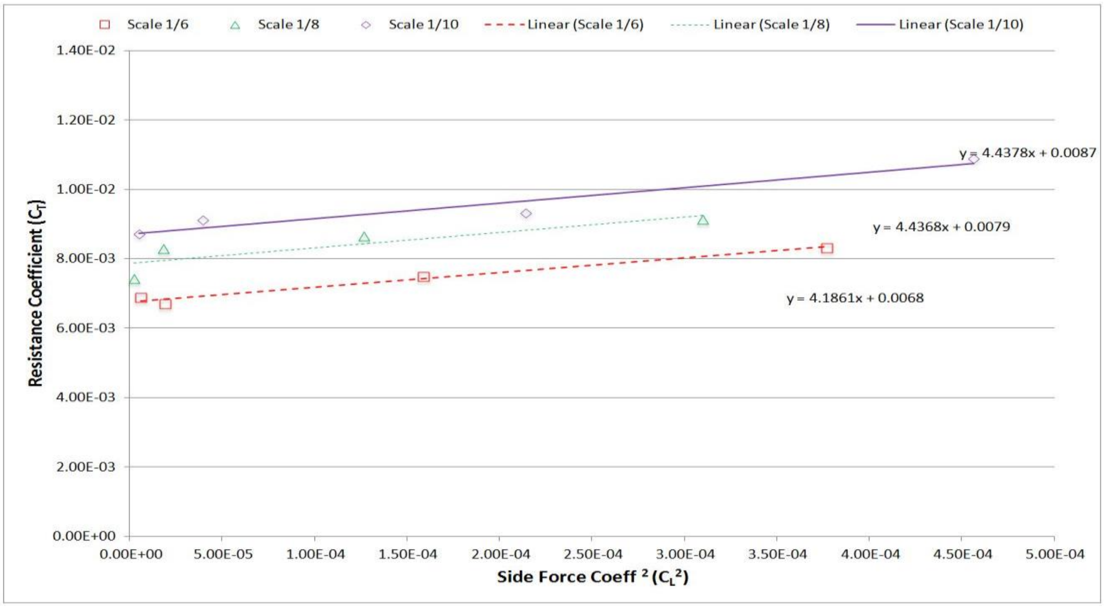

Figure 3.36 : Side force- resistance relationship for 25 degrees of heel at 9 knots of full scale boat speed.

The trends in the slope variation may better be seen when C_L^2-C_T curve slope variation is plotted against scale. The C_L^2-C_T curve slope variation for 1/10, 1/8, 1/6 model scale, 5, 7, 9, 11 knots of full scale boat speed for 10 degrees of heel angle is given in Figure 3.37. There is a variation of slope by model scale and speed, in fact both leading to change in Reynolds number. As the model size decreases, hence for lower Reynolds numbers, slope increases. This may be interpreted as lower side force generation for equivalent induced resistance for the smaller models.

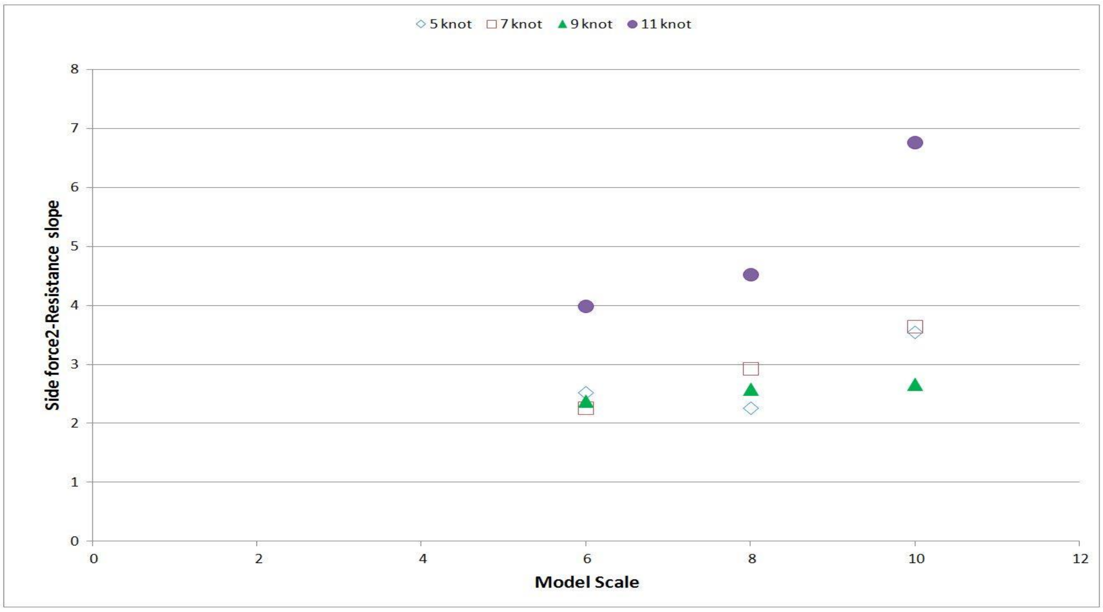

Figure 3.37 : $C_L{}^2$-C_T curve slope change with model size (10 degrees of heel).

3.4.3 Discussions on the results of heeled tests

The scale error in the $C_L{}^2$-C_T curve slope can be assessed by comparing different scale model slopes. Figure 3.38 demonstrates the error due to model scale. When percentage error in slope is plotted against model length, it is evident that the model scale and accuracy relationship obtained from the heeled tests are similar to the findings in the upright tests. In this plot, the error has been calculated by plotting the ratios between the slopes at each scale and the average slope. As the actual lift generation characteristics at full scale could not be obtained by testing, the average slope was used for error calculations rather than the actual slope.

The scale error in $C_L{}^2$-C_T versus model length curve is a good indication of cost versus accuracy for testing campaigns. The error in the slope for the 1/10 scale model is as high as 40 % of the slope value, decreasing down to less than 10 % for a model scale of 1/6, i.e. 2.6 m model. At 1/4 scale, the error is less than 5%. When the curve is analyzed, it is evident that model sizes less than 4 meters are not suitable for appendage development as the differences in investigated parameters between alternatives are most possibly lower than the expected error. It can also be seen that there is good accuracy above 4 meters, but relative merit decreases with increase in model size. There is dramatic increase in model size and hence cost per 1% increase in accuracy. Therefore, model size decision has to be made by the experimenter and the designer with a good knowledge of expectations from the towing tank and the level of accuracy required for the specific task associated with the testing campaign.

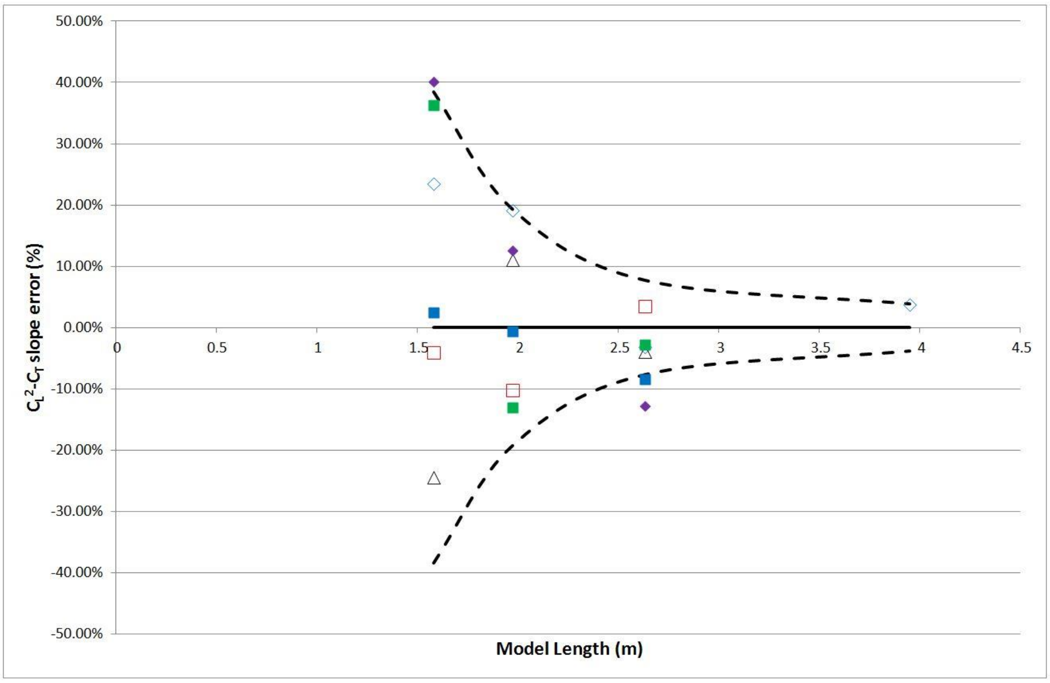

Figure 3.38 : C_L^2-C_T curve slope change error with model length.

4. CFD ANALYSIS

The aim of carrying out CFD analysis is to extend the scope of experimental investigations by making use of certain advantages in CFD. Ease of obtaining certain features of the flow field like streamlines, wave profiles, velocity and pressure contours without disturbing the flow field is one of the fundamental differences between experimental and computational approaches. Also, ease of varying the geometry -different designs or in this case different scales of the same design- is a crucial advantage and time saver compared to experiments. However, the analyst should be aware of the limitations of the computational methodology being utilized, and the correlation between the expectations from the analysis and the capabilities of the method to fulfill these expectations. In this case, the expectation is to aim reproducing sensible results within reasonable accuracy and most importantly to capture similar trends in variation of investigated parameters such as wave profile, viscous drag, wave making drag, form factor, side force etc… Also, the method enables investigations at larger scales that may not be tested in a towing tank.

4.1 CFD Methodology

The utilized methodology makes use of the CFD as a supplementary tool to experiments. The main interest is to assess if the observed trends in the variation of the investigated parameters are compliant with the findings of the experiments. This would enable the statement that the findings from the experiments may well be captured with CFD. The ease of processing the flow field parameters without actually disturbing the flow field in CFD would enable further investigations to supplement the experimental findings. Also, CFD tool would enable investigations on a broader scale range, possibly upto full scale.

4.1.1 The method: RANS

CFD is the use of numerical techniques to solve the equations defining fluid flow around, within, and between bodies. The equations solved are numerical approximations to mathematical models describing the physics of the actual fluid flow.

The utilized RANS approach in CFD makes use of the finite volume method in which a finite solution domain is discretized into finite set of cells. General continuity and conservation of momentum's partial differential equations are discretized into algebraic equations. With the aid of numerical methods, an iterative procedure allows the solution of these algebraic equations. Below are the general forms of the continuity and conservation equations known as the Navier Stokes equations:

$$\frac{\delta\rho}{\delta t} + \nabla(\rho V) = 0 \qquad \textbf{(4.1)}$$

$$\frac{\delta(\rho U)}{\delta t} + \nabla(\rho U \boldsymbol{V}) = -\frac{\delta P}{\delta x} + \nabla.(\mu\nabla U) + S_U \qquad \textbf{(4.2)}$$

Direct solution of these equations are only possible for specific applications at low Reynolds numbers. For complex geometries like sailing yachts with appendages and flows with complex physics like turbulent, unsteady, multiphase flows, modifications are necessary to Navier Stokes equations.

Various methods are available for solving turbulent flows. Direct Numerical Simulation (DNS), a research tool rather than an industrially accepted procedure is the solution of time dependent Navier Stokes equations where all the eddies in the turbulent flow field are captured within the solution domain. This results in enormous number of cells and is not feasible for industrial applications. It is being used as a research tool for very simple geometries and low Reynolds numbers.

In Large Eddie Scales (LES) approach, large eddies are resolved in the flow field whereas small eddies are modelled. This approach is also very demanding in terms of computational power and is yet not feasible for use in the industry.

In Reynolds averaging, transport equations are solved for the mean quantities of flow and all scales of turbulence are modelled. The unsteady flow due to turbulence is resolved by replacing instantaneous values by mean and fluctuating components and the resulting equations are time averaged. The resultant equations are Reynolds averaged Navier Stokes equations or RANS equations as below:

$$\frac{\delta(\rho U)}{\delta t}+\nabla(\rho U\boldsymbol{V})=-\frac{\delta P}{\delta x}+\nabla.(\mu\nabla U)-\frac{\delta\left(\rho\overline{u'^2}\right)}{\delta x}-\frac{\delta(\rho\overline{u'v'})}{\delta y}+\frac{\delta(\rho\overline{u'w'})}{\delta z}+S_u$$

$$\frac{\delta(\rho V)}{\delta t}+\nabla(\rho V\boldsymbol{V})=-\frac{\delta P}{\delta y}+\nabla.(\mu\nabla V)-\frac{\delta(\rho\overline{u'v'})}{\delta x}-\frac{\delta\left(\rho\overline{v^2}\right)}{\delta y}+\frac{\delta(\rho\overline{v'w'})}{\delta z}+S_u \tag{4.3}$$

$$\frac{\delta(\rho W)}{\delta t}+\nabla(\rho W\boldsymbol{V})=-\frac{\delta P}{\delta z}+\nabla.(\mu\nabla W)-\frac{\delta(\rho\overline{u'w'})}{\delta x}-\frac{\delta(\rho\overline{v'w'})}{\delta y}+\frac{\delta\left(\rho\overline{w'^2}\right)}{\delta z}+S_u$$

The added terms due to Reynolds averaging (averaged products of fluctuating velocities) represent the transfer of momentum due to turbulent fluctuations. These terms are named as Reynolds stresses. Due to the addition of these terms, the Reynolds averaged Navier Stokes equations can not be closed. In order to aciheve a solution, turbulence models are required to relate these Reynolds stresses to the mean velocity field. Various turbulence models have been developed from experimental data obtained for fluctuating velocities in different flows.

Figure 4.1 is useful in understanding the extra effort associated with resolving eddies rather than modelling.

The feasibility of LES and DNS methods is not sufficient to use in industrial applications and with turbulence models satisfactory levels of accuracy may be achieved depending on the expectations (Fluent INC, 2005).

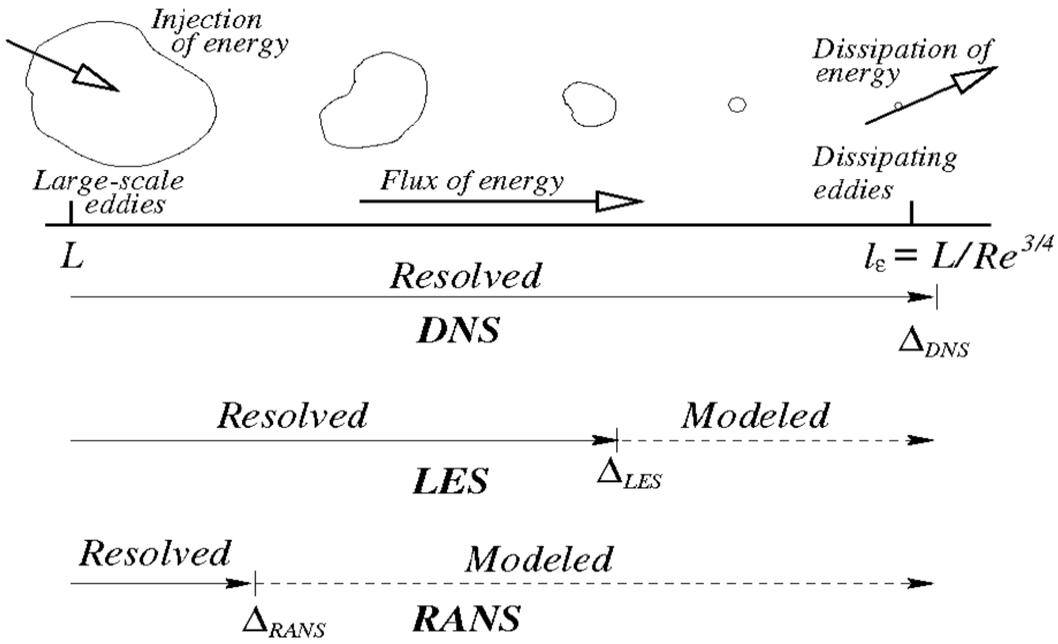

Figure 4.1 : Details of solving for eddies in different approaches (Fluent INC, 2003).

4.1.2 The solver: ANSYS CFX

Ansys CFX solver has certain advantages amongst others when high-speed hydrodynamics is the area of investigation. This software has several options for multiphase modeling that enables a better representation of the free surface interphase at high speeds.

In free surface flows, the fluids are assumed to be mixed at a macroscopic level with a distinctive interface between them (Ansys, 2010).

Two options are available in Ansys CFX for multiphase modeling. In homogenous flows, all fluids share the same velocity field. A single equation is solved for each momentum component.

In non-homogenous flows, fluids have separate velocity and pressure fields; separate equations are solved for momentum components of each phase. As this method is time consuming and less stable compared to homogenous flow option, it performs best at high speed cases as it does not enable unphysical mixing of phases (air entrapment) under the hull at large trim angles.

4.1.3 Determination of modelling and analysis strategies

For adjusting the mesh strategies and solution parameters to be utilized for the experimental validation and further analysis; a study needs to be conducted. The objective of the study was to assess the the effect of user inputs on the results obtained and to develop means in order to adjust these parameters for achieving a valid solution with minimum computer sources and CPU time. During this study, ITTC's recommended CFD procedures were used as guidelines. The main procedures followed were procedure 7.5-03-01-01 "CFD General, Uncertainty Analysis in CFD, Verification and Validation Methodology and Procedures" and 7.5-03-01-04 "CFD, General CFD Verification".

CFD Verification methodology of ITTC has been adopted from American Society of Mechanical Engineers' (ASME) CFD guidelines. The main areas of interest in the verification analysis are to investigate the effects of iterative convergence, spatial and temporal discretization convergence, order of accuracy and grid dependence.

Verification, in conjunction with validation are the principal means to evaluate accuracy and reliability in computational simulations. Society of Computer Simulation defines model verification as "substantiation that a computerized model represents a conceptual model within specified limits of accuracy" (Oberkampf, 2002). This implies that the computerized model, (the code) should mimic the conceptualized model accurately. The slightly modified AIAA definition of verification is "the process of determining that a model implementation accurately represents the developer's conceptual description of the model and the solution of the model". They define validation as" the process of determiming the degree to which a model is an accurate representation of the real world from the perspective of the intended uses of the model" (Oberkampf, 2002).

As seen from the definitions and areas of interest, verification is a process that is largely related within the development phase of the code in consideration. As pointed out by IEEE, verification is "the process of evaluating the products of a software development phase to provide assurance that they meet the requirements defined for them by the previous phase" (Oberkampf, 2002). Basic objective in verification studies is to provide proof that the conceptual model (the continuum mathematics) is solved correctly by the discrete mathematics computer code. It is an indication of how well the numerical approach adopted within the code represents the concepts behind it. It also gives an indication of the accuracy achieved within the code with the objective of identifying and quantifying the errors in the computational model and its solution.

Normally, during verification activities, accuracy of the solution to a computational model is assessed with the aid of known solutions. These have to be highly accurate benchmark solutions like analytical or very accurate numerical solutions (Oberkampf, 2002). Limited availibility of these solutions and their simplicity in terms of geometry and physics involved are the major downsides in code verification.

AIAA, ASME and ITTC procedures for verificaiton studies are based on the work of Roache who states that verification is a mathematics and computer science issue; not a physical issue (Oberkampf, 2002). These activities are principally performed during the development phase of a computational code.

The major part of the verification study associated with the solution part is the grid independence requirement. Therefore, this part of the verification will be of greatest interest.

ASME Guideline 4 suggests that solution achieved is grid independent. In order to conform to this requirement, all the important grid parameters that influence the solution are recommended to be assessed independently (ITTC, 1999). Some of these parameters are stated as the grid type (C-Grid, O-Grid...), number of grid points, clustering near walls, leading edges and trailing edges; location of inlet and outlet boundaries, minimum spacing requirements for turbulence models according to the procedure. To add to the list stated in the procedure mesh density in the wake regions, free surface grid spacing (in x, y and z directions), mesh density in the vicinity of the hull and appendages may be listed.

The grid independence analysis approach was adopted peculiar for sailing yacht analysis. In this approach, the influence of the mesh sizes to the drag & lift components and wave profiles are aimed to be assessed separately. Therefore, for the viscous drag component, the effect of wall treatment is assessed by grid doubling approach in near wall regions. For different size yachts, achieving Y+ values in the range recommended by the software vendors may become impracticable and maybe inapplicable. The adopted approach starts with a mesh size near walls with same size cells as used at the free surface and the grid size for the hull surface and appendages should be halved until the drag values become constant. The number of cases to be investigated will not be fixed since for different size vessels, the necessitices will be different. This will be considered on a case by case basis with engineering judgement.

After obtaining the right wall treatment, the mesh density in the hull vicinities, appendage leading edges, wake regions and free surface should be refined.

4.2 Upright Analysis

It has been investigated in the experimental analyses that the wave resistance component obtained during the experiments was constant in coefficient form for scales 1/8, 1/6 and 1/4 in the naked condition. However, during the upright tests with the appended configuration, it has been found out that, with the current resistance breakdown methodology, an agreement between the wave resistance components could not be obtained among the tested models. The wave pattern resistance analyses showed that this component of resistance is identical among the tested models and it is not affected by the existance of the appendages. As the actual wave resistance characteristics should be identical amongs the tested models, the differences in the wave resistance values obtained during experiments are assumed to be originating from non-wave related sources and these are being included in the wave resistance component owing to the utilized resistance breakdown methodology. As the sources of the excess in the obtained wave resistance values could not be identified by force measurement during experiments, it was decided to investigate the possible sources of the mentioned variation with CFD tools. The CFD study was programmed in a systematic manner to be able to isolate the effect of possible individual sources leading to the variations. As a result, the expected appendage effects were investigated by adding each component (keel, bulb, rudder, hull) to the computational model step by step. Therefore, it has been decided to follow the below sequence for CFD investigations:

- 2-D keel,
- Keel & Bulb combination,
- Keel & Bulb & hull & rudder combination (underwater geometry only),

- Fully appended geometry in multiphase condition,
- Naked geometry in multiphase condition.

The multiphase analyses in the naked and appended conditions were conducted on two different scales, i.e. 1/8 and 1/4 in order to enable the assessment of scalability of the hull resistance components.

4.2.1 Variation of appendage resistance

4.2.1.1 2-D keel drag

The study has been initiated with the investigation of appendage resistance. Initially, the compliance with the empirical formulation utilized during the experiments has been of major interest. Therefore, a study on the keel has been conducted individually.

The study has been conducted for the keel geometry at 1/8 scale at Froude number 0.5. The keel geometry has been placed in a solution domain that terminates at the keel root and tip in the vertical direction in order to achieve a two dimensional flow. The domain may be seen in Figure 4.2.

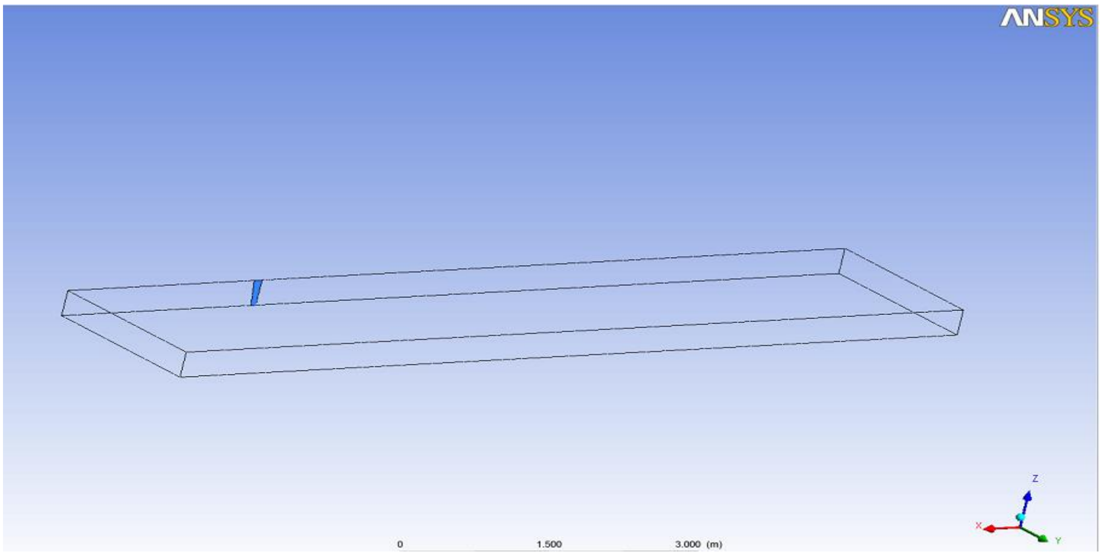

Figure 4.2 : Solution domain utilized for 2-D keel computations.

A symmetry condition has been utilized at the keel center plane in order to reduce computational effort. The 2-D keel analysis has been conducted at Froude number 0.5 (2.2008 m/s) in water. The mesh in way of keel and the boundary layer may be seen in Figure 4.3. Initial grid independence analyses indicated the need to resolve the wake of the foil as the analyses without wake resolving tended to overpredict the normal resistance.

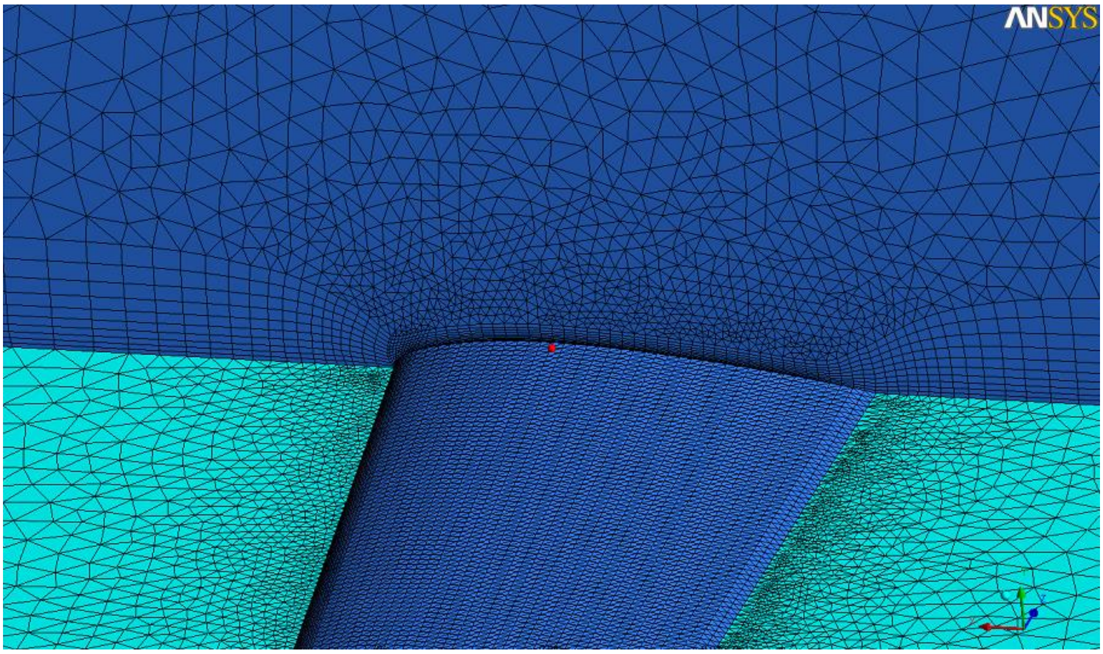

Figure 4.3 : Mesh refinement in way of keel.

The normal resistance obtained from the computations that corresponds to the form resistance calculated empirically for the whole keel is 0.0320 kilograms. The empirically calculated resistance is 0.0308 kilograms. In figure 4.4, the force convergence history for the half body model may be seen.

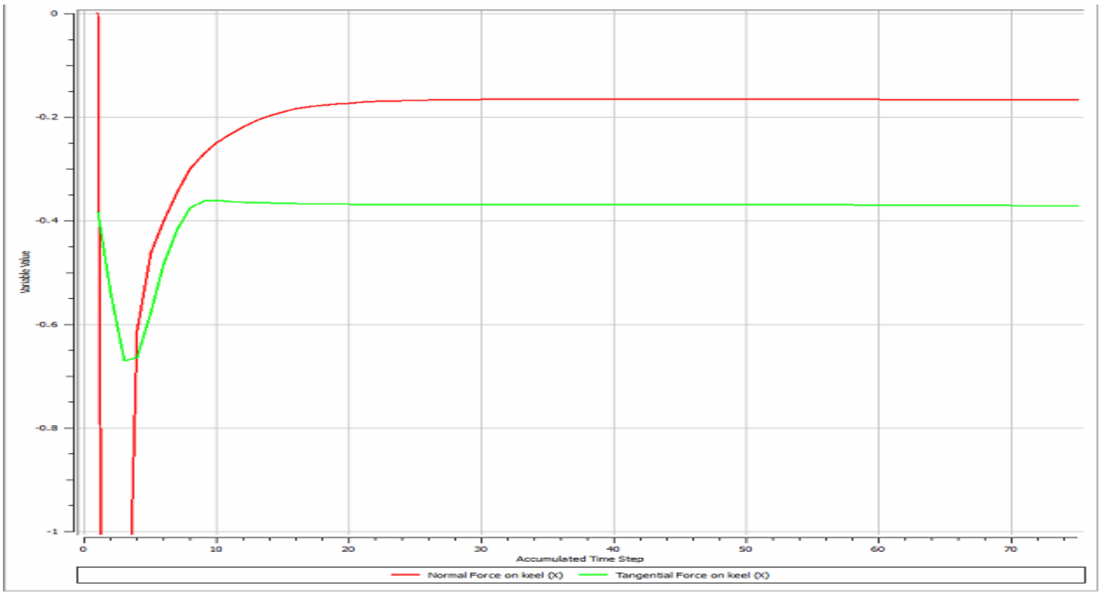

Figure 4.4 : Force convergence history for the keel computation.

The pressure coeeficient plot for the keel may be seen in Figure 4.5. This plot will be utilized to compare the pressure differences as the keel operates in 2-D condition, with bulb connected, with bulb connected & under the hull (underwater only), and with bulb connected & under the hull with free surface effect consecutively.

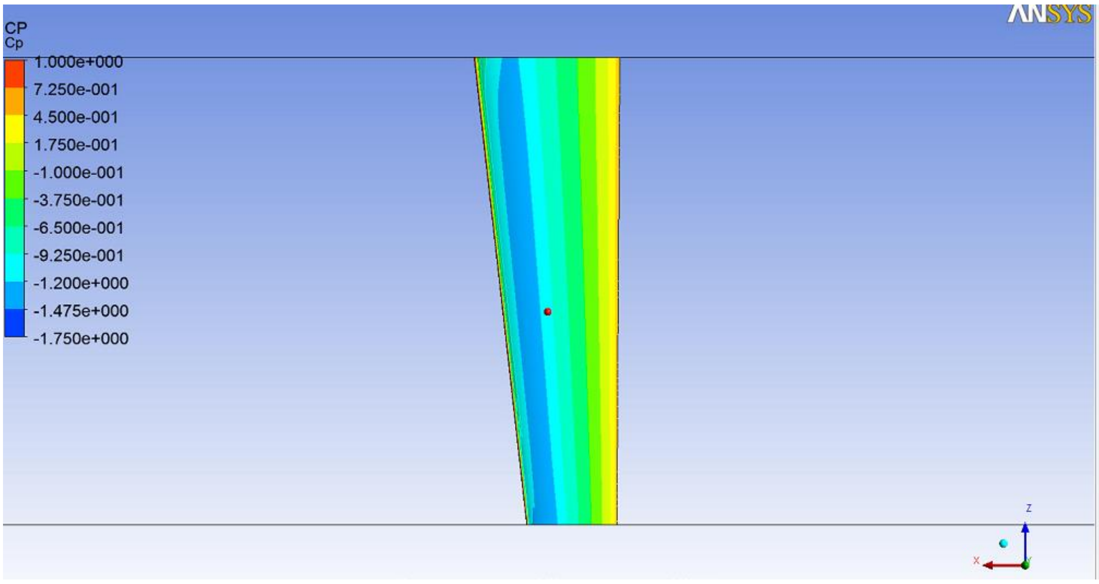

Figure 4.5 : Pressure coefficient in way of keel.

It is seen from the pressure coefficient plot that the pressure variation is almost uniform along the span, in accordance with the sweep angle of the keel. This implies that 2 dimensionality is present in the flow around the keel. However, there are minor variations present at the root and the tip.

4.2.1.2 Keel with bulb connected

The next step during the computational phase after the 2-D keel investigation was to assess the influence of the bulb connection on the normal resistance of the keel. The solution domain has been utilized in order to apply symmetry condition at the center plane and at the top (keel root). The domain may be seen in Figure 4.6.

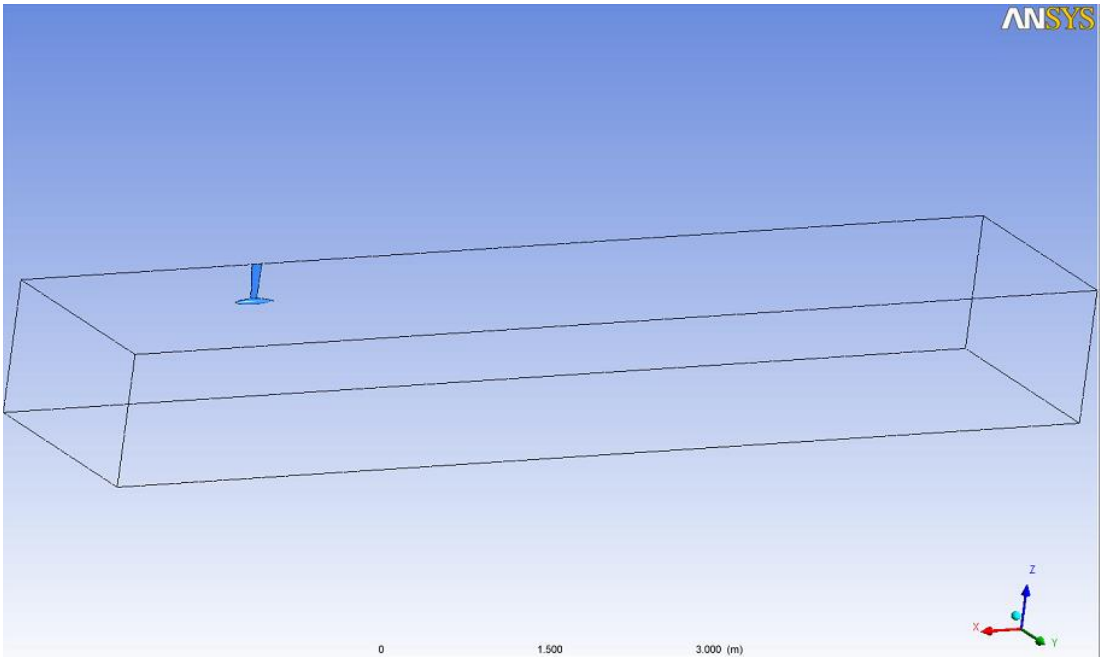

Figure 4.6 : Solution domain for keel-bulb combination analysis.

The meshing strategy is similar to the previous analysis; refinement has been made in both trailing and leading edges of the bulb whereas for the keel, only leading edge refinement has been made. The mesh in way of keel and bulb may be seen in Figure 4.7.

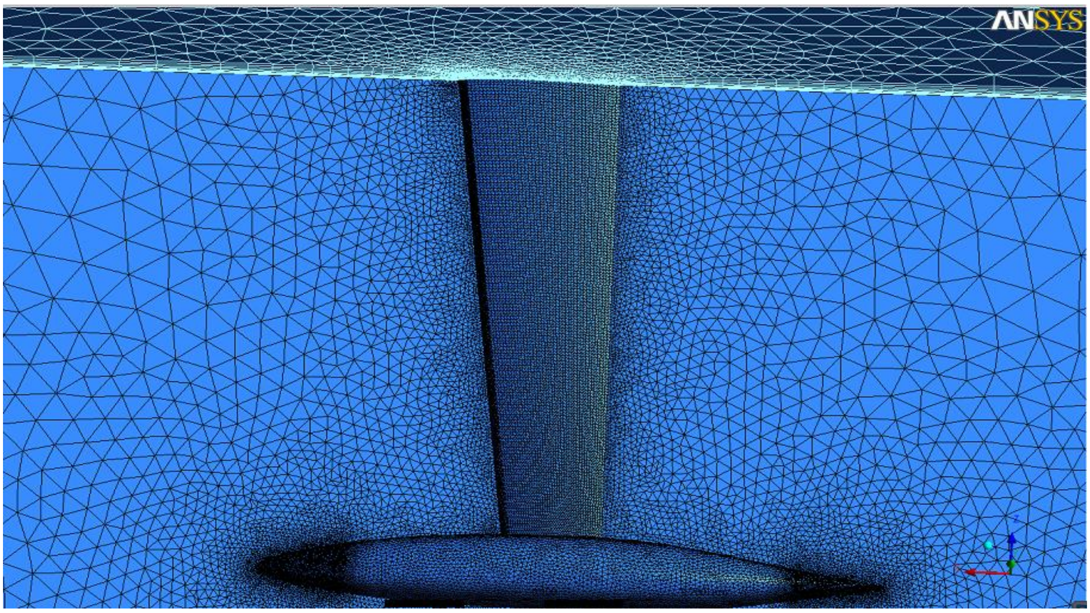

Figure 4.7 : Mesh in way of keel and bulb.

In the analysis, when the bulb is connected to the tip of the keel, it has been found out that the normal resistance of the keel increases to 0.0340 kilograms from 0.0320 kilograms. The increase in the normal resistance is at the order of 6%.

The resistance of the bulb has been found to be identical to the computations with only the bulb geometry.

From the pressure coefficient curve in Figure 4.8, it is seen that there is change in the pressure coefficient distribuiton at the keel tip, the area with the highest pressure loss has been enlarged compared to the pressure distribution in the 2-D case that may be seen in Figure 4.5. With the attachment of the bulb at the tip of the keel, a 3 dimensionality is introduced to the flow.

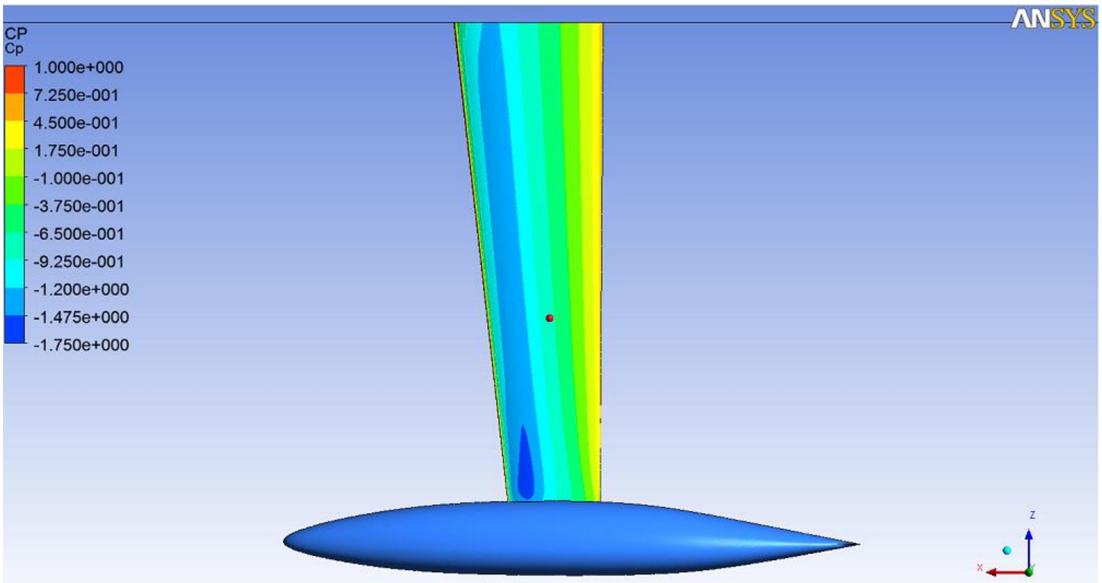

Figure 4.8 : Pressure coefficient in way of keel.

4.2.1.3 Keel & bulb & hull combination (underwater)

The next step in the computations after the keel and bulb combination was to assess the influence of trim and the hull connection on the keel and bulb resistance. The underwater body of the hull was added to the current model and the whole model was trimmed to 2 degrees. The solution domain may be seen in Figure 4.9.

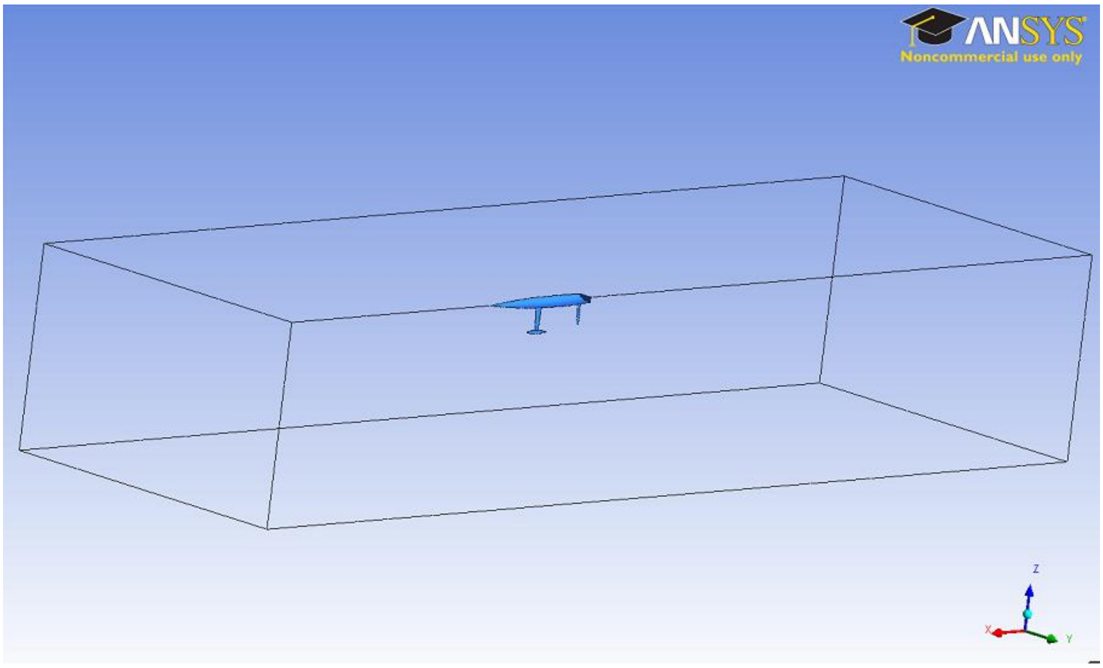

Figure 4.9 : Solution domain for keel & bulb & rudder & hull combination (underwater).

In this phase of the computations, the influence of the existence of the hull and the induced trim on the resistance of the appendages was investigated. The analysis was conducted at Froude number 0.5 as in previous computations. The hull, keel, bulb and rudder geometry was trimmed to 2 degrees. Mesh modelling strategy was similar to previous cases as it may be seen in Figures 4.10 and 4.11.

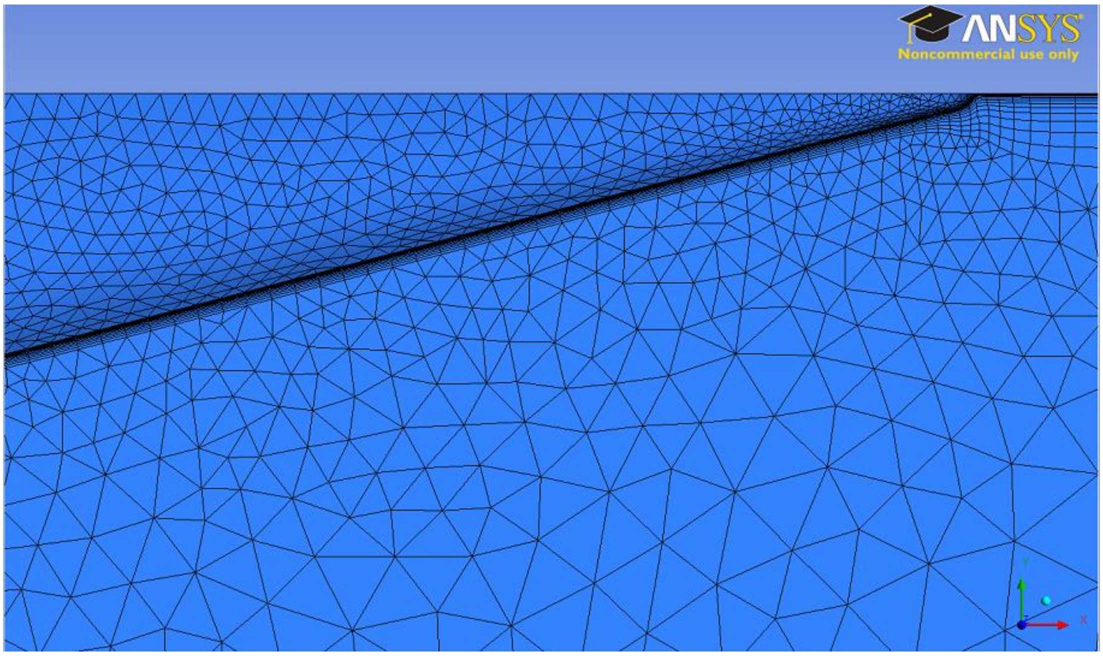

Figure 4.10 : Mesh refinement & boundary layer modelling in way of hull entry.

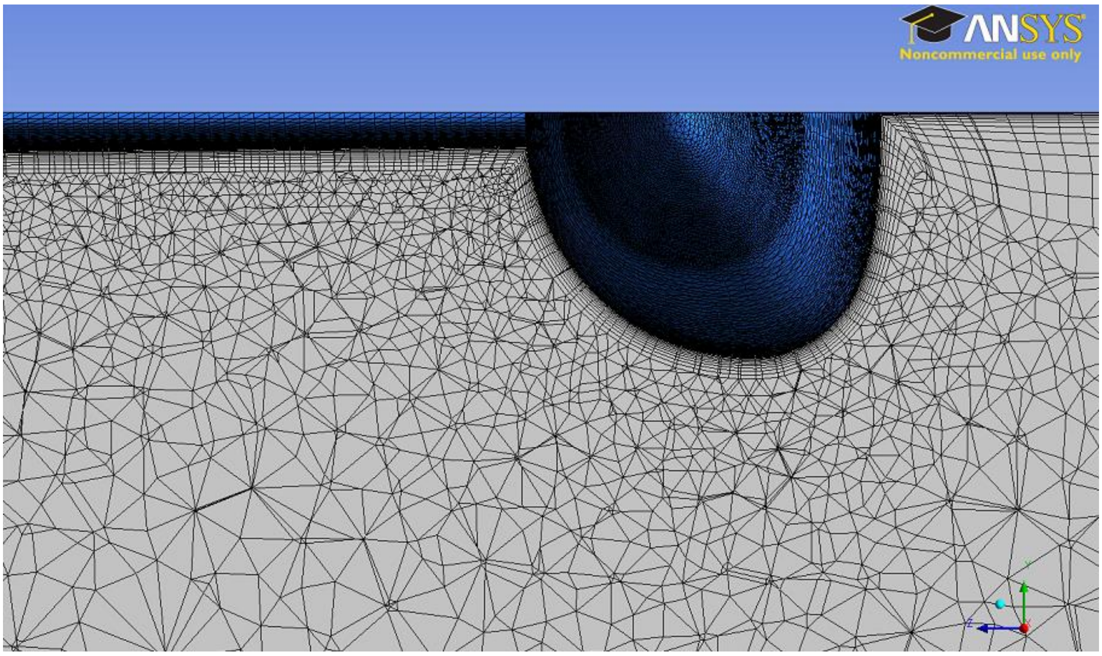

Figure 4.11 : Mesh refinement & boundary layer modelling in way of keel & bulb connection.

As the hull is connected to the root of the keel at 2 degrees of trim, the normal resistance of the keel increased from 0.0340 kilograms to 0.0381 kilograms. The increase is at the order of 11% compared to the previous keel and bulb combination case and 19% compared to the keel only case.

The bulb normal resistance in this case has been calculated as 0.0128 kilograms for the whole body of the bulb. The keel& bulb combination case, the bulb resistance has been calculated as 0.050 kilograms. The increase is more than 150% indicating that the bulb resistance is more vulnerable to trim changes compared to other appendages.

The change in the pressure coefficient plot is similar to the keel & bulb combination case; the downward suction of the highest pressure drop is more pronounced in this case. The 3 dimensionality of the flow is more apparent in this case compared to previous investigations.

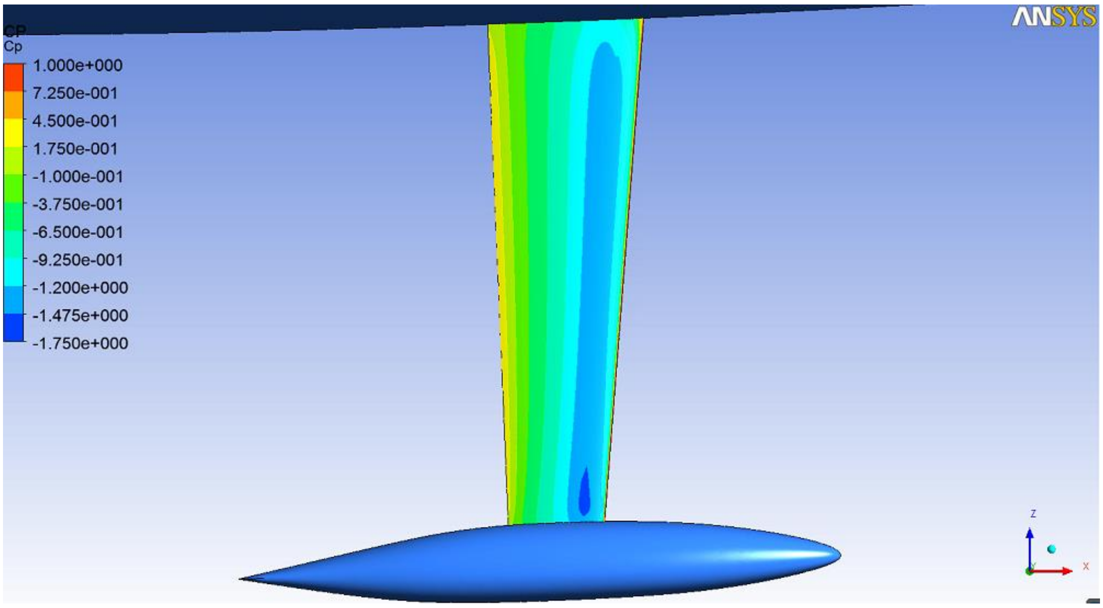

Figure 4.12 : Pressure coefficient in way of keel.

4.2.2 Variation of hull resistance components with scale

The fully appended and naked computations in multiphase condition would enable the assessment of scalability of the resistance components as the analyses would be conducted at different scales; i.e. scale 1/8 and scale 1/4. Computations were carried out at Froude number of 0.5 at the mentioned scales.

4.2.2.1 1/8 Scale fully appended geometry in multiphase condition

The fully apppended geometry was placed in the solution domain as seen in Figure 4.13.

The meshing strategy may be seen in Figure 4.14. A total of more than 3.000.000 cells were used. The mesh has been refined in the vertical direction in way of the expected free surface. Boundary layer modelling is similar to previous case in way of hull and appendages. As it has not been possible to include the refined mesh in way of the wake of appendages, any variation in the normal resistance of the appendages with respect to the case in section 4.2.1.3 will not be considered in this part of the analysis..

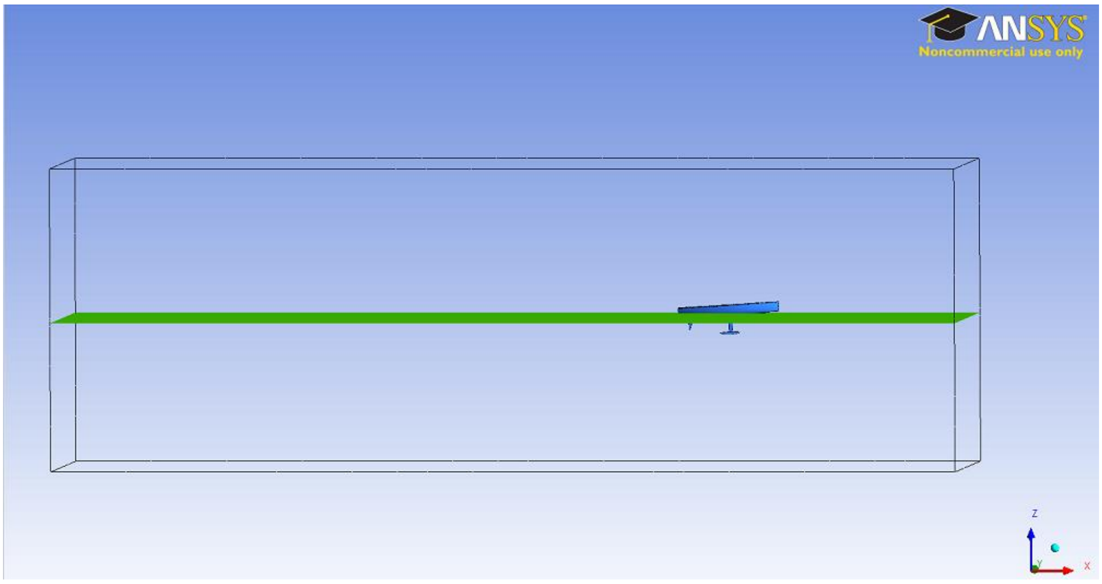

Figure 4.13 : Multiphase solution domain with the fully appended geometry.

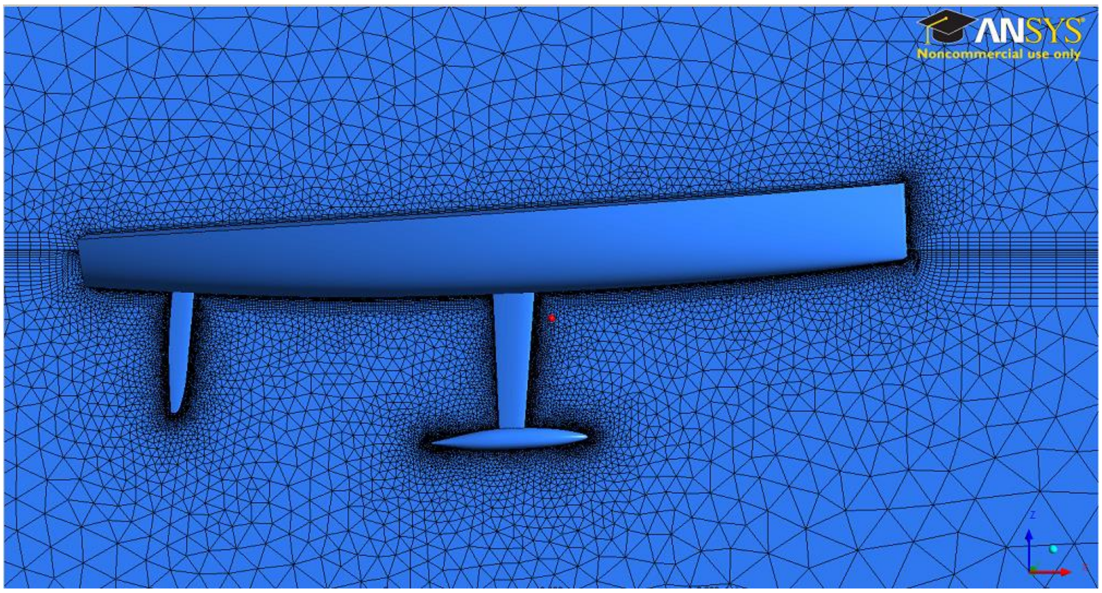

Figure 4.14 : Mesh in way of free surface.

The analysis was conducted with 2 degrees of trim and on a draft with the consideration of sinkage values obtained from experiments. Homogenous multiphase modelling was used with k-e turbulence modelling accounting for the curvature of the flow.

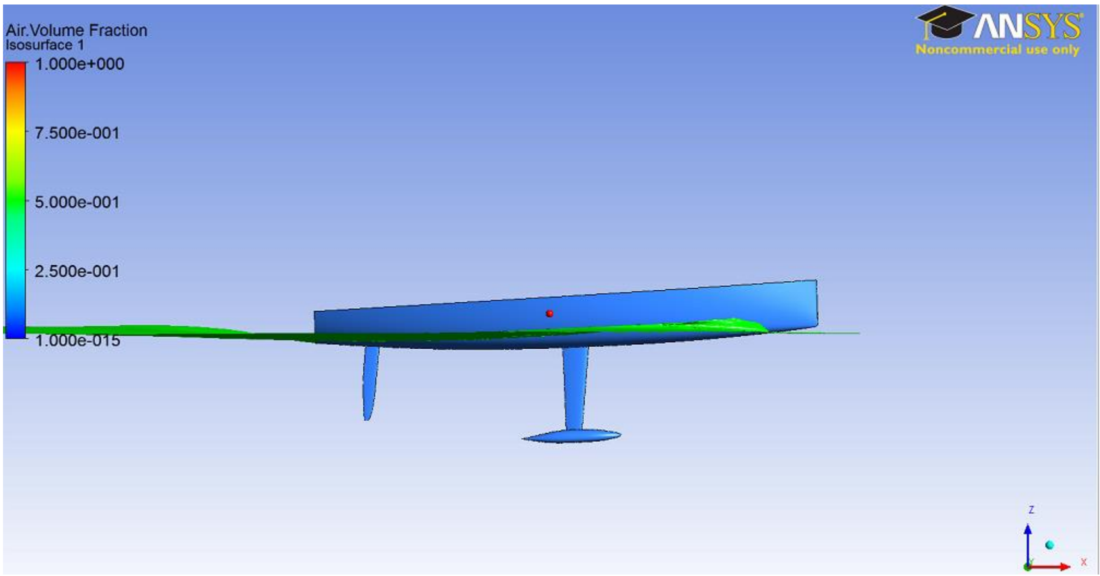

Figure 4.15 : Wave profile at Froude number of 0.5 for 1/8 scale.

The generated wave profile may be seen in Figure 4.15. It is seen that the bow and transom waves are substantial at this Froude number as the sinkage is at its maximum across the Froude number range. In Figure 4.16, the wave formation may be seen from perspective view.

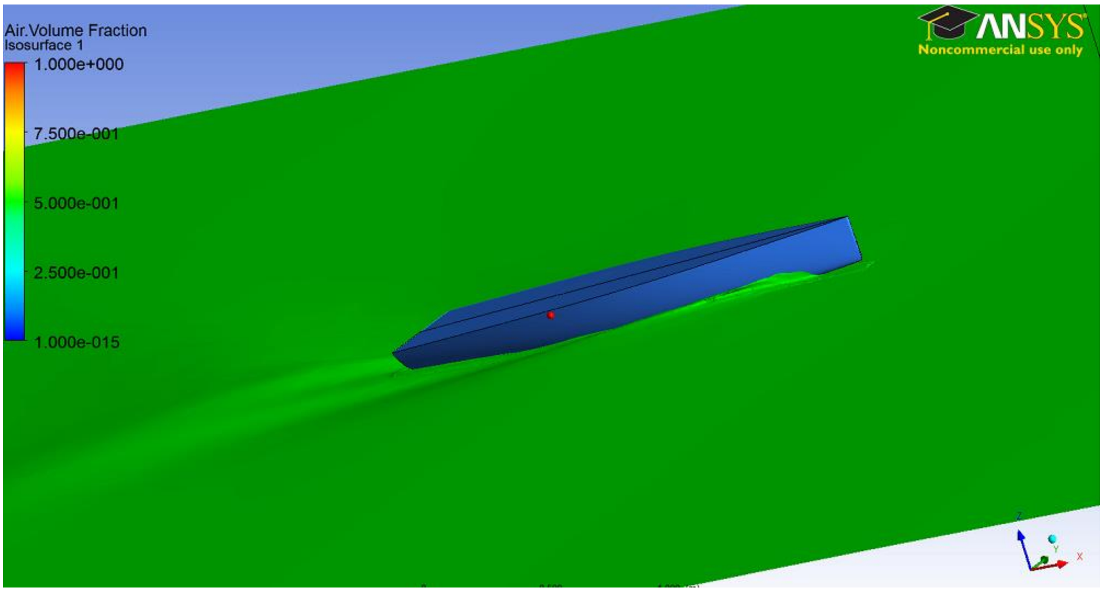

Figure 4.16 : Wave formation at Froude number of 0.5 for 1/8 scale.

The utilized CFD code enables the breakdown of forces in such a manner that the tangential and normal forces on a wall may be monitored during the computations. This corresponds to the methodology of resistance breakdown as frictional and residuary resistance components. In addition, during the post processing of the results, the total forces with respect to different fluids may be extracted. Therefore, the total friction drag and residuary drag as well as the total air drag and total water drag may be extracted from the computations.

The force extraction drawback makes it impossible to extract the actual wave resistance of the hulls from the computations. Therefore, some assumptions have to be made. In the post processing of the results, the total air drag has been deducted from the total computed drag to obtain the the total water drag.

The total water drag has been computed as 1.2389 kilograms for the whole body, comprising of a frictional drag of 0.5995 kilograms and a residuary drag of 0.5294 kilograms.

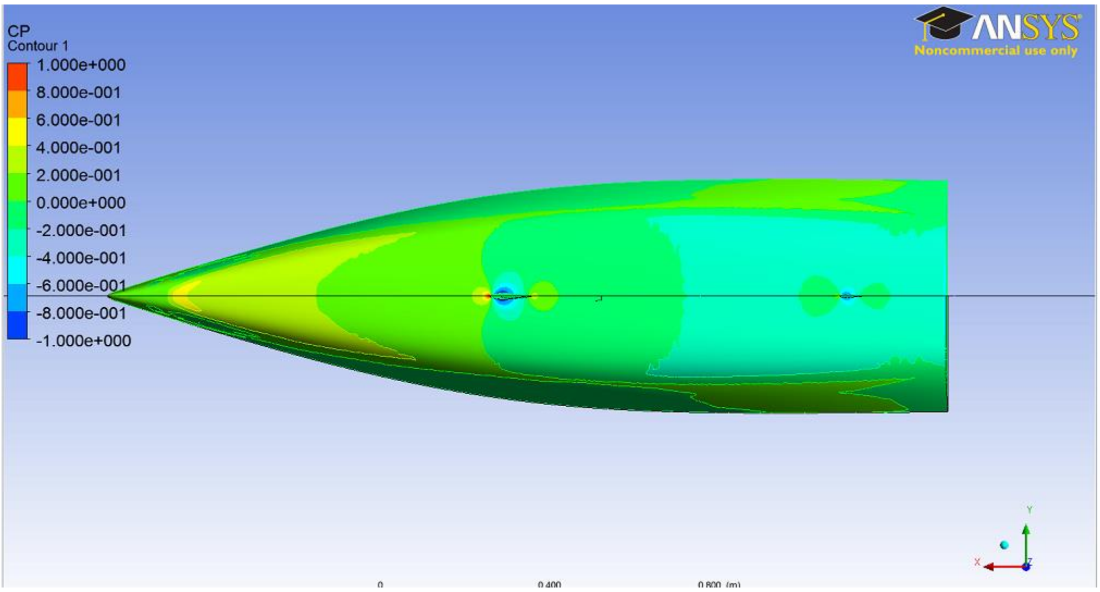

Figure 4.17 : Pressure coefficient plot in way of hull.

The pressure ceofficient plot for the hull is given in Figure 4.17. It is seen from the plot that the pressure is dropping under the hull as the water flows toward the stern. However, there are favourable pressure gradient regions in way of hull appendage connections where the pressure increases due to the existence of the keel and rudder as seen in Figures 4.18 and 4.19.

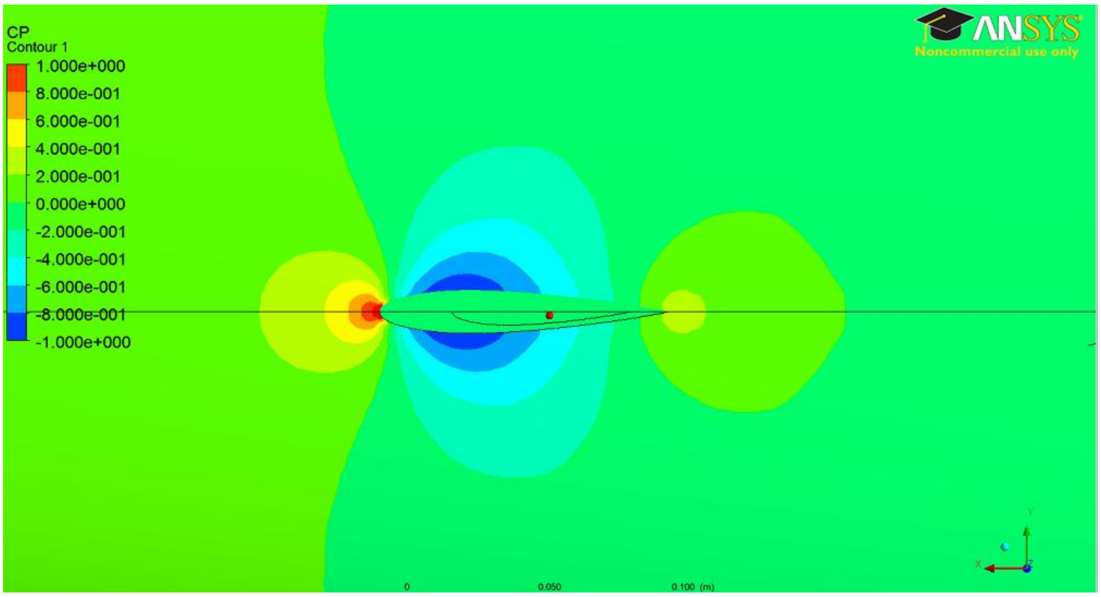

Figure 4.18 : Pressure coefficient plot in way of hull at keel root.

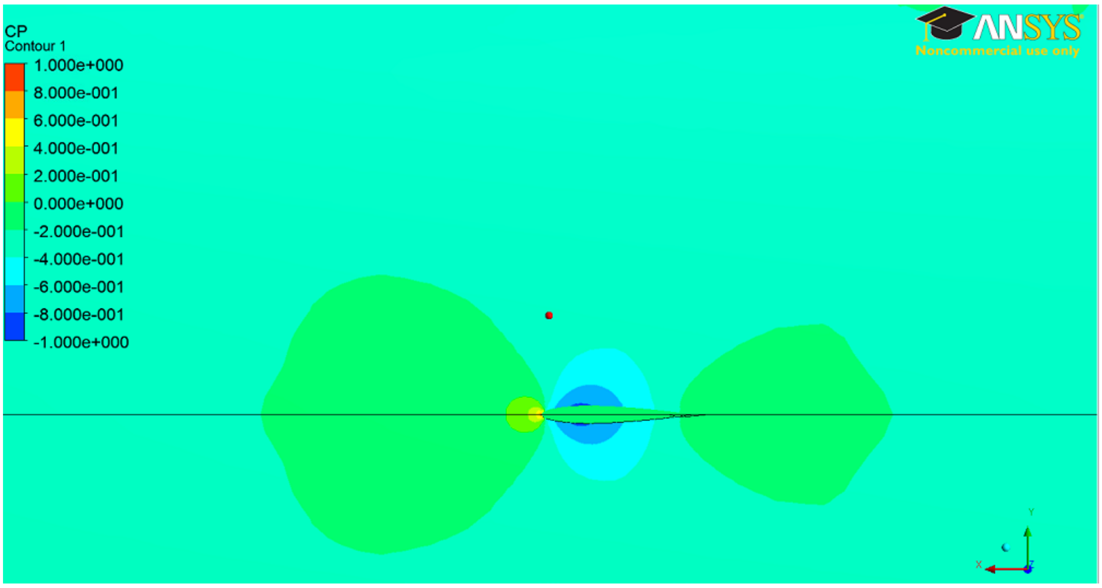

Figure 4.19 : Pressure coefficient plot in way of hull at rudder root.

4.2.2.2 1/8 Scale naked geometry in multiphase condition

The naked hull analysis in multiphase condition are conducted in order to assess the existance of any variation in the hull resistance components due to the existance of the appendages. Also, it would enable the scalability assessment if the analyses are conducted at different scales.

The solution domain and mesh details may be found in Figure 4.21 and 4.22 respectively. A total of 1.500.000 cells were used to discretize the solution domain. Computational modelling is similar to section 4.2.2.1.

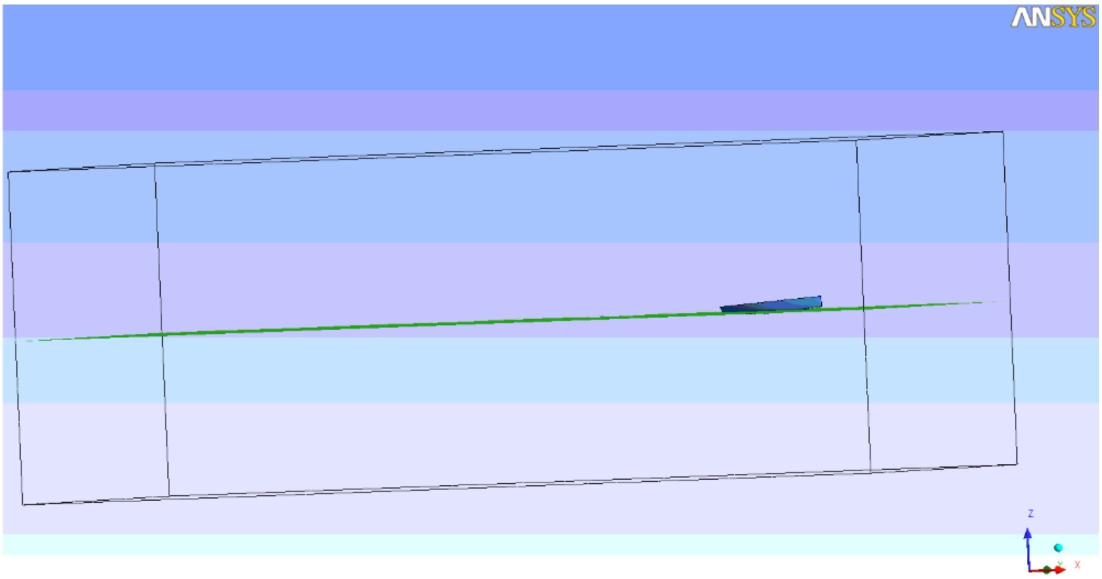

Figure 4.20 : Solution domain for the naked geometry.

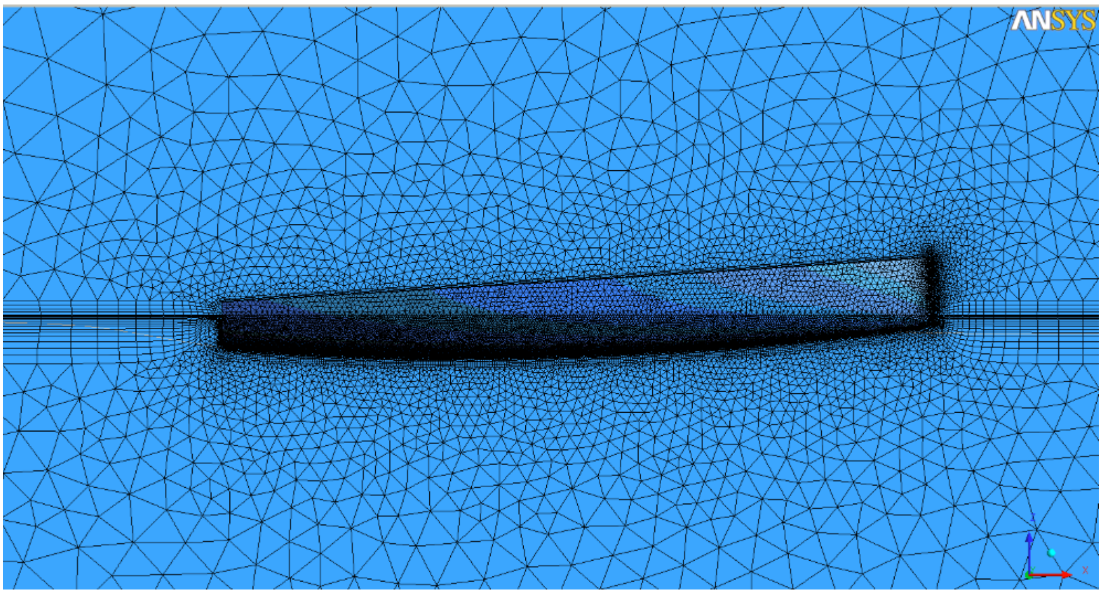

Figure 4.21 : Mesh in way of naked hull and free surface.

The wave profile at Froude number 0.5 for 1/8 scale in the naked condition may be seen in Figure 4.23. Similarity with the appended condition is apparent.

The resistance calculations show that there is approximately 1% difference in frictional resistance between the naked and appended condition (higher in the naked condition). Comparisons with the empirical Grigson frictional line indicate that the CFD code underpredicts the frictional resistance at the order of 2% with the current meshing strategy.

The residuary resistance difference between the naked and appended conditions is at the order of 3% (higher in naked condition). As the wave profiles and generated wave patterns are identical between the naked and appended conditions, the differences in the residuary (normal) resistances are attributed to the local pressure differences due to the existence of the appendages.

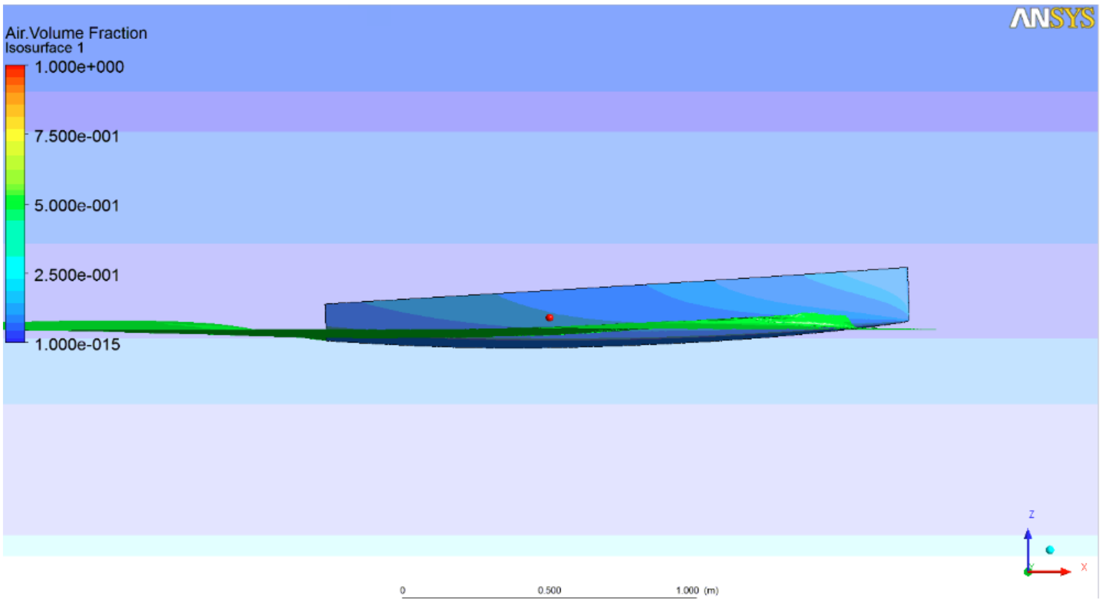

Figure 4.22 : Wave profile for naked geometry for 1/8 scale at Froude number of 0.5.

The computations in the naked condition state that, compared to the experimental results, the code underpredicts the frictional resistance at the order of 2%, overpredicts the residuary resistance at the order of 9% and the total water drag is overpredicted at the order of 6%.

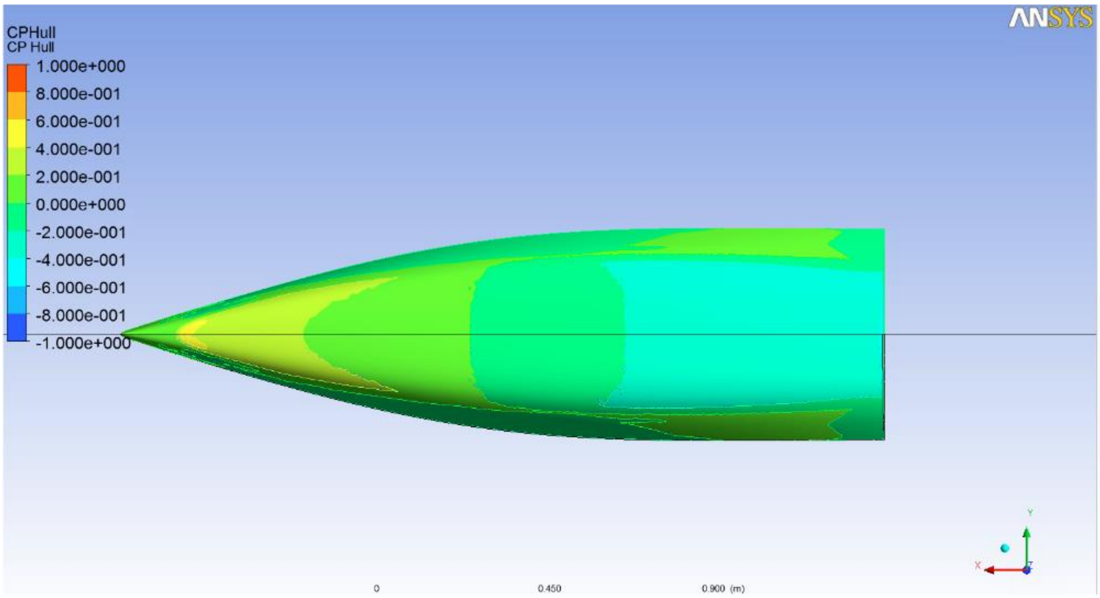

Figure 4.23 : Pressure coefficient plot in way of naked hull.

The pressure coefficient distribution in the naked condition, as seen in Figure 4.24 is similar to the appended condition. The pressure drop contours along the hull bottom are identical to the appended condition. The local pressure increases due to the existence of the appendages are the main difference between the pressure coefficient contours between the two conditions.

4.2.2.3 1/4 Scale fully appended geometry in multiphase condition

The appended computations in multiphase condition conducted at 1/8 scale have been repeated for the 1/4 scale at Froude number of 0.5. The main emphasis of the 1/4 scale investigations was to assess the variation of hull resistance components with scale.

The same mesh and solution modelling strategies were utilized with respect to 1/8 scale computations. The mesh has been scaled to comply with the 1/4 scale geometry.

The generated wave profile in the appended condition at 1/4 scale may be seen in Figure 4.25. The wave pattern at the center line may be seen in Figure 4.25.

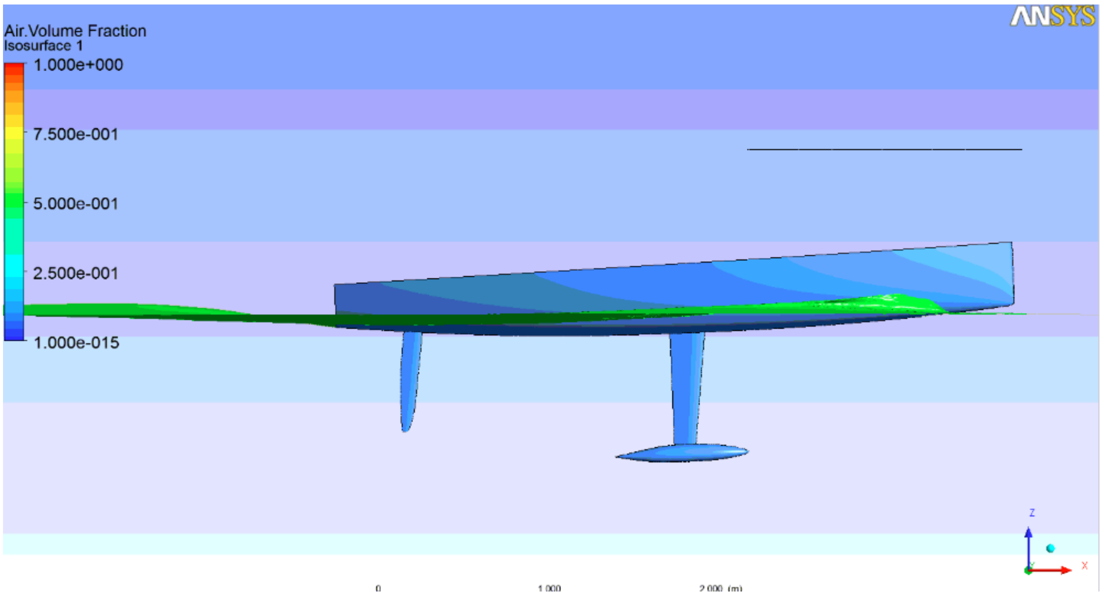

Figure 4.24 : Wave profile at 1/4 scale in the appended condition at Froude number of 0.5.

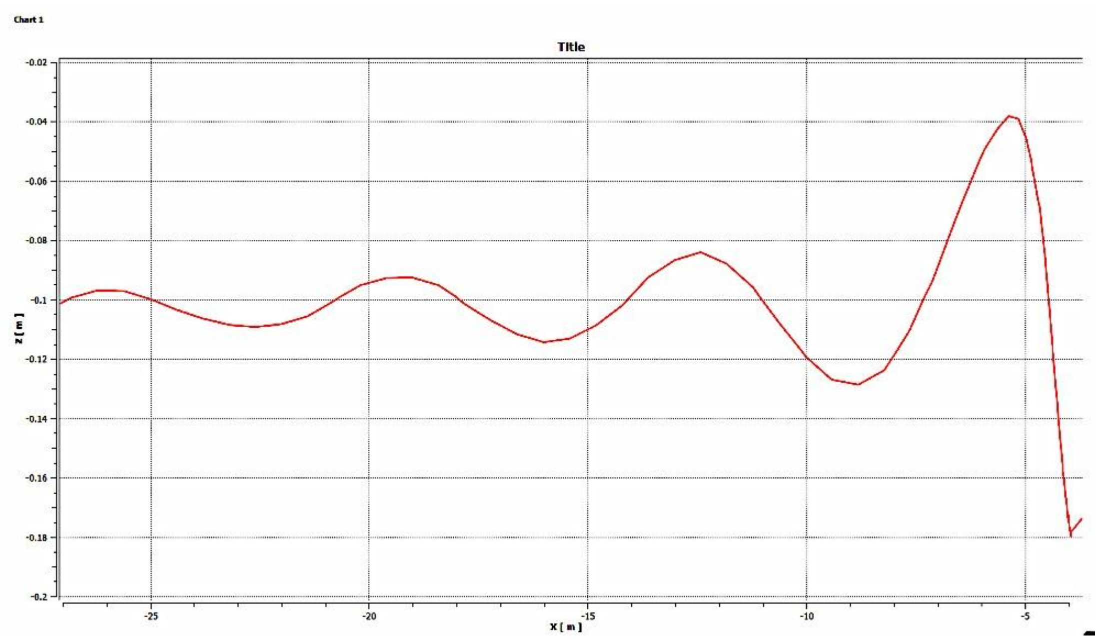

Figure 4.25 : Wave pattern at center line for 1/4 scale in the appended condition at Froude number of 0.5.

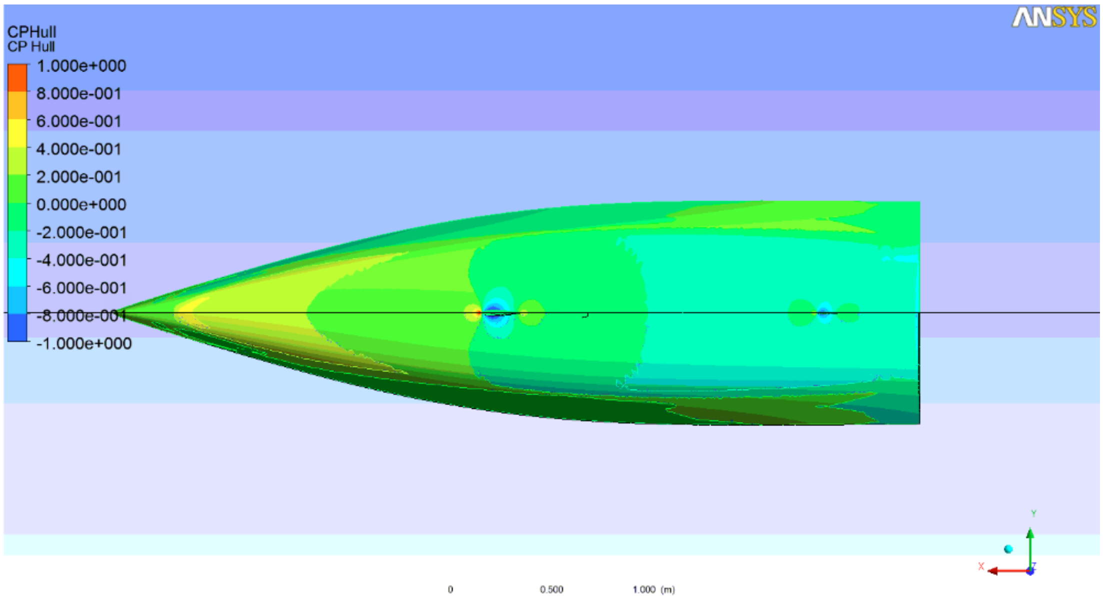

Figure 4.26 : Pressure coefficient contours in way of hull for 1/4 scale appended model at Froude number 0.5.

The calculated total resistance for the 1/4 scale appended case is 8.3967 kilograms for the whole body. This is comprised of 3.6173 kilograms of frictional resistance and 4.7794 kilograms of residuary resistance. The pressure coefficient plot is given in Figure 4.26.

4.2.2.4 1/4 Scale naked geometry in multiphase condition

The mesh modelling and solution strategy is similar to previous sections and will not be reported. The wave profile and wave pattern plot may be seen in Figures 4.27 and 4.28 respectively.

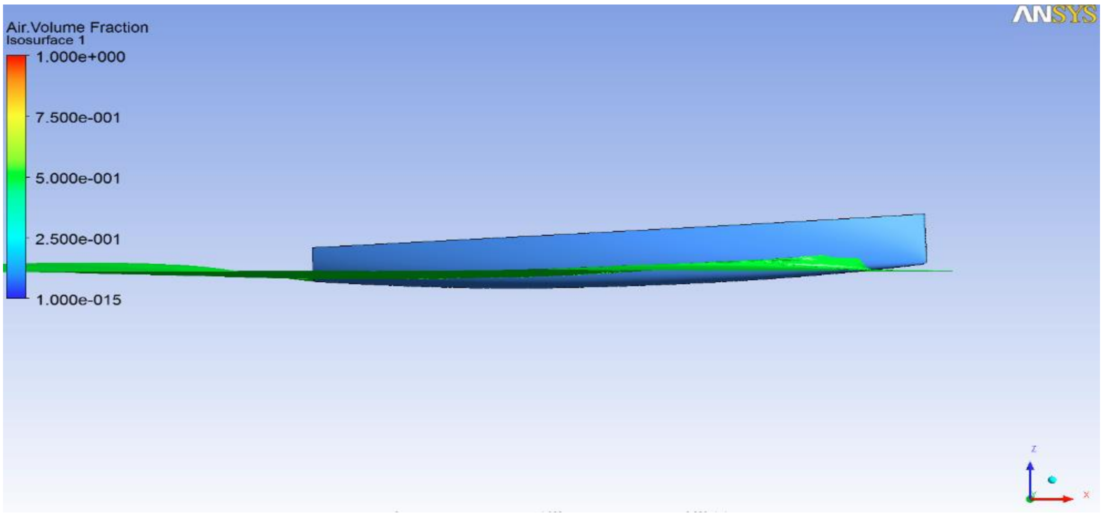

Figure 4.27 : Wave profile for 1/4 scale naked model at Froude number 0.5.

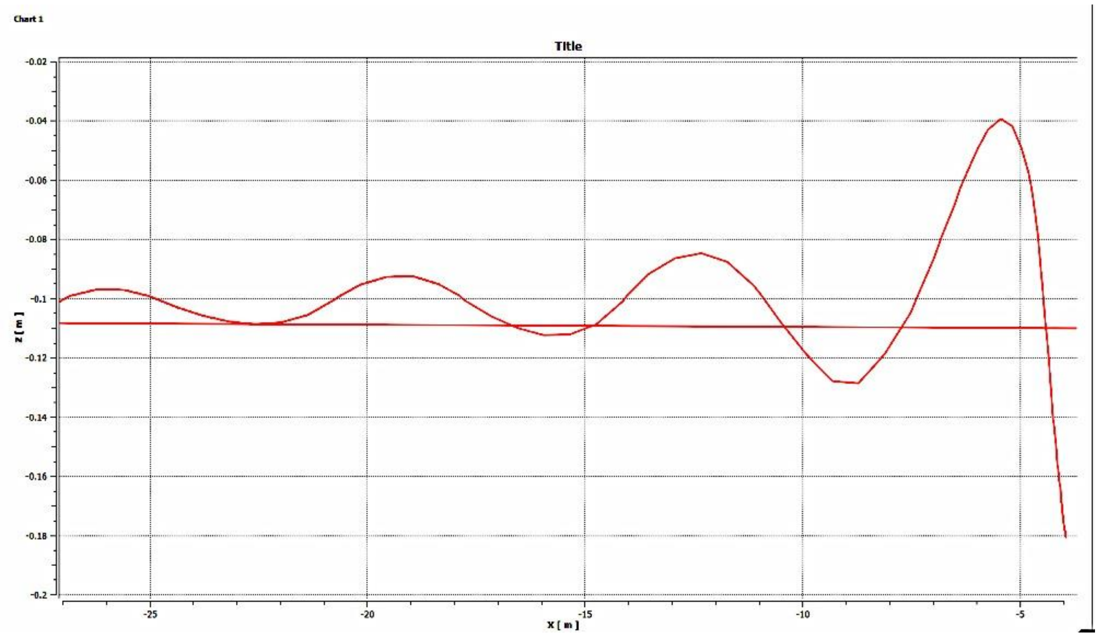

Figure 4.28 : Wave pattern at the center line for 1/4 scale naked model at Froude number 0.5.

The resistance results reveal a total resistance of 4.9058 kilograms for the whole body, comprising of 3.6409 kilograms of friction drag and 4.9058 kilograms of residuary drag. Results obtained are similar to the 1/8 scale condition; ~1% difference in frictional resistance and ~3% in residuary resistance (both higher in the naked condition).

The pressure contours, as seen in Figure 4.29 are also identical to the naked condition at 1/8 scale.

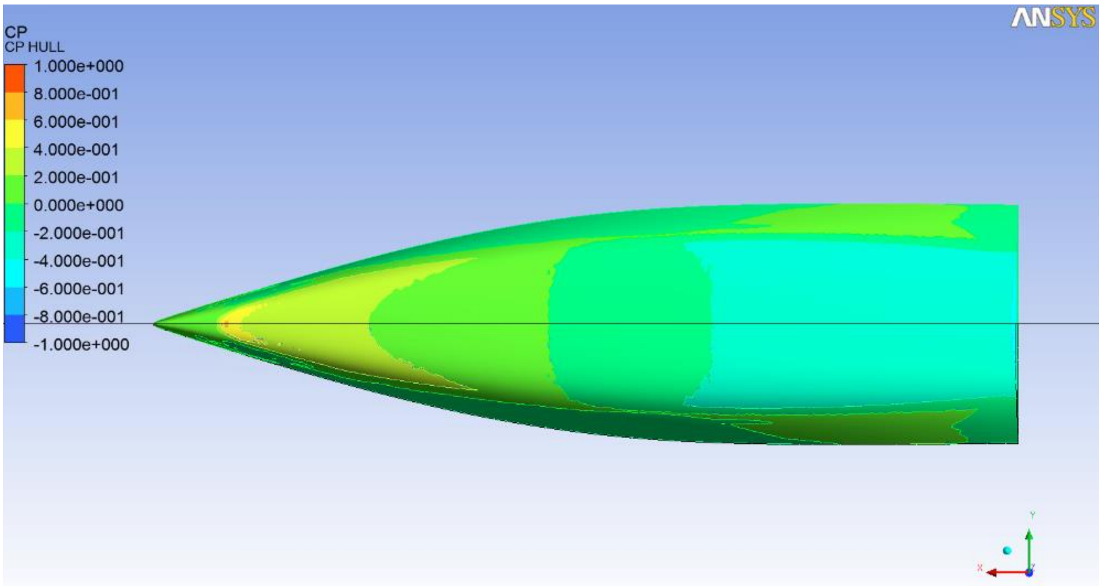

Figure 4.29 : Pressure coefficient contours in way of hull for 1/4 scale naked model at Froude number 0.5.

4.2.3 Discussions on the results of the upright CFD calculations

A number of conclusions may be drawn from the current set of CFD analyses in terms the ability of the utilized code in predicting the physical phenomenon associated with saling yacht testing.

It has been found out that the consideration of empirical friction lines associated with form factor formulations may not be sufficiently accurate as they may not account for the interaction between the hull and appendages as well as the dynamic attitude of the vessel.

The keel and rudder normal resistances are affected by the existence of the hull (upto 24%) whereas the normal resistance of the bulb changes substantially as the vessels adopts a trim angle (56%). Even though there are formulations that enable the accounting of the interactions, these are based on few key parameters and do not account for the dynamic attitude of the craft.

For minimisation of error at small scale testing, it is recommended that CFD computations are used for the calculation of the appendage resistances for the test matrix during the post processing of the experimental results. If a RANS code is being utilized, consideration of the underwater geometry with respect to the dynamic attitude (trim and sinkage) would yield results in a reasonable time frame.

Any variation on the appendage normal resistance due to the existence of the free surface has not been considered in the computations as suitable modelling of the wake of the appenages has not been possible with the utilized meshing tool and strategy.

The variation of the components of the hull resistance, mainly the residuary resistance has been investigated by analyses at two different scales. It is found out that under the assumption of identical trim and sinkage, the residuary resistance is exactly scalable with the cube of the scale factor, however there are differences between the residuary resistance components between naked and appended conditions.

The differences are less than 3% between cases in the two scales investigated. These differences are highly associated with the pressure differences caused by the existence of the appendages as the rest of the pressure contours and wave patterns are identical between the investigations.

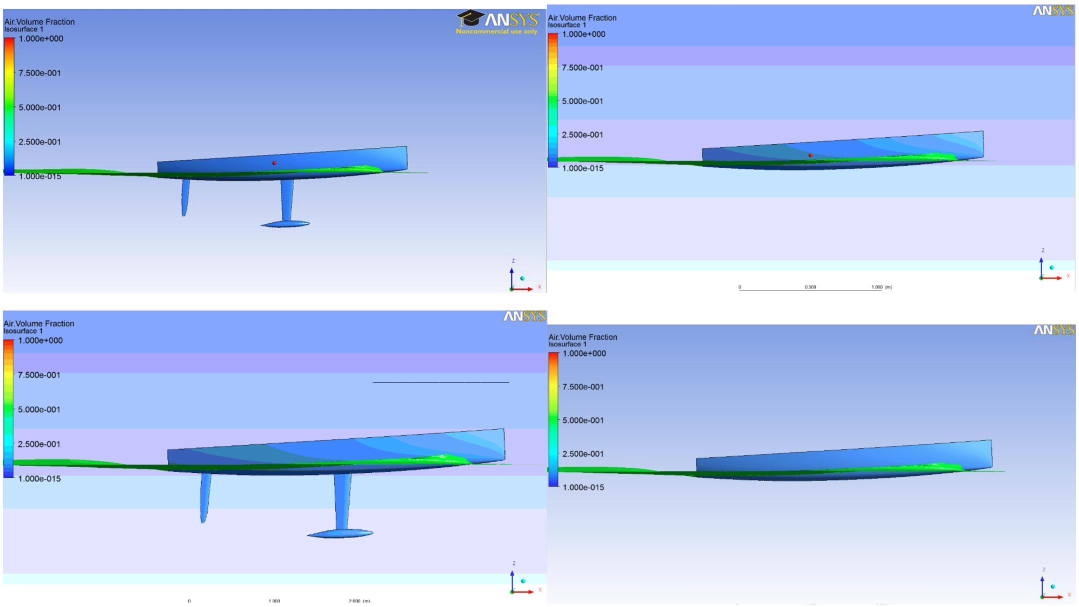

Figure 4.30 : Wave profiles for 1/8 appended, 1/8 naked, 1/4 appended and 1/4 naked cases respectively.

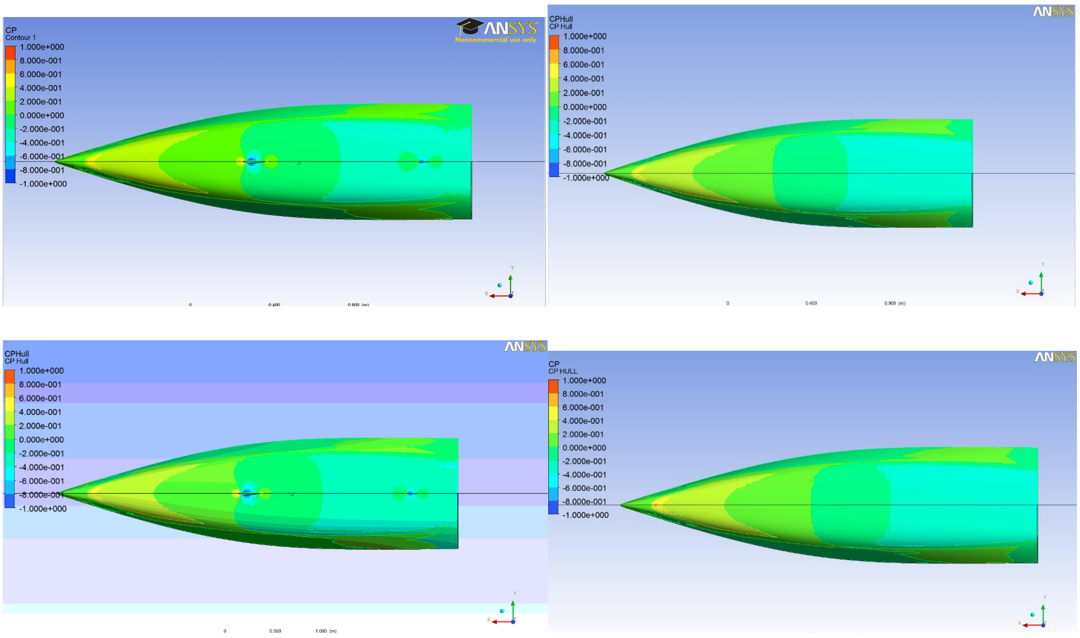

Figure 4.31 : Pressure contours for 1/8 appended, 1/8 naked, 1/4 appended and 1/4 naked cases respectively.

As mentioned in the previous sections, with the modelling strategy involved, and compared to the experimental results, the code underpredicts the frictional resistance at the order of 2%, overpredicts the residuary resistance at the order of 9% and the total water drag is overpredicted at the order of 6% for the hull. The normal resistance of the appendages are overpredicted by 3%, the frictional resistance is underpredicted by almost 15% for foil shaped appendages compared to the utilized empirical formulations. The underprediction is at the order of 1% for bulb fricitonal resistance. Even though theere are differences between the numbers, it has been seen that the consistency of the results enabled identification of the trends between cases considered.

As the variation in the normal resistance of the appendages due to are operating under the hull at a certain trim angle at Froude number of 0.5 are accounted for in the post processing of the experimental results –by adding in the differences between the empirical form resistances and the computed normal resistances-, the wave resistance coefficient plot becomes as in Figure 4.32.

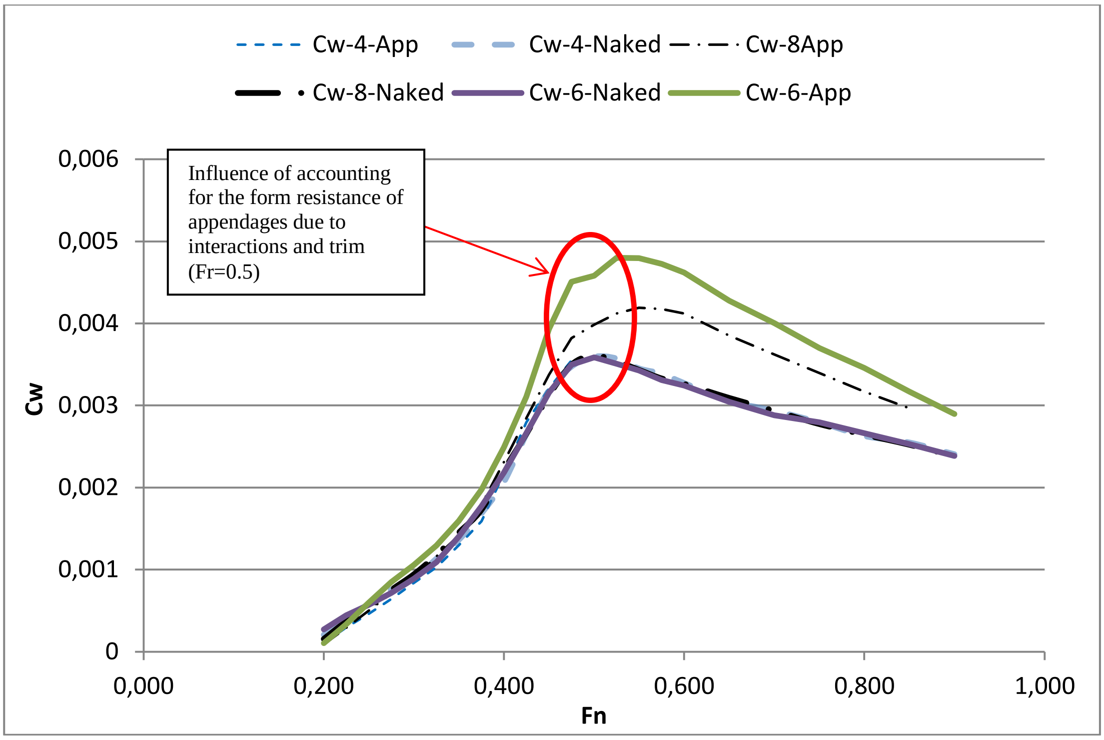

Figure 4.32 : Wave resistance coefficients accounting for the appendage resistance variations for 1/8, 1/6 and 1/4 scales.

In Figure 4.32, it is seen that with the accounting of appendage resistance variations, the wave resistance coefficient at 1/4 scale appended condition fits with the wave resistance coefficent at 1/4 scale in the naked condition. The wave resistance coefficient at 1/8 scale appended condition is 11% higher compared to the wave resistance coefficients for 1/4 scale appended condition and 1/4, 1/6 and 1/8 scale naked conditions.

4.3 Heeled Analysis

The heeled CFD analyses were conducted for scales 1/8, 1/4 1/1 scale at 15 degrees of heel for a full scale speed of 8 knots. Leeway angle was varied as 0, 2, 4 and 6 degrees. Only underwater part of the hull was modeled in the appended condition as the sole aim of these computations was to investigate the lift generation efficiency characteristics with scale. Therefore, the variation of wave resistance with leeway and its influence on the measured drag was isolated from the investigation by considering the underwater part of the hull and the appendages. Turbulence modelling was same as upright analysis.

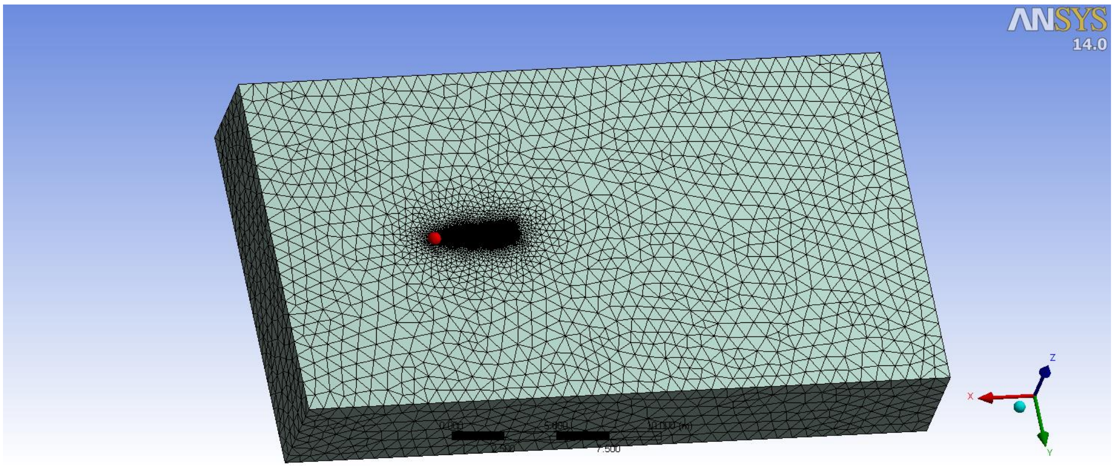

Figure 4.33 : Solution domain used for heeled computations.

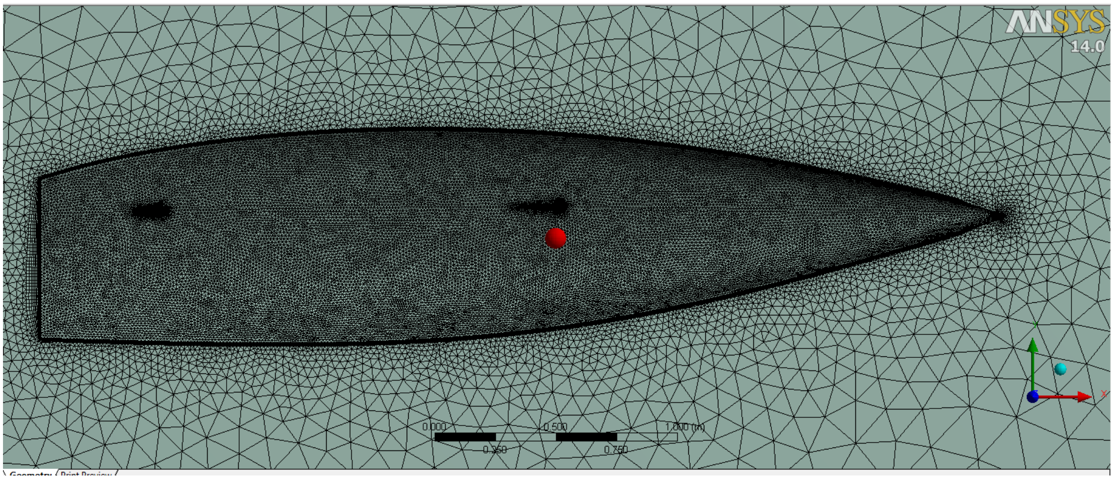

Figure 4.34 : Mesh details in way of hull.

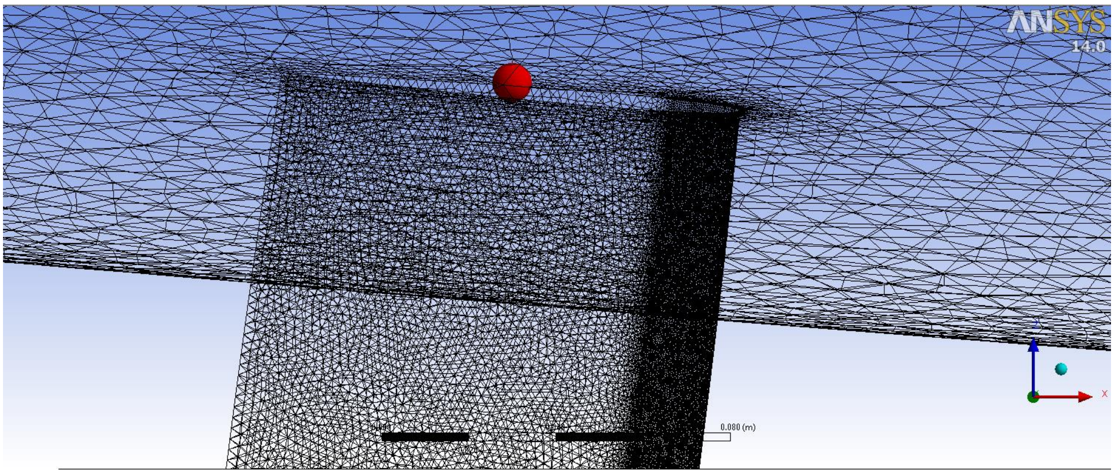

Figure 4.35 : Mesh refinement in way of rudder.

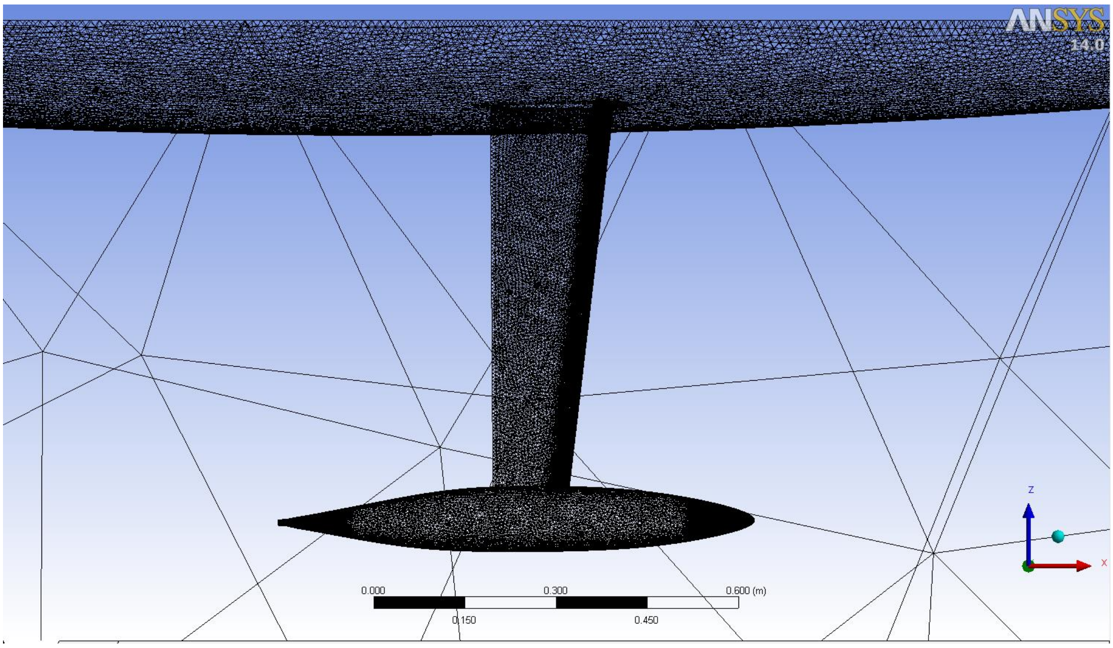

Figure 4.36 : Mesh refinement in way of keel and bulb.

As seen on the mesh detail figures, boundary layer meshes were created for the hull, rudder, keel and bulb for proper turbulent wall treatment. Appendage leading edges were refined substantially to capture the right geometry and improve frictional resistance calculations.

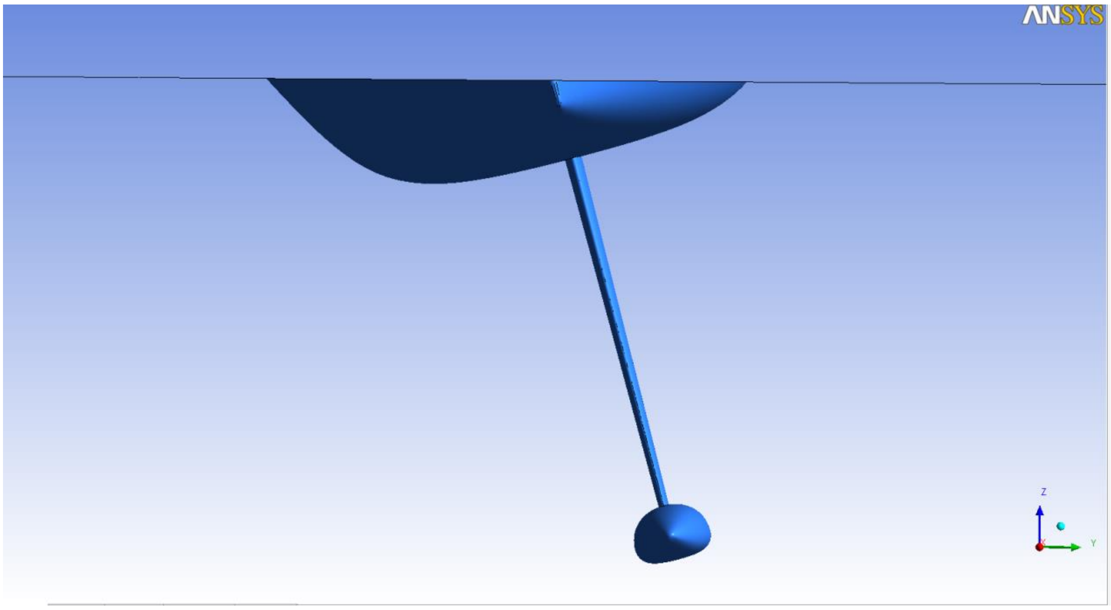

Figure 4.37 : Section view of the hull in 15 degrees of heel, no leeway.

Variation of side force, heel drag and induced drag was assessed for 3 scales. In the below figure, variation of total drag versus side force square is shown.

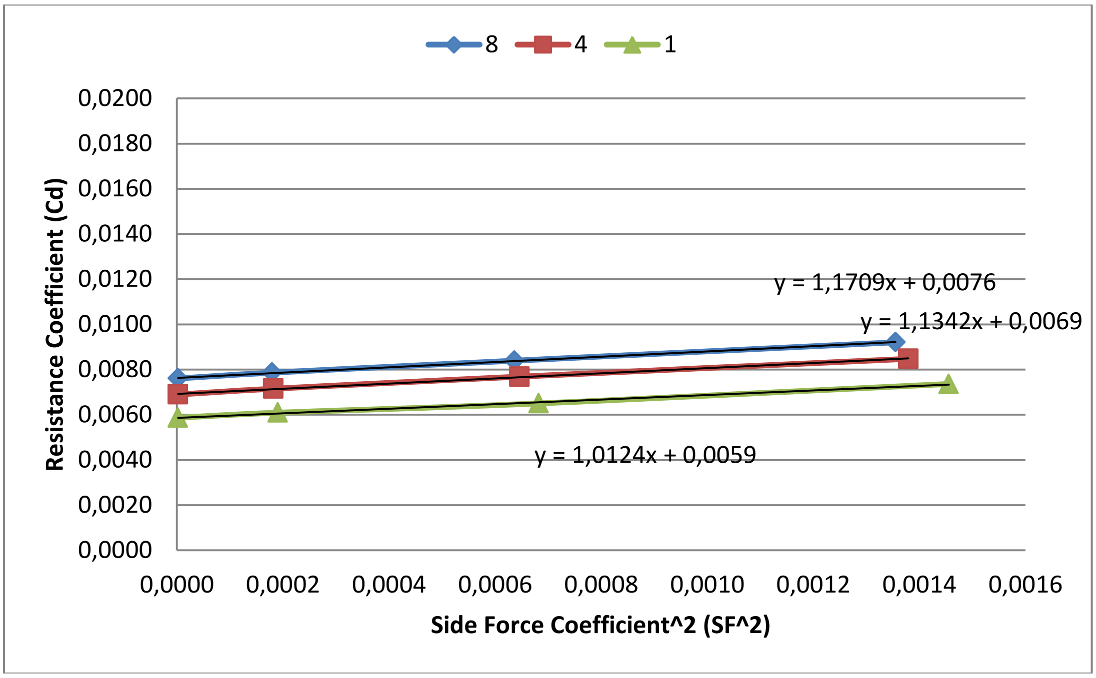

Figure 4.38 : Variation of total drag versus side force.

The base drag values in coefficient form consisting of the total upright drag and heel drag are different for all scales as expected. This phenomenon was observed in the upright towing tank tests for scales ranging from 1/10 to 1/4. The sideforces in coefficient form are very similar, some deviations are present in the higher leeway angles within the expected range. This has not been observed in the experimental results as the experiments were prone to Reynolds Number dependency, especially at smaller scales. The fully turbulent flow assumption in the RANS CFD methodology results in similar lift generation characteristics among the scales analyzed.

The most crucial fact about the comparison of experimental and computational data in the heeled condition is in the slope of the curves which is an indication of the efficiency of the boat as a lifting surface. The decrease in the slope of the curves is also present in the experimental data as scale is decreased. Also, it is evident that the slopes in the experimental data are higher compared to computations. This is owing to the fact that the computations have been carried out with the underwater hull only, i.e. the wave drag is not accounted in the computations. Therefore, it can be stated that the portion of the induced drag in relation to lift generation is variable with scale and this feature can be captured with CFD tools.

5. CONCLUSIONS AND RECOMMENDATIONS

A number of conclusions can be drawn from the current series of sailing yacht model tests and computational analysis. It has been found that both upright and heeled/yawed sailing condition tests are prone to scale effects. The magnitude of scale effects increases with smaller model size. Hence, a towing tank test campaign has to balance the cost of large model size and uncertainty of small model size. Guidance on the sailing yacht model size based on required accuracy is beneficial, especially if small size models are chosen for the first optimized form search and large model size for the realistic performance estimations.

In the upright condition, scale effects on viscous resistance and wave resistance needs to be considered mainly for the appended model case with the keel, bulb and rudder.

The wave pattern resistance similarity in upright condition is easily satisfied by using Froude similarity. This has been confirmed with the results of the upright wave pattern tests. The wave pattern resistance is consistent for all 4 models tested both in naked hull and appended conditions. The appendages have negligible effect on the wave pattern as they are deeply submerged in the water.

On the other hand, the scale effects are encountered both for naked and appended configurations, as the viscous resistance similarity is achieved only by using turbulence stimulation devices and compliance with the skin friction formulae. The upright naked hull total resistance tests results clearly indicate that except smallest model, this approach has been satisfactory. Therefore, a model size of approximately 2.0 meters may be sufficient in the naked condition. Smaller model total resistance test results are considerably higher than the other models leading to much higher uncertainty in resistance. Although larger models are required for appended tests, small size models (between 2m and 4m) may be utilized for earlier stages of testing campaigns; in which the main emphasis of the tests are hull form optimization, wave drag minimization or similar. This would yield a more cost-effective solution with a quicker turn around time and would also enable testing in a smaller facility as well as makes it possible to utilize a larger test matrix in a given time interval.

Models smaller than the above mentioned sizes suffer from high uncertainty as the measured forces are comparable to the repeatability of the utilized measurement system. At 1/10 scale, the repeatability of the force measurement system is approximately 10% of the calculated wave resistance at slow speeds. Also, displacement similarity is hard to achieve at these scales even though very light weight model construction is utilized.

The effect of appendages in upright condition cannot be taken into account in a straightforward fashion. The currently utilized methodology of using of turbulent flow stimulation and skin friction formulae combined with individual form factor estimations for each appendage may not be sufficiently accurate with model sizes less than 4 meters. Further investigations of hull-appendage interaction, variation of hull and appendage viscous resistance with speed could be beneficial to gain a better understanding of resistance in upright condition with scale.

The currently utilized ITTC methodology for force breakdown and extrapolation is based on certain empirical assumptions and works with sufficient accuracy for conventional ship types which are equipped with appendages that are smaller in size and wetted area compared to sailing yachts. Therefore, any error regarding the estimation of appendage frictional and form resistance components is easily compensated with the power margins while extrapolating for these ship types. When it comes to yacht testing, as the appendages comprise a substantial portion of the wetted area, the error in the estimation of the appendage resistance parameters limit the accuracy of the experimental results.

In addition, the high-speed nature of sailing yacht tests causes inaccuracies with conventional force breakdown and extrapolation methods. As the yacht models are towed at high Froude numbers ($Fr>0.4$), the dynamic attitude results in sinkage and dynamic trim, leading to changes in the wetted surface area. This change has been found to be up to the order of 15% during the experiments. Therefore, utilization of static wetted area for frictional resistance calculations would lead to exact same percentage of error in frictional resistance estimation. The influence of this error on the obtained wave resistance depends on the speed; the effect is higher at slow speeds where frictional resistance is dominant and reduces down to 10% at high speeds at which wave resistance is the dominant portion of the total resistance.

In the currently utilized methodology, the appendage resistance values are estimated from empirical formulae that are valid under certain circumstances. The keel and rudder form resistances are estimated from formulae that have been developed for aerodynamic purposes and that are based on formulations accounting for the thickness and chord ratio of the foils. These formulations are valid for 2-D sections and are lacking to account for any interaction between the hull and the appendages. Also, most importantly any influence of the change in the trim of the yacht may not be accounted for with these formulations.

During the CFD studies it has been found out that the empirical formulation results for foil like appendages may be estimated satisfactorily with CFD for a 2-D case (solution domain terminating at the keel root and tip). Also, it has been found out that the keel form resistance increases 6% when the keel operates above the bulb. As the keel operates at the test case trim angle between the hull and the bulb, the increase in the form resistance is 19% compared to the 2-D case (at Fr=0.5).

The CFD study revealed that the bulb form resistance is less prone to interactions from the existence of the keel. However, the bulb form resistance has been found to be extremely vulnerable to trim; the increase in the form resistance has been found to be approximately 150% at Fr=0.5.

As the variations in the appendage resistance due to interactions and trim may not be accounted with empirical formulations, other methods are necessary. CFD investigations accounting for the underwater body may be beneficial, as the turnaround time may be kept reasonable with sufficient accuracy. With this methodology, the dynamic attitude of the vessel has to be known or estimated. Another method that may be proposed is to use the Froude scaling and assume that the pressure based total resistance or the total residuary resistance (frictional resistance of the hull, keel, bulb and the rudder deducted from the total resistance) would be identical in coefficient form between different scales at similar Froude numbers. Further investigations are required to assess the accuracy associated with this methodology.

Stripping of the appendages into vertical sections for calculation of Reynolds number and frictional resistance has been proposed during previous research attempts. As the modern appendages are less sweeped compared to the ones in the past, the utilization of stripping has less than 1% effect on the frictional resistance estimation of the appendages. Therefore, this method is only recommended for appendage configurations with higher sweeped geometries.

The scale effects on the hydrodynamic side force are another concern as sailing condition shall be largely dependent on the relationship between resistance and side force. Traditionally the slope of C_L^2-C_D plot is utilized commonly in the performance estimation. There is considerable slope uncertainty increase in the small model sizes. A minimum model size could be specified as 2.5 m to limit uncertainty in the side force less than 10 % of the slope. The uncertainty drops to 5% at a model size of 4.0 meters. Above this size, the marginal gain of increasing the size decreases; there is a substantial size increase for a unit of improvement in accuracy. Therefore, the model size decision in appendage development studies has to be made in conjunction with a proper assessment of expectations from the experiments prior to testing.

Guidance on the model size considerations mentioned previously has naturally been based on a TP52 yacht and these may reflect the scaling conditions for similar size vessels. Although the experimental correctness has been based on model size, some other parameters need also be considered for generically applicable conclusions like speed, Reynolds number of the hull and appendages, measurement system capabilities, displacement similarity issues, blockage effects and etc. Previous researchers have recommended that the artificial stimulation devices may not produce the desired effects if the Reynolds number is less than 100000. The decision on the speed range of the tests and the model size should take into consideration the Reynolds number of the appendages as a starting point. Other parameters need to be considered on a case-by-case basis.

CFD methods may be used for testing campaigns as a supplementary tool in conjunction to towing tank tests. Especially RANS based CFD tools have been found to be performing well in estimation of viscous and wave based resistance components. It has been found out that the friction-based resistance could be calculated within less than 2% differences compared to empirical friction lines. The uncertainty in terms of wave based resistance sources within the utilized CFD code was found to be in the order of 9%. This could be reduced with the refinement of the mesh in the cost of increasing the computation time. The tool is also sufficiently sensitive to geometry variations; making it a useful tool to compare alternative designs. These tools would be convenient for initial design candidate evaluation, local flow investigation and optimization studies. As these tools are still time consuming due to the computational power requirements owing to peculiar features of the method, their popularity in the near future may be restricted to high budget campaigns, especially for detailed analyses.

The CFD studies have been found to be beneficial in the investigation of scale effects in sailing yacht testing up to a certain extent. Some of the deficiencies of the currently available testing and extrapolation methods could have been identified with the usage of CFD tools. Even though there are certain differences in the obtained values between the code and the experiments, the method has been found to give consistent results.

The fully turbulent nature of the utilized code prevents the identification of the discrepancies experienced during small scale towing tank testing, as the viscous flow condition similarity between the simulation and experiments may not be achieved. Therefore, further research should be concentrated to the flow conditions at low Reynolds number testing, with proper measurement devices in order to identify the sources of error while testing at this condition.

5.1 Recommendations for Further Research

It is recommended that the appendage behavior at small scales be investigated with suitable methods enabling a better understanding of the variation of flow field characteristics. Wind tunnel investigations as well as PIV measurements may be stated as suitable methodologies.

As accuracy of velocity performance of full-scale boat is the main aim of tank tests, further studies of errors in the velocity performance prediction due to uncertainties in the sailing yacht model testing are beneficial. Such a study may enable the designers to make trade off analysis between the model testing cost and accuracy of the results.

The measurement of boundary layer speeds with hot-wire instruments would also be beneficial in obtaining a better understanding of the physics behind the scale effects.

Repetition of these measurements in non-studded and studded conditions would improve the understanding. The studding pattern may be varied systematically, especially at small scales and the effects on the measurement parameters may be investigated.

In case of experimental investigations as in the current work; utilization of a three-post system is recommended as testing with a single post system causes difficulties at high speeds on big models.

It is also recommended that the experiments be repeated on both tacks and in order to eliminate scatter; the number of leeway angles should be increased at both tacks. In addition, measurement of resistance and side force for the hull, keel, bulb and rudder individually for different scales may improve the knowledge on scale effects.

REFERENCES

Ansys INC. (2010). *ANSYS CFX-Solver Modelling Guide*. Retrieved from http://www1.ansys.com/customer/content/documentation/120/cfx/xmod.pdf.

Binns, J.R., & Klaka, K., & Dovell, A. (1997). Hull-Appendage Interaction of a Sailing Yacht, Investigated with Wave-Cut Techniques, *CSYS '97. Proceedings of the 13th Chesapeake Sailing Yacht Symposium,* (pp.159-210). USA: Annapolis, January 25.

Brown, P. W., & Savistsky, D. (1987). Some Correlations of the 12 Metre Model Test Results, *ITTC '87. 18th International Towing Tank Conference - ITTC 87- Proceedings* (pp.471-473). Japan: Kobe, October 18-24.

Campbell, I., & Claughton, A. (1987). The Interpretation of Results from Tank Tests on 12M Yachts, *CSYS '87. Proceedings of the 8th Chesapeake Sailing Yacht Symposium* (pp.91-108). USA: Annapolis, March 7.

Caponetto, M. (1993). A Review on Il Moro di Venezia Design, *CSYS '93. Proceedings of the 11th Chesapeake Sailing Yacht Symposium*, (pp. 107-112). USA: Annapolis , January 29-30.

Claughton, A. (1989). The Effect of Counter Length on Hull Resistance, CSYS '89. *Proceedings of the 9th Chesapeake Sailing Yacht Symposium,* (pp.109-118). USA: Annapolis, March 11.

Davidson, K. M. (1936). Some Experimental Studies of the Sailing Yacht, *Transactions of the Society of Naval Architects and Marine Engineers,* 44.

DeBord, F. (1987). Review of the State of the Art for Sailing Yacht Model Tests and Future Challenges Facing Test Facilities. *ITTC '87. 18th International Towing Tank Conference -ITTC 87- Proceedings* (pp.463-470). Japan: Kobe, October 18-24.

DeBord, F. J., & Teeters, J. (1992). Accuracy, Test Planning and Quality Control in Sailing Yacht Performance Model Testing, *New England Sailing Yacht Symposium*, New London, Connecticut, USA: March 28-29.

DeBord, F., Kirkman, K., & Savistsky, D. (2004). The Evolving Role of the Towing Tank for Grand Prix Sailing Yacht Design, *27th ATTC Conference,* Newfoundland, Canada: August 6-7.

Fluent INC. (2003). Fluent Software Training - *Advanced Turbulence Notes.*

Fluent INC. (2005). Introductory Fluent Notes - *Modelling Turbulent Flows*. Fluent User Services Center.

Gerritsma, J., Keuning, A., & Onnink, R. (1991). The Delft systematic Yacht Hull (Series II) Experiments, *CSYS '91. Proceedings of the 10th Chesapeake Sailing Yacht Symposium*, (pp.27-40) . USA: Annapolis, February 9.

Gomez, A. G. (2000). On the Form Factor Scale Effect, *Ocean Engineering*, *26*, 97-109.

Graf, K., & Böhm, C. (2005). A New Velocity Prediction Method for Post Processing of Towing Tank Test Results, *CSYS 2005. Proceedings of the 17th Chesapeake Sailing Yacht Symposium*, (pp.67-78). USA: Annapolis, March 4-5.

Hockirch, K., & Brandt, H. (1999). Full Scale Hydrodynamic Force Measurement on the Berlin Sailing Dynamometer, *CSYS '99. Proceedings of the 14th Chesapeake Sailing Yacht Symposium*, (pp.33-44). USA: Annapolis, January 30.

Hoerner, S. F. (1965). *Fluid Dynamic Drag: Practical Information on Aerodynamic Drag and Hydrodynamic Resistance*. Bakersfield, CA: Hoerner Fluid Dynamics.

Insel, M. (1990). *An Investigation into the Resistance Components of High Speed Catamarans* (Doctoral dissertation).

ITTC. (2002). *Recomended Procedures and Guidelines - Testing and Extrapolation Methods - Resistance*.Retrieved from http://ittc.sname.org/2002_recomm_proc/7.5-02-02-01.pdf.

ITTC. (1999). *Recommended Procedures and Guidelines - CFD General CFD Verification.* Retrieved from http://ittc.sname.org/2002_recomm_proc/7.5-03-01-04.pdf.

Keuning, J.A., Sonnenberg, U.B. (1998). Approximation of the Hydrodynamic Forces Based on the Delft Systematic Yacht Hull Series, *International HISWA Symposium on Yacht Design and Construction,* Amsterdam, RAI: November 16.

Kirkman, K. L. (1974). Scale Experiments with the 5.5 Metre Yacht Antiope, *CSYS '74. Collected Papers of the Chesapeake Sailing Yacht Symposium*. USA: Annapolis, January 19.

Kirkman, K. L., & Pedrick, D. R. (1974). Scale Effects in Sailing Yacht Hydrodynamic Testing, *SNAME Annual Meeting*, New York, NY, USA: November 14-16.

Kirkman, K. (1985). The Evolving Role of the Towing Tank, *CSYS '79. Proceedings of the 4th Chesapeake Sailing Yacht Symposium* (pp.129-155), USA: Annapolis, January 20.

Letcher, J. S., & McCurdy, R. C. (1987). Data Collection and Analysis for the 1987 Stars & Stripes Campaign, *CSYS '87. Proceedings of the 8th Chesapeake Sailing Yacht Symposium* (pp.147-154). USA: Annapolis, March 7.

Murdey, D. C., Molyneux, W. D., & Killing, S. (1987). Techniques for Testing Sailing Yachts, *ITTC '87. 18th International Towing Tank Conference -ITTC 87- Proceedings* (pp.453-462). Japan: Kobe, October 18-24.

Nagami, Y. (1993). The Nippon Challenge America's Cup 1992 - Progress in Hull Development, *CSYS '93. Proceedings of the 11th Chesapeake Sailing Yacht Symposium,* (pp.113-122). USA: Annapolis, January 29-30.

Oberkampf, W. L., & Trucano, T. G. (2002). Verification and Validation in Computational Fluid Dynamics, *Progress in Aerospce Sciences*, 38, 209-272.

Parsons, B. L., & Pallard, R. (1997). The Institute for Marine Dynamics Model Yacht Dynamometer, *CSYS '97. Proceedings of the 13th Chesapeake Sailing Yacht Symposium,* (pp.153-162). USA: Annapolis, January 25.

Seragg, C. A., Chance, B., Talcott, J. C., & Wyatt, D. C. (1987). The Analysis of Wave Resistance in the Design of the Twelve Metre Yacht Stars & Stripes, *CSYS '87. Proceedings of the 8th Chesapeake Sailing Yacht Symposium* (pp.109-122). USA: Annapolis, March 7.

Teeters, J. R. (1993). Refinements in the Techniques of Tank Testing Sailing Yachts and the Processing of Test Data, *CSYS '93. Proceedings of the 11th Chesapeake Sailing Yacht Symposium,* (pp. 13-31). USA: Annapolis, January 29-30.

Todter, C., Pedrick, D., Calderon, A., Nelson, B., DeBord, F., & Dillon, D. (1993). Stars and Stripes Design Program for the 1992 America's Cup, *CSYS '93. Proceedings of the 11th Chesapeake Sailing Yacht Symposium,* (pp.207-222). USA: Annapolis, January 29-30.

Url-1 < http://nutkulab.itu.edu.tr/lab.htm >, date retrieved 01.10.2012.

Url-2 < ittc.sname.org/ITTC-News&2056.pdf>, date retrieved 17.02.2013.

van Oossanen, P. (1985). The Development of the 12 Meter Class Yacht Australia II, *CSYS '85. Proceedings of the 7th Chesapeake Sailing Yacht Symposium,* (pp.81-101). USA: Annapolis, January 19.

van Walree, F., & Luth, H. R. (2000). Scale Effect on Foils and Fins in Steady and Unsteady Flow, *R.I.N.A. Conference on Hydrodynamics of High Speed Craft*, London, UK: November.

APPENDICES

APPENDIX A: Model Dimension Check Data
APPENDIX B: Sail Trim Moment Data
APPENDIX C: Raw Measurement Data

APPENDIX A : Model Dimension Check Data

Table A.1 : Dimension check for the tested models.

	1 / 10	1 / 8	1 / 6	1 / 4	1 / 1	
girth - bulb cl to wl fr9	384	478	636	955	3801	Measurement Values
girth - cl to fr 8 wl	167	208	277	410	1636	
girth - keel trailing edge to rudder leading edge	573	720	966	1435	5782	
bulb trailing edge - transom	723	903	1208	1795	7170	
bulb leading edge - wl alt fr 20	698	874	1167	1762	7010	
girth - bulb cl to wl fr9	3840	3825	3816	3820	3801	Values scaled up to 1/1
girth - cl to fr 8 wl	1670	1665	1662	1640	1636	
girth - keel trailing edge to rudder leading edge	5730	5760	5796	5740	5782	
bulb trailing edge - transom	7230	7220	7248	7180	7170	
bulb leading edge - wl alt fr 20	6980	6995	7002	7048	7010	
girth - bulb cl to wl fr9	-1.03%	-0.63%	-0.39%	-0.50%		% Error
girth - cl to fr 8 wl	-2.08%	-1.77%	-1.59%	-0.24%		
girth - keel trailing edge to rudder leading edge	0.90%	0.38%	-0.24%	0.73%		
bulb trailing edge - transom	-0.84%	-0.70%	-1.09%	-0.14%		
bulb leading edge - wl alt fr 20	0.43%	0.21%	0.11%	-0.54%		
average error	1.05%	0.74%	0.69%	0.43%		

APPENDIX B : Sail Trim Moment Data

Table B.1: Trim moment in Nm for each scale at specified full-scale speed.

Full Scale Speed (knots)	1 / 10	1 / 8	1 / 6	1 / 4
5	0.41	1.11	3.793346153	20.48837127
6	0.63	1.67	5.654579931	30.27887779
7	0.89	2.37	8.008003632	42.81384527
8	1.26	3.33	11.20812495	59.80066023
9	1.95	5.09	17.06291114	90.67274643
10	2.99	7.77	25.8802156	137.036113
11	4.23	10.93	36.3086138	191.8388326
12	5.33	13.76	45.64229	240.961888
13	6.21	16.03	53.16537	280.6697224
14	7.02	18.13	60.13406	317.5168566
15	7.80	20.15	66.86165	353.1378995
16	8.30	21.46	71.29638	376.8266712
17	8.77	22.72	75.57813	399.7487925
18	9.31	24.15	80.40968	425.5711723
19	10.04	26.06	86.80643	459.5919388
20	10.85	28.18	93.91516	497.3588974

APPENDIX C : Raw Measurement Data

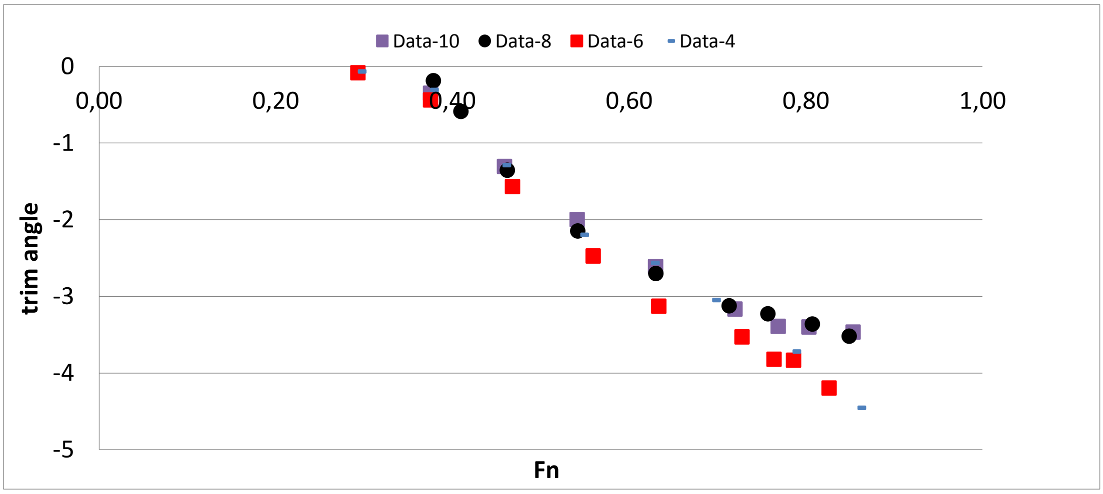

Figure C.1 : Raw trim data in upright naked condition.

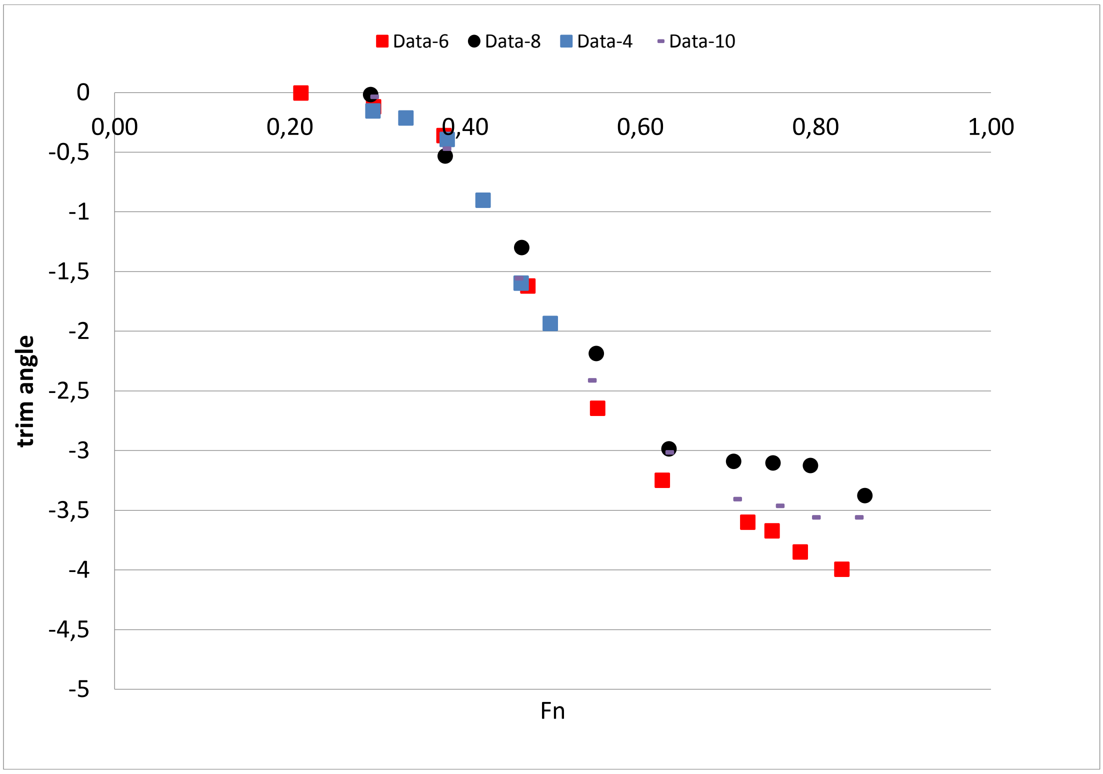

Figure C.2 : Raw trim data in upright appended condition.

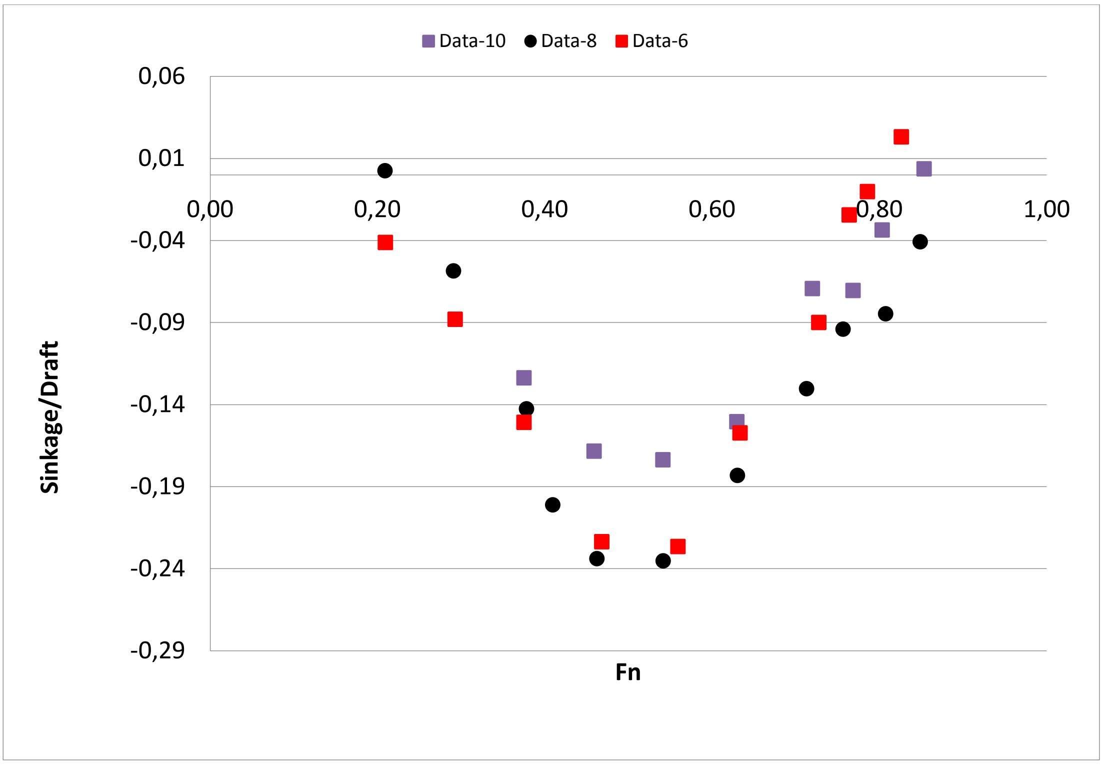

Figure C.3 : Raw sinkage data in upright appended condition.

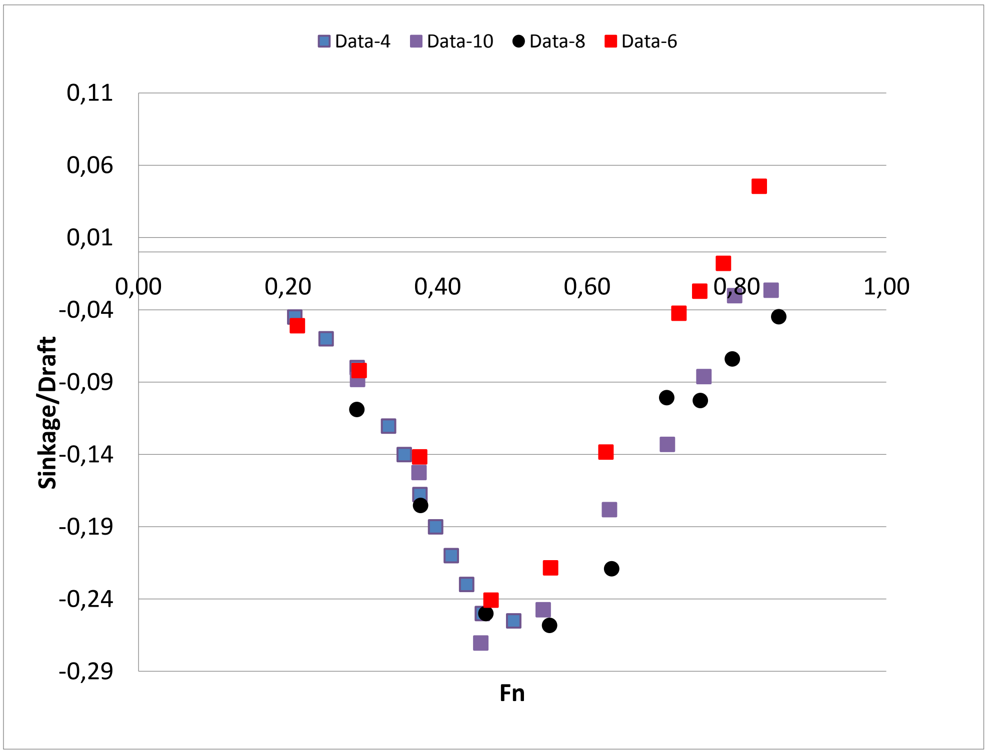

Figure C.4 : Raw sinkage data in upright appended condition.

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PUBLICATIONS/PRESENTATIONS ON THE THESIS

▪ **Saydam, A.Z.**, Insel, M. (2012), Investigation of Scale Effects in Sailing Yacht Hydrodynamic Testing. 4th High Performance Yacht Design Conference, 12-14 March 2012, Auckland, New Zealand.

▪ **Saydam, A.Z.**, Insel, M. (2013), Investigation of Scale Effects in Sailing Yacht Performance Prediction by Experimental Methods. 21th Chesapeake Sailing Yacht Symposium, March 2013, Annapolis, USA.