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HYDRODYNAMICS WITHOUT INTEGRALS

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SEAKEEPING - THE FORGOTTEN FACTOR

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Paper presented at

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HYDRODYNAMICS WITHOUT INTEGRALS

2 November 2000, Fremantle, Western Australia

Seakeeping – The Forgotten Factor

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INTRODUCTION

Like any design process, the design for favorable ship motion or the design of a motion control system requires some rational basis. A measurable set of requirements must be defined for a design to evolve and be objectively assessed. *Good seakeeping* and *comfortable* have no meaning on their own, they are as specific as requesting a *nice* boat. In order to deliver a product or service to meet a requirement, one must first be able to define what the requirement is.

A challenge to motion analysis is the transition from qualitative statements of seakeeping and comfort to a set of deliverable requirements. This brief consideration of seakeeping is intended to raise awareness on the subject and provide the reader with some assistance in reaching the objective of (at least) ensuring the end user gets what they expect. This will be considered from a risk management perspective.

THE RISK ASSESSMENT PERSPECTIVE

Ship design and construction may be perceived as an exercise in risk management, where the designer works to minimize the risk of failure through rational process. In a design sense this may be a safety related risk reduction or in a commercial sense this could be a reduction in the risk of commercial failure.

Risk: *to expose to the chance of injury, loss or hazard. Where loss is the failure to preserve or maintain a condition*

Reducing the risk of a particular outcome typically involves removing the hazard. The cost of hazard removal may be considered as insurance.

Insurance: *the premium paid for guaranteeing against risk.*

Designers and builders work to manage the risk by insuring against certain outcomes. Stability calculations are made as insurance against capsize, structural calculation are made to insure against structural failure and a designer may insure against poor speed performance by calculation or model test.

There is also the consideration of monetary insurance cover against unforeseen risks such as fire and damage. Most builders would not think twice about insuring against construction damage, but how many owners or operators spend a similar amount ensuring the vessels operability is not impacted through poor seakeeping?

To invest in insurance is to protect oneself against a possible future setback

SEAKEEPING – THE FORGOTTEN FACTOR

In relation to other design and construction related risks, seakeeping is very identifiable, keenly discussed but is rarely well quantified.

Seakeeping: *the quality of the environmentally induced ship motion*

Seakeeping is the one quality that tends to remain associated with a particular vessel for life. How often have you heard someone say, "She's a good sea boat" or "she rides well"? It is rare for any other quality to be used so frequently in describing a particular ship or boat as seakeeping. While seakeeping is one of the most spoken of qualities it is possibly one of the least considered design qualities.

Many other factors in design are covered by standards setting minimum threshold for compliance. It would be difficult for instance to construct a ship to classification standards without being structurally sound. Equally so, it would be difficult to pass statutory requirements in respect of stability and be inherently unsafe. It is however very possible to build a ship that has poor seakeeping qualities in the context of classification rule and statutory requirements.

Seakeeping is one quality of ship design that is not forced by regulation. This is particularly relevant for new or different vessels outside zone of ones familiarity. Perhaps this is why seakeeping is often given less consideration in the design process than say, speed.

There is no reason why seakeeping cannot be managed in the same manner as any other ship design and construction risk.

REDUCING THE RISK OF POOR SEAKEEPING

The process of reducing the seakeeping risk to an owner, operator or builder may be broken down into a rational process. This type of process may include the following steps:

1. Identify the motions that are important to the vessel operation;
2. Define the motions in measurable terms;
3. Define acceptable limits to the motions;
4. Select a technique to determine the motions;
5. Assess the motion against the criteria, then
6. Act on the outcome.

Each of these points will now be discussed.

1. **Identify the motions that are important to the vessel operation**

Assess what motions are critical to your operational success. An example of the types of motion related criteria that may be relevant is given in Table 1.

Ship Subsystem	Criteria with regard to:											
	Slam	Deck Wetness	Air Exposure	Vertical accel	Lateral accel	Roll	Pitch	Heave	Vertical Velocity	Relative Motions	MII	MSI
Ship hull												
Propulsion machinery												
Ship equipment												
Cargo												
Personnel effectiveness												
Passenger comfort												
Helicopter operations												
Sonar operations												
Lifting operations												

Table 1: Limiting criteria that is relevant to a vessel's subsystem

2. **Define the motions in measurable terms**

Assign a measurable quantity to each of the critical operational parameters. The motion must be measurable to be assessed. This includes being able to define the environment in which the vessel is to operate, speed, loading mass distribution, heading relative to weather and the points of reference on the vessel at which the assessments are to be made. Table 2 includes a sample of the measured quantities that may be used in the analysis.

- Motions in six degrees of freedom
- Relative vertical motion
- Probability of slamming
- Probability of green water on deck
- Probability of air exposure
- Vertical acceleration according to ISO 2631 (motion sickness)
- Forces in body-fixed coordinate system
- Motion Induced Interruptions (MII)
- Motion Sickness Incidence (MSI)

Table 2: Motion assessment variables for any chosen position

3. **Define acceptable limits to the motions**

For each of the critical motion measurements set a threshold or criteria for each. The criteria may be based on a maximum value, a probability of occurrence or a number of occurrences. A selection of acceptable motion criteria from a number of sources has been given in Table 3.

Description	Criteria	Comment	Reference
MSI: (vertical acceleration)			
Exposure: 1/2 hour	0.1g (RMS)	Equivalent to 10% motion sickness incidence	ISO 2631/3
1 hour	0.075g (RMS)	ratio (MSI) among infrequent travellers	1987 & 1982
2 hour	0.05g (RMS)	of the general public	
8 hour	0.025g (RMS)		
VERTICAL ACCELERATION:			
Simple light work possible	0.275g (RMS)	Most attention devoted to keeping balanced	Connolly 1974
Light manual work possible	0.20g (RMS)	Causes fatigue quickly. Not tolerable for longer periods	Mackay 1978
Tolerable at bridge	0.20g (RMS)		Comstock et al. (1982)
Heavy manual work	0.15g (RMS)	Limit in fishing vessels	
Work of more demanding type	0.10g (RMS)	Long term toerable for crew	Payne 1976
Passengers on a ferry	0.05g (RMS)	Limit for people unused to ship motions	Goto 1983
Passengers on a cruise liner	0.02g (RMS)	Older people. Lower threshold for vomiting to take place	Lawther 1985
ROLL:			
Light manual work			
Demanding work			
Passenger on Ferry	3.0 deg (RMS)	Short route safe footing	Karppinnen 1986
Passengers on a cruise liner			
PITCH:			
Navy crew			
Light manual work			
Demanding Work	1.5 deg (RMS)	Assumed limit for ferry passenger	Hosoda 1985
LATERAL ACCELERATION:			
Passenger on a ferry	0.025g (RMS)	1-2 Hz frequency. General public.	
Navy crew	0.050g (RMS)	Non-passenger and navy ships.	
Standing Passenger	0.07g (max.)	99% will keep balance without need of holding	Hoberock 1997
	0.08g (max.)	Elderly person will keep balance while holding	Hoberock 1997
	0.15g (max.)	Average person will keep balance while holding	Hoberock 1997
Seated Passenger	0.15g (max.)	Nervous person will start holding	Hoberock 1997
	0.25g (max.)	Average person maximum load balance when holding	Hoberock 1997
	0.45g (max.)	Person will fall out of seat	Hoberock 1997
MOTION INDUCED INTERUPTIONS (MII):			
	0.1/minute	Possible risk level	Graham, R. (1990)
Passenger ship level	0.5/minute	Probable risk level	"
Navy vessel level	1.5/minute	Serious risk level	
	3.0/minute	Severe risk level	
	5.0/minute	Extreme risk level	

Table 3: Comfort criteria guide levels for passengers and crew (Continued over page)

Description	Criteria	Comment	Reference
SLAMMING:			
Navy vessel	60/hour	At 0.15L	Kehoe (1973)
	20/hour	Bow slamming	Comstock et al. (1982)
	0.03	Maximum probability level	Walden and Grundmann (1985)
Merchant ship	0.03	"	Ochi and Motter (1974)
	0.01	"	Shipbuilding Research Association of Japan (1975)
	0.03 - 0.04	"	Aertssen (1963, 1966, 1968, 1972)
	0.02	"	Yamamoto (1984)
Passenger vessel	<20/hour	Suggested maximum levels	
PROPELLER EMERGENCE:			
Merchant ship	120/hour	Max. acceptable level	Lloyd and Andrew (1977)
Navy vessel	90/hour	Max. acceptable level	
Passenger ships	<60/hour	Suggested maximum levels	
DECK WETNESS:			
Navy vessel	60/hour at FP	Acceptable frequency of occurance	Kehoe (1973)
	36/hour	"	Lloyd and Andrew (1977)
	90/hour	"	Andrew and Lloyd (1981)
	30/hour	"	Comstock et al. (1982)
	0.07	Acceptable probability	Walden and Grundmann (1985)
Merchant ship	0.02	Acceptable probability at FP	Yamamoto (1984)
	0.07	Acceptable probability	Ochi and Motter (1974)
Passenger vessel	<20/hour	Suggested criteria	

Table 3: (Continued) Comfort criteria guide levels for passengers and crew

4. Select a technique to determine the motions

Having set the measurement and the criteria against which to assess the measured quantity, the assessment may be made utilizing a number of techniques with a variety of outcomes. One should not over look the existence of trials data for ships in service in making the assessment.

Assessment Methods:

- Experience;
- Model testing;
- Existing ships through service records or trials data;
- Simulation and/or,
- Assessment against a benchmark.

Considerations in selecting an assessment technique:

- Availability of information;
- Association to other design activities;
- Reliability and accuracy of approach;
- Cost, and
- In absence of any prior information, simulation is typically the most cost effective solution.

Since a ship does not spend all of its time at one speed or at a constant heading in a particular wave height and frequency, the analysis performed in respect of seakeeping may be repeated for each of a variety of conditions or may be completed for a select few cases such a head and beam seas.

Each criterion of importance may be considered in isolation or combined using wave scatter diagrams to account for the proportion of time spent in each condition. This approach yields an overall operability, usually expressed as a percentage.

The typical stages of analysis leading to the assessment of operability against seakeeping criteria are described as follows.

- Transfer function
  - Ship response to regular waves. No consideration is given to the irregular wave environment.
- Ship response to irregular waves
  - Response of the vessel considering the distribution of wave energies across the wave spectrum.
- Operational limit boundaries
  - The wave height over which a criterion will be exceeded.
- Operability
  - The overall level of ship capability considering the time spent in each sea condition, heading and speed. This is usually expressed and a percentage figure.

Figure 1 indicated the four stages of analysis.

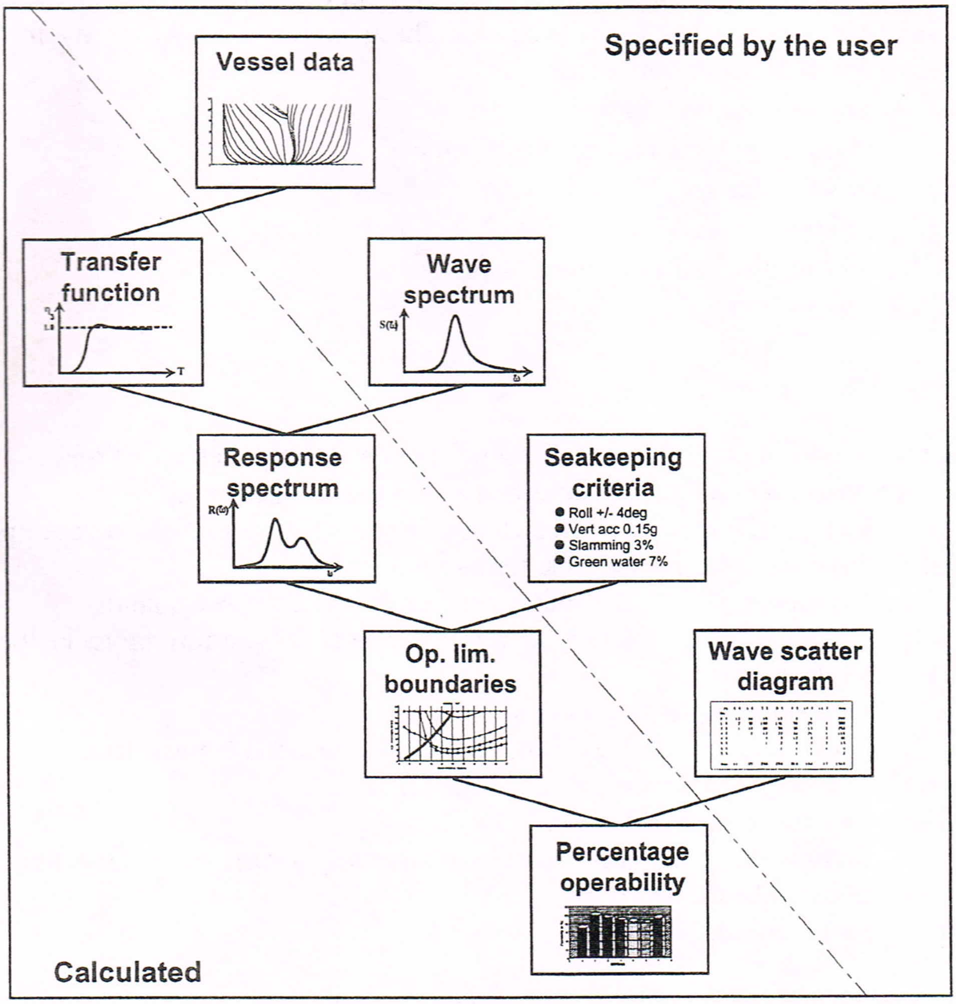

Figure 1: Four Stages of analysis (as defined by Marintek)

If simulation is to be utilized for analysis the simulation tools should be appropriate for the task. The key-differentiating features of the various simulation tools available relates to how they deal numerically with speed and the non-linear terms.

Considerations in selecting a simulation tool should consist of:

- Whether it is an appropriate simulation foundation, high speed or slow speed;
- How it deals with nonlinear response due to wave height;
- Whether control surface simulation is possible and handled appropriately;
- Nonlinear response caused by instantaneous underwater shape changes, and
- Whether it handles the nonlinear responses imposed on a vessel due high forward speed.

5. Assess the motion against the criteria

Comparing the actual measurement to the imposed threshold

6. Act on the outcome

Be realistic, honest and objective in your assessment of the outcome. Change may be necessary to address any unacceptable risks. These outcomes may lead to a following realization or course of action,

- Vessel may not be appropriate;
- One may have to just accept the motions;
- Modify the design;
- Fit an appropriate Motion Control System;
- Review original criteria selected, or
- Change the vessel type.

SUMMARY

A risk minimization process for seakeeping may be summarized as follows:

1. Identify the motions that are important to the vessel operation,
    a. Identify the motion related characteristics that are critical to your operation
2. Define the motions in measurable terms,
    a. Express each of these characteristics as a measurable quantity
    b. Establish the environmental conditions and the motion reference locations on the vessel
3. Define acceptable limits to motions,
    a. Establish a numerical threshold for each motion characteristic
4. Select a technique to determine the motions,
5. Assess the motions against the criteria,
    a. Review the measured or calculated quantity against the established thresholds or operability,
6. Act on the outcome,
    a. Be realistic
    b. Assess options to address any unacceptable risks
7. Repeat the process if necessary until a suitable outcome is achieved.

PAPER TWO

SAILING DINGHY DESIGN AND OPTIMISATION

Presented by Damien Smith , Austal Ships / University of New South Wales, Australia

Paper presented at

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Sailing Dinghy Design and Optimization

Adventures in Budget Boat Design

by Damien Smith

The authors OK dinghy

The OK dinghy is a single handed sailing dinghy. Analytical And experimental techniques were used to find performance improvements. A Velocity Prediction Program (VPP) was written and used to identify the affects of different performance variables. A detailed analysis of the centreboard and rudder was performed, identifying an optimum configuration. A range of possible hull shapes was analyzed by using the VPP to calculate an elapsed time around a standard course. A mast design and analysis spreadsheet was developed in conjunction with wind tunnel tests. Improved mast geometry mast geometry was identified. A spreadsheet was used to design a new mast based to meet the required bend and strength criteria. New designs based on this work were produced and construction of a new boat commenced.

Introduction

Originally designed by Knud Olsen in 1962, the International OK dinghy was conceived as a training boat for the International Finn, the men's Olympic single hander. The original boats were constructed of plywood with unstayed timber masts, features shared with their big brother the Finn. During the history of the class there have been a number of technological improvements.

The OK dinghy is defined by a document known as the "International OK Dinghy Class Rules" [i] and the measurement form that accompanies them. These rules are quite restrictive, they apply maximum and minimum measurements to all of the key variables that the rule makers considered important. Controlled items include weights of hull and mast, hull shape, mast dimensions, boom dimensions, Rudder and Centerboard planform and thickness. Materials are also restricted to limit costs. The tolerances within these rules do however, allow quite different boats to be produced.

In the early 1970's aluminum masts were introduced, rapidly making wooden masts obsolete. The aluminum masts were lighter than their timber predecessors reducing the total weight of the boat and the pitching moment, an important factor in waves. They also were smaller in diameter reducing aerodynamic drag. Finally aluminum masts bend consistently making it easier for the sailmaker to produce a good sail.

In the mid 1970's Bob Miller (Ben Lexan) redesigned the hull shape. Exploiting the tolerances within the class rules, he produced a flatter hull shape with a higher prismatic coefficient. These boats were significantly faster than their predecessors, particularly down wind. Derivatives of these shapes continue to dominate the class.

Experimentation and development of hull, appendages, and rig are important to racing successfully in an OK dinghy. Leading sailors have universally invested considerable time in the development of their equipment. Most of this development work is done on a trial and error basis that eventually may produce good results. However it is costly in terms of time and money. These methods often fail to illuminate the reasons why a particular piece of equipment or combination of equipment is faster. This project aims to produce a fast OK dinghy with less random experimentation than has been used in the past.

Velocity Prediction Programme

Velocity Prediction Programs (VPP) solve the equilibrium equations for the forces to give a velocity for the vessel at a given wind speed and heading. Using a VPP it is possible to quantify the performance change to a design. For this work a velocity prediction program was written in an excel spread sheet using a method given by van Oossannen[ii].

To compare hull designs the time around a typical course was calculated. The OK dinghy at national or international level sails a triangular course. This course consists of three windward legs to broad reaches and a run. A velocity was calculated for the two off wing legs. A Velocity made good was found for the windward legs The total elapsed time of the six nautical mile course was calculated for wind speeds between 6 and 24 knots. Each one of these times wind speeds was weighted according to their probability of occurring. When the elapsed times were added a average elapsed time was found across probability distribution of wind strengths. A typical set of results is shown below.

wind strength (kn)	elapsed time (hr)	weighting	elapsed time x weighting
6	2:02:52	.103	.2109
9	1:41:27	.103	0.1742
12	1:34:12	.206	0.3235
15	1:29:13	.206	0.3064
18	1:25:44	.206	0.2900
21	1:23:24	.103	0.1432
24	1:23:20	.068	0.0945
	Total	1	1:32:34

Typical results from Velocity Prediction Program

The results for the elapsed time around a course of a OK as calculated by the VPP are close to those found in practice, the 1 hr 32 min average elapsed time is very close to the elapsed time seen a major events

A series of runs was done to evaluate the effect of changing various performance variables. The results are shown in the table below

configuration change	elapsed time (hr)	change from base configuration (min)
base	1:32:34	0
rough hull surface finish	1:33:32	+0:58
add 5 KG to the skipper	1:31:57	-0:44
move the skipper 50mm further out board	1:31:42	-1:01
add 5 KG to the hull	1:33:00	+ 0:26
Improved Appendages	1:27:14	-5:19

Predictions of effects of changes made to the base configuration

Use of The Velocity Prediction Program to Find an Improved Hull Shape

The velocity prediction program can be used to compare the performance of different hull shapes. A basis hull is defined and the performance of other hull shapes compared to the basis hull.

The hull shape of the OK dinghy is controlled by the class rules. A hard chine hull shape is specified by four stations and the stem profile. controlled to tolerances of ±10mm. The transverse section curvature of bottom panel is controlled to a maximum . Within these limitations it is possible to produce quite different hull shapes.

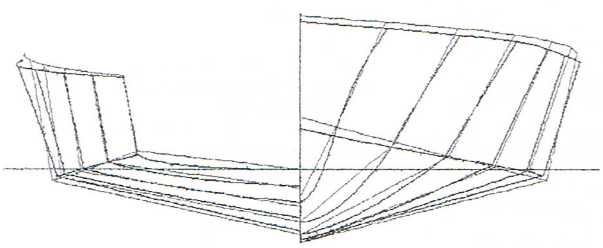
A computer program was used to draw hull shapes that fit within the class rules. Hydrostatic calculations were done on these hull shapes to find the input parameters for the VPP.

The first shape tested was a hull shape in the middle of all of the tolerances this was used as a bench mark for the other hulls in the series. The next hull in the moved all the tolerances to one extreme or the other

The class measurer was interviewed to what trends where apparent amongst the fast hulls. The following observations were made

- A kink in the keel line of hull near the center board case is a common feature.
- Some section curvature at the transom is common
- The shape of the bow is not consistent across all boats, however it does appear to be a significant variable

the lines plan of the benchmark hull superimposed upon the fastest legal hull tested using the VPP is shown below. The benchmark hull is close to the original hull shape designed for the OK dinghy by Knude Olsen.



The fastest hull shape as predicted by the velocity prediction program superimposed on the bench mark hull

The key design trade off in the lines plan was between length and prismatic coefficient. Increasing transverse section curvature of the bottom panel raised the prismatic coefficient but also caused the hull to float higher lifting the bow out of the water reducing the waterline length of the hull. To limit the amount the hull is raised out of the water, the hull around midships was reduced in volume as much as possible. by eliminating the transverse section curvature and raising the keel line. The result is the kink in the keel. A number of different bow shapes seemed to give a similar result.

The final hull is 2:40 minutes faster than bench mark hull around the standard course.

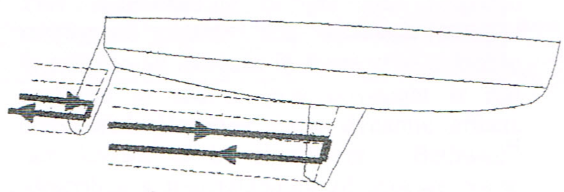
Rudder and centreboard design

The method adopted to analyse the foil system uses discreet horseshoe vortices that allow the induced drag and down wash effects of the centerboard and rudder to be analyzed.

The arrangement of the panels used in the analysis are shown the figure below The effective angle of attack and induced drag is calculated for each panel. The corresponding lift and drag values for each section give the total lift and drag of the center board and rudder combination

The two dimensional aerofoil lift and drag data used for the models was generated by a computer program written by Herppele[iii]. This program uses a conformal mapping method to obtain a velocity distribution around an aerofoil, a boundary layer model which includes a separation bubble model

The method used by the velocity prediction program to calculate the appendage drag assumes the viscous drag coefficient is a function of Reynolds number. However at the Reynolds numbers that a dinghy centreboard is operating this is not assumption is not valid. It is clear that from the graph that surface finish, Reynolds number and local angle of attack are important variables.



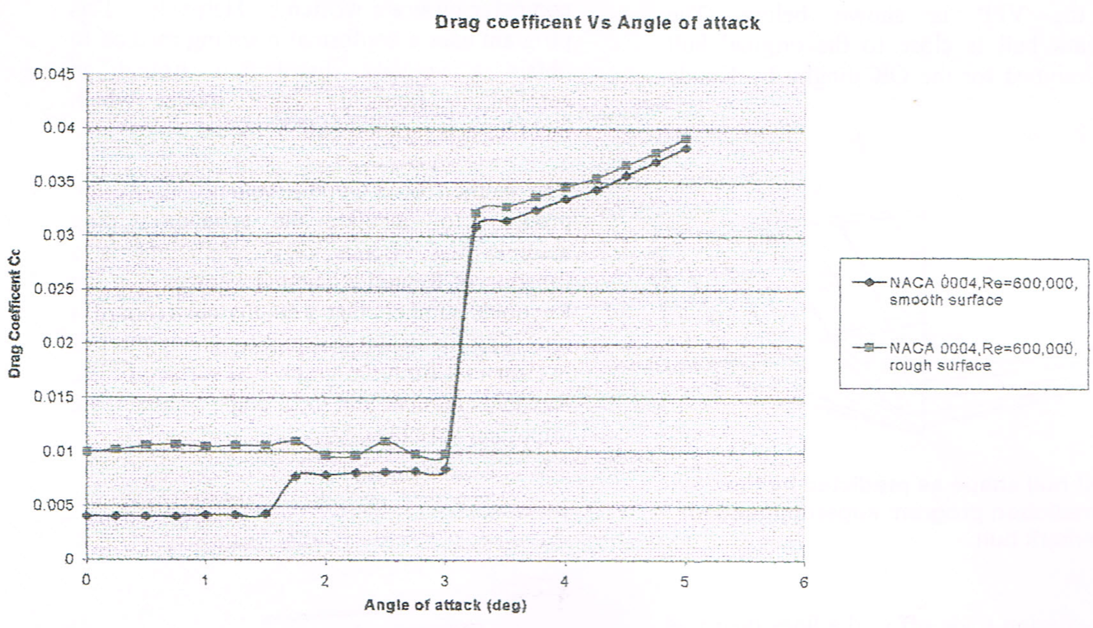
Panels used to model centreboard and rudder

The graph below shows Drag Vs angle of attack for two aerofoils run through this program. Both are NACA 0004 sections at a Reynolds number of 600,000 The only difference between the two runs is specifying a "rough" surface finish instead of smooth surface finish input into the boundary layer variable of the program..

Some features of this graph are worthy of note. Firstly the drag coefficient of the smooth aerofoil is approximately half the drag coefficient of the rough aerofoil below an angle of attack of 1.5 degrees. As the angle of attack is increased a transition to turbulent flow occurs. The drag increases to a value near that of the rough aerofoil. Ideally a lifting surface would operate within this "laminar flow bucket" to achieve minimum drag. As the angle of attack is increased to around 3 degrees, separated flow begins to occur at the trailing edge, with an increase in drag. These points at which the drag increases are important from a design point of view where possible the where possible the appendages should be operating in a laminar flow regime.

configuration. the table below summarizes these results. the difference between these two results is approximately 4 N. The total resistance of an OK dinghy sailing to windward is approximately 95 N, making this drag reduction significant.

All runs, V_B =2.32 ms^{-1} SF= 291 N	Total rudder and centreboad drag (N)	leeway angle β (degrees)
Max area centreboard Minimum area rudder rudder lowered 50mm rudder raked fwd 4.5 deg	21.55	4.54
Minimum area centreboard maximum area rudder rudder raked aft 4.5 deg	25.77	3.84



Comparison of drag of smooth ad rough aerofoil sections

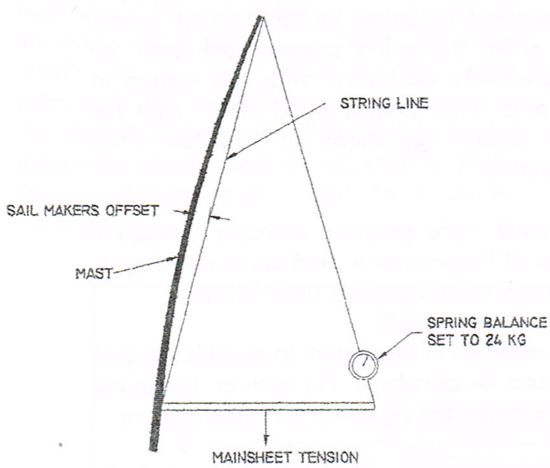
Within the class rules there are tolerances regarding the size shape and position of the appendages each of the variables was independently checked to identify an optimum

Mast Design and Analysis

The mast of an OK dinghy is required to for fill a number of different and sometimes conflicting requirements. The class rules[i] require a mast to have a minimum weight of 8.5 kg and centre of gravity above 2.1 meters above the heel. The class rules also specify geometric restrictions requiring the mast to be straight and restricting the maximum diameter. Sufficient strength is required prevent structural failure.

The bending characteristics of the mast are used to control the sail shape. Sails are cut to match the bending characteristics of a mast. When the curved front edge of the sail, known asthe luff, is placed on a straight mast the excess cloth moves to the body of the sail giving the sail its shape or camber. By increasing or reducing the for and aft bend of the mast the camber of a sail can be changed to suit different wind strengths.

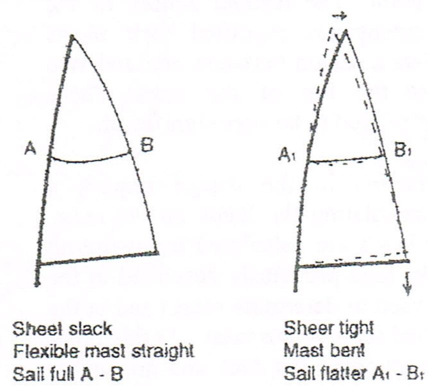
To measure mast bend a standard method has been developed[iv]. The boat is laid on its side with the mast and boom fitted, a line is attached between the end of the boom and the top of the mast. This line is fitted with a spring balance and adjusted to be the same length as the as the sail. Sheet tension is applied until the spring balance shows a tension of 24 kg. The mast offsets are then



Mast bend experiment

measured as shown in the above. The for and aft bend of the mast should fall within the optimum range found by previous experience. A maximum sailmakers offset is typically 130mm.

The optimum sideways bending has not been defined with the precision applied to for and aft mast bend, indeed achieving the correct sideways bend is considered to be one of the "black arts" of OK dinghy sailing. Sideways bend controls the "gust response " of the rig. Gust response being the response of the sail and mast during a sudden increase in wind velocity. The conventional understanding of



Effect of mast bend on sail shape

this mechanism is described by Machaj[v]. The increase in aerodynamic force causes increased deflection of the mast. The top of the sail to twists to leeward, unloading the upper portion of the sail lowering the aerodynamic center of effort. This reduction in centre of effort comes with a large induced drag penalty. When this type of mast was introduced in the 1960's it was possible for very light sailors to be competitive. Heavier sailors then began to use masts that were stiffer sideways, reducing the sail twist and induced drag. The stiffer masts were faster but much harder to control, being less responsive to gusts. these masts are much faster and as a result masts are currently made stiffer in a sideways direction.

This understanding of the gust response mechanism assumes that the mast responds linearly to an increase in force. The dynamic response to sudden gust onslaught is not considered. However these dynamic effects are known to be significant. Bethwait[vi] describes a understanding of gust response that has emerged from experimentation in other classes. In this model the response is purely dynamic requiring the rig to "open" for approximately three quarter seconds giving the sailor, time to react to the gust. My interviews with other OK sailors brought out interesting observations. A number of sailors reported at one time sailing with masts that were very fast and exhibited unusual behavior during gusts. The two common elements in these descriptions were firstly that the mast reacted much faster than normal to a gust and secondly that the top of the mast appeared to flick to leeward in an unusual way with the upper portion of the mast appearing to "hinge" about one point. The leading sailors in the class had extensively modified their masts particularly in a region between one and two meters from the top of the mast. These observations proved to be very significant.

Structural factors in the design require a method of calculating the loads on the mast. For and aft loads are calculated by assuming that the 24 kg load previously described in the experiment used to determine mast bend is the maximum load seen by the mast. At this point the boom is touching the deck and no further deflection is possible, hence no extra force.

The side force cannot be calculated according to normal yacht practice of using a vessels righting moment at 30 degrees of heel. When an OK dinghy sailed down wind the mast rotates allowing a much larger side force, limited by the longitudinal stability of the vessel. In practice stability is not a limiting factor. The model used divided the sail into strips. A maximum design velocity is assumed and a maximum lift coefficient of 1.3 taken from the IMS sail coefficients. The sail force calculated for each strip was divided and 60% applied directly to the mast and the remainder applied to the leach of the sail. Half of the total leech load was applied to the top of the mast and half to the end of the boom. The maximum bending moment due to sideways bending was found to be considerably smaller than the For and aft bending moment.

All the calculation methods used divide the mast into 40 strips. These methods are easily implemented in an excel spreadsheet. Visual Basic was used to automate some of the solution procedures.

Beam theory was used to analyze the mast. The bending moment distribution, calculated from the loads the mast bend experiment was integrated along the length of the mast to give a deflection. The calculated deflection was corrected to give the sail makers offsets.

A sail camber model was incorporated into the mast design spread sheet. Sail camber can be calculated if the difference is known between the distance between the s and c measurements in. The camber is given by Bellair[vii]

$$C=(E/0.027)^{0.5}$$

Where:

C= Sail Camber (%)
E = Excess Cloth (%)

The maximum mast bend offsets are obtained from either, the mast bend experiment or calculated from the spreadsheet. These offsets are corrected by dividing the offset by to give a distance parallel the foot. A base length for the foot component is calculated using similar triangles. These calculations are repeated using a new couthaul position and sheet tension to give a camber distribution

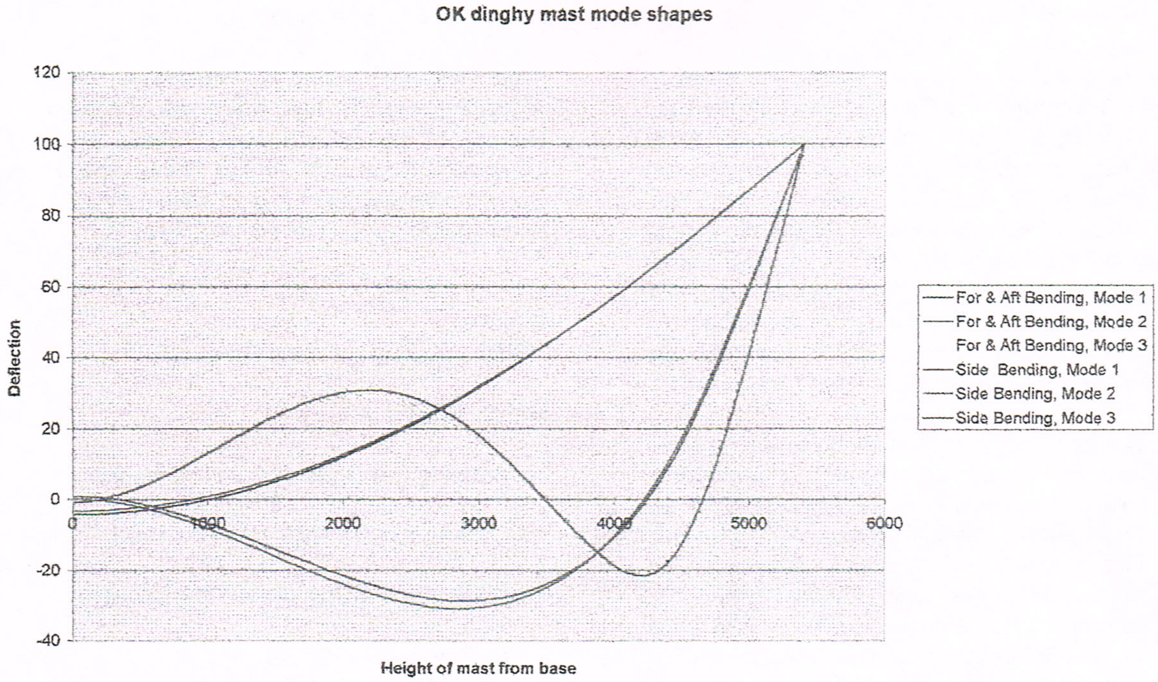
Dynamic gust response

Mykalstead's[viii] method was used to calculate the natural frequency and mode shapes of the mast. This method replaces the continuous beam of the mast with a series of lumped masses and masssless interconnecting beams. The slope, deflection, moment and shear are progressively calculated from one section to the next. This method fitted neatly with the data already generated to calculate mast deflection.

A visual basic program searches through a range of frequencies to find the modal shapes. A typical output graph is show below.

The response of the system to an arbitrary gust can now be calculated. For each of the natural mode shapes the equation of motion is given

The runge kutta method is used to find the response of each mode shape to a force function f(t). The first two mode shapes are summed giving the deflected shape as a function of time.



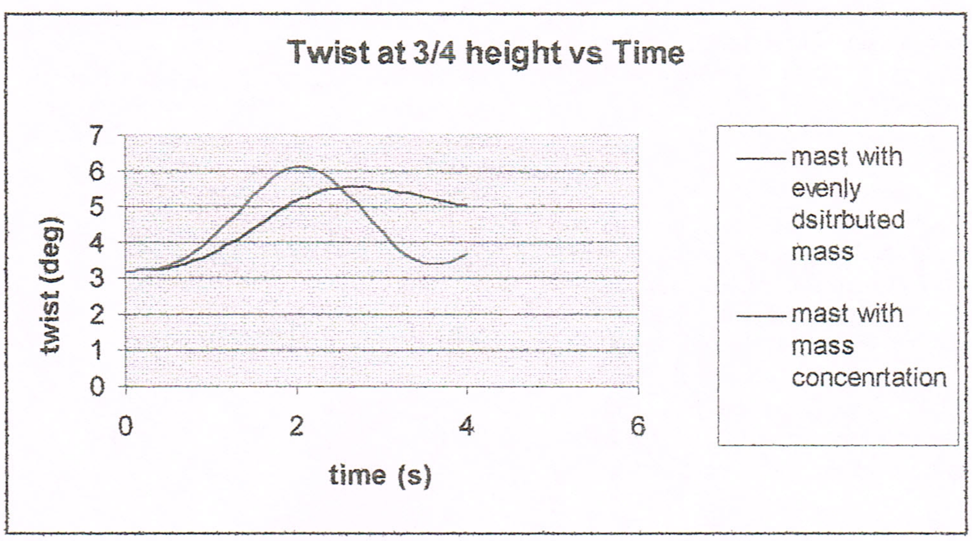
Design Procedure

After establishing analysis methods it became possible to design a new mast. The first stage establishes a suitable for and aft second moment of area distribution using an iterative technique. This distribution is aimed at achieving the suitable mast bend. An initial distribution is assumed and mast bend calculated. The sail luff curve is derived from the calculated mast bend. The "Sail Controls" within the spread sheet are adjusted to prove that the required range of sail shapes can be achieved.

A separate excel spread sheet was used to calculate the section properties of mast cross sections using variables to define the sections. An optimisation routine was set up using the solver function to search a suitable cross section subject to the following goals and constraints.

- minimise mass per unit length.
- Target l/w =1.3
- Target Iyy/Ixx =1.15
- maximum stresses < 0.33 modulus of rupture.

A visual basic loop iterated the solver solution for each section. The results reported back to the main spread sheet. The modal shape and frequency analysis are now performed, the



total mass and centre of gravity are now calculated.

The center of gravity and mass were found to be below the minimums required by the rule. This requires ballast to be added to the mast. Changing the mass distribution of the mast can affect the dynamic response of the mast. Consider the first two mode shapes, the second is fits the observations of leading OK sailors both in terms of the “hinging” of the upper mast and the deflected shape of the sail during gust response. An attempt to stimulate the second modal shape of the mast seem justified.

The frequency response analysis was repeated with the ballast redistributed, concentrating mass at the nodal point of the second Mode. The results of this change were quite dramatic, The second mode became dominant. An animation of the gust response fitted the described fast response exactly. The twist angel at 3/4 height in. concentrating the ballast in the deflection causes a dramatic but brief reduction in the angle of attack of a sail, this gives the sailor time to react to gusts. This is the rig “opening” as described by Bethwate The slower response of the mast without mass concentration in the same gust transmits the additional force to the vessel much faster not giving the sailor time to react. In addition the sail maintains the aerodynamically inefficient twisted shape much longer. In gusty conditions this will lead to significantly increased drag.

The two fastest sailors at the last national champion ships were both using old highly modified masts. One had been cut and sleeved a number of times the other had lrrge amouts of fiber glass added to the mast. Both of these mat were modified at the nodal point of the second vibration mode. These modifications added weght at a point which will stimulate a second mode response.

Conclusions

Using the velocity prediction program it is possible to answer the "how much faster will a new boat go?" The first boat tested uses the base hull and a configuration with the hull appendages in the center of all the tolerances. The elapsed time around a course for this vessel is 1:32:42. The improved hull and appendages give an elapsed time of 1:27:43. If the aerodynamic lift of the rig is increased to simulate the effect of an improved rig the elapsed time is reduced to 1:26:41.

to put these difference in to perspective at state of national level the fleet will finish within ten minutes of the leaders making these small differences significant. While "sailor speed" will always be required it is possible for a slow boat to handicap a sailor out of contention.

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[i] *1999 OK Dinghy Class Rules.* ISAF

March 1999

[ii] *Predicting the Speed of Sailing Yachts* P

van Oossanen SNAME Vol 101 1993

[iii] *Analise an Aerofoil*, Martin Herppele

[cited October 1999] URL:

http//beadec1.ea.bs.dlr.de/airfoil

[iv] Terry Bellair in a letter to Howard Taylor

1977

[v] Aero Hrydronautics of Sailing 1989.

[vi] *High Performance Sailing,* Frank

Bethwaite, Waterline 1993

[vii] *National Ok Dinghy News*, July 1977,

p60

[viii] *Theory of vibration with applications,*

William .T. Tompson Chapman & Hall

1996

PAPER THREE

PROGRESS IN STRIP THEORY PREDICTION OF WAVE LOADINGS AND SHIP MOTIONS

Presented by Jinzhu Xia, University of Western Australia

Paper presented at

Conference

HYDRODYNAMICS WITHOUT INTEGRALS

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Progress in strip-theory prediction of wave loads and ship motions

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Abstract

An overview is given of the development of time-domain non-linear strip-theory models for practical calculation of wave loads and motions of ships. Such theories are particularly useful when a ship is moving in rough sea conditions, where linear theories often fail to provide accurate prediction due to the theoretical limitation. A brief summary is provided of a newly developed strip-theory formulation, including the rational and practical modeling of the hydrodynamic memory effect, slamming and green water on deck. This is followed with the demonstration of numerical results for the widely studied ship form S175 and the comparison with other strip-theory predictions and 3-dimensional computations recently reported by the International Ship and Offshore Structures Congress (ISSC, 2000). Special interest is given to the prediction of non-linearities in wave-induced heave and pitch motions and vertical bending moments of different hull forms. Some state-of-the-art issues are raised: coupled large-amplitude heave, pitch and roll motions, high-speed ship problems, and determination of extreme and fatigue loading on ships.

1. Overview

In the last decades, much effort has been made in the development of 3-D linear and non-linear computational naval hydrodynamics. It is, however, generally agreed that many 3-D computer codes are not yet widely adopted in practice because of long computing time and complexity in application, and also because of their own accuracy problems. On the other hand, 2-D approaches to the prediction of wave-induced loads and ship motions based on strip-theory formulations continue being widely used and under further development. The trend has been to deal with non-linearities based on time-domain modeling. It is expected that research work will continue in this direction and 2-D non-linear codes will be quickly introduced in design practice (ISSC, 2000).

Time-domain models directly extended from the frequency-domain strip theory formulations have been reported quite successful (for example, Meyerhoff and Schlachter, 1980; Yamamoto, Fujino and Fukasawa, 1980; Fujino and Yoon, 1986; Soares, 1989; Chen and Shen, 1990; Petersen, 1992). Hydrodynamic memory effects are neglected in these semi-empirical time-domain models as the hydrodynamic coefficients in the equations of motion are derived for a specific frequency. This is obviously a theoretical weakness when simulations are performed in irregular waves.

Partly nonlinear strip theory models based on time-domain potential flow representation or the Fourier transform of frequency-dependent transfer functions have been developed by, for example, Xia et al. (1987), Xia and Wang (1997), de Kat and Paulling (1989) and Fonseca and Guedes Soares (1998). The fluid action consists of linear and nonlinear parts. The linear fluid forces are expressed by a time convolution as discussed by Cummins (1962), the nonlinear hydrostatic restoring force and the Froude-Krylov force are calculated accurately, and the slamming action can be included using momentum-slamming theory. These approaches are theoretically more consistent and have been verified with good agreement between predictions and the experimental results from ship model tests (see e.g., Xia and Wang, 1997). However, the computation of time convolution limits the practical applications, and it is difficult to extend the theory to account for non-linear memory effect.

As discussed by Tick (1959), because of the free surface wave motion, linear time-domain ship-motion equations are integral equations, and set forth some conditions under which they may be approximated by higher order differential equations with constant coefficients. With such approximation, the memory effect can be modeled with very small computational effort. In addition, it provides a straightforward empirical procedure for a non-linear time-domain generalization. Soeding (1982) proposed a strip theory representation to relate the relative ship-wave motion and the dynamic fluid force in the time domain by a higher order differential equation. The theory has been used by Schlachter (1989), Jensen et al. (1991) and Wang and Xia (1992) to predict vertical ship motions and wave loads. The numerical calculation showed that there are significant discrepancies between its linear simulation and other strip theory solutions when the ship moves with mean forward speed. As the higher order derivatives are taken as total derivatives, it does not coincide with any of the usual linear strip theory formulations. Also, its non-linear generalization does not seem to predict reasonably well the non-linear behavior of ship motions with forward speed (Wang, 1992).

In order to accurately predict both wave-induced rigid-body motions and structural loads in rough seas, a rational time-domain strip method without convolution was developed by Xia, Wang and Jensen (1998) for the vertical-plane problems. The objective was that the linear solution of such a theory should be basically equal to the relevant frequency-domain (e.g., Salvesen et al., 1970) and time-domain integral-equation solutions, while its non-linear extension is able to give reasonable predictions of the non-linear behavior of both rigid motions and structural loads for ships moving in large-amplitude waves. A higher-order ordinary differential equation was used to approximate the hydrodynamic memory effect due to the free surface wave motion. The hydrodynamic and restoring forces were estimated exactly over the instantaneous wetted surface. The 'momentum slamming' force was automatically obtained in the formulation. The fluid force expression was coupled with the structure represented as a Timoshenko beam to form a hydroelasticity theory. By specifying wave amplitudes, non-linear frequency response functions were presented for the S175 Containership in head seas, including the heave and pitch motions, bow acceleration and sagging/hogging bending moments. Two different bow shapes of the ship were considered to demonstrate the relationship between the bow flare of ships and the non-linearity of the responses. The predicted results were compared with available experimental data from the elastic model test made by Watanabe, Ueno and Sawada (1989) and the experimental investigation by O'Dea, Powers and Zselecsky (1992). Very good agreements were obtained between the predictions and the measurements for wave-induced rigid-body motions and bending moments. The application of the theoretical model proposed by Xia, Wang and Jensen (1998) to several ships traveling in different wave headings has been reported by Wang et al. (2000).

Recently, computations have also been made for high-speed vessels by, for example, Wu and Moan (1996) and Hermundstad et al. (1999) based on the so-called 2.5-D theory developed by Faltinsen and Zhao (1991). Fontaine and Tulin (1998) give a history of the method that they call 2D+t to solve the non-linear problem in the cross flow plane and pseudo-time step the solution in the downstream direction. Generally speaking, hydrodynamics of high-speed ships is still a very challenging area. It may be agreed that there is not a high-speed ship theory that works as mature and reliable as conventional strip theories for conventional ships.

In what follows, a brief theoretical introduction is given to the non-linear time-domain strip-theory model developed by Xia, Wang and Jensen (1998). The modeling of the longitudinal distribution of vertical green water load is also discussed.

2. Recent formulation

Wave and slamming loads

According to Xia, Wang and Jensen (1998), the non-linear time-domain hydrodynamic force $F(x, t)$ at the longitudinal position x on the hull may be expressed by

$$\begin{cases} F(x,t) = \dfrac{DI}{Dt} \\ \sum_{j=0}^{J} \left(B_j I - A_j \dfrac{D\bar{z}}{Dt} \right)^{(j+1)} = 0 \end{cases} \tag{1}$$

where I in represents both the impulsive and memory effects in the hydrodynamic momentum; D/Dt is the total derivative with respect to time t, $\frac{D}{Dt} = \frac{\partial}{\partial t} - U\frac{\partial}{\partial x}$, with U being the forward speed of the ship; $()^{(j)} = \frac{\partial^j}{\partial t^j}$; $A_j(x,\bar{z})$ and $B_j(x,\bar{z})$ are the so-called frequency-independent hydrodynamic coefficients derived by a rational approximation from the frequency dependent added-mass and damping coefficients. Furthermore, the relative motion $\bar{z}(x,t) = w(x,t) - \bar{\zeta}(x,t)$, where $w(x,t)$ is the vertical motion of the hull and $\bar{\zeta}(x,t)$ is the wave elevation with Smith correction.

If the frequency-independent hydrodynamic coefficients $A_j(x,\bar{z})$ and $B_j(x,\bar{z})$ are taken as functions of only x, i.e. the change of wetted body surface is neglected, Equation (1) represents a time-domain counterpart of the linear strip theories, for example, Salvesen, Tuck and Faltinsen (1970). Generally, $J = 3$ suffices for most sectional shapes for symmetric ship motion problems.

By integration of the higher order differential equation in Equation (1) and by incorporation of the hydrostatic buoyancy force f_b under the instantaneous wave surface and the green water force f_{gw}, the total non-linear external fluid force $Z(x,t)$ acting on a ship section can be expressed as

$$Z(x,t) = -\overline{m}\frac{D^2\overline{z}}{Dt^2} + U\frac{\partial\overline{m}}{\partial x}\frac{D\overline{z}}{Dt} - \frac{\partial\overline{m}}{\partial\overline{z}}\left(\frac{D\overline{z}}{Dt}\right)^2 - \frac{Dq_J}{Dt} + f_b + f_{gw} \tag{2}$$

where $\overline{m}(x,\overline{z})$ is the added mass of the ship section when the oscillating frequency tends to infinity; $\frac{Dq_J}{Dt}$ accounts for the 'memorial' hydrodynamic effect with q_J governed by the following set of differential equations

$$\begin{cases} \dfrac{\partial q_j(x,t)}{\partial t} = q_{j-1}(x,t) - B_{j-1}\, q_J(x,t) - (\overline{m}B_{j-1} + A_{j-1})\dfrac{D\overline{z}}{Dt} \\ q_0(x,t) = 0 \end{cases} \qquad j = 1, 2, ..., J \tag{3}$$

The third term of $Z(x,t)$ in Equation (2) is the momentum slamming force. It is assumed to be zero when the ship section exits water. The still-water response of the ship due to the difference of the distribution of the weight and the buoyancy forces is ignored in the calculations.

Green water loads

A brief introduction to the formulation of the green water sectional force f_{gw} (in Equation 2) is given below, whereas a detailed derivation can be found in Wang, Jensen and Xia (1998) and Wang (2000).

The vertical load f_{gw} per unit length due to green water on deck in a longitudinal position x and at a time t is taken to be

$$f_{gw}(x,t) = -gm_{gw}(x,t) - \frac{D}{Dt}\left[m_{gw}(x,t)\frac{Dz_e}{Dt}\right] \tag{4}$$

directed positively upwards. Here $z_e(x,t)$ is defined as the effective relative motion and m_{gw} denotes the instantaneous mass per unit length of green water.

The effective relative motion $z_e(x,t)$ is taken to be a function of the nominal relative motion $z_n(x,t) = w(x,t) - \zeta(x,t)$ based on the undisturbed wave elevation $\zeta(x, t)$,

$$z_e(x,t) = \begin{cases} (2.1 - C_s)z_n(x,t); & \text{Deck emerged} \\ (3 - 2C_s)z_n(x,t); & \text{Deck submerged} \end{cases} \tag{5}$$

Here C_s is the Smith correction factor for the instantaneous wetted body sections.

Define an effective water height on deck, $h_e(x,t) = -z_e(x,t) - D_f(x)$, $D_f(x)$ being the freeboard. The green water mass is taken to be proportional to the effective water height on the deck:

$$m_{gw}(x,t) = \rho\, B_e(x)\, h_e(x,t) \tag{6}$$

$B_e(x)$ is an effective breadth of the green water. In the present study $B_e(x)$ is taken to be half the sectional breadth, $B_d(x)$, of the deck, i.e., $B_e(x) = 0.5\, B_d(x)$.

Equation 4 is of the same form as suggested by Buchner (1994, 1995), except that the forward speed effect is included in the definition of velocities and accelerations. The present approach simply treats the green water load in the same way as the added mass of water for a submerged section. The change in wave profile due to the bow and the flow of water on the deck is accounted for by using $z_e(x,t)$, instead of $z_n(x,t)$. The Smith correction is introduced because it has been successfully used in the strip theories to account for the diffraction effect of the incident waves, and gives a plausible variation of z_e with the geometry of the submerged part of the section, the wave elevation and the frequency. For instance, an increasing bow flare will increase C_s and thus decrease z_e. When the wavelength is long or the wave frequency is small, the effective relative motion is close to the nominal relative motion, which indicates a physically rational asymptotic behavior of the dynamic wave deformation. Equation 5 yields a rather good agreement with the measurements over the whole frequency range, whereas the nominal relative motions z_n deviate quite strongly from the measurements. In a stochastic sea an average value C_s is applied with the individual wave amplitudes as weight factors. Numerical results for the relative motion in stochastic seaways and comparison with experimental results can be found in Wang et al. (1998).

3. Benchmark study

As discussed in Section 1 of this paper, various modifications of the linear strip theory approach have been suggested in the past to account for non-linear effects occurring when a ship is sailing in moderate or rough seas. Especially, a substantial number of successful results for the bending moment in ships in large-amplitude waves have been reported. In the following, a systematic validation of some of the non-linear seakeeping methods are carried out in order to demonstrate the consistency of the methods in the prediction of wave loads, rigid-body motions and structural responses. The methods chosen are those directly available for the ISSC (2000) committee members and other formulations might be just as appropriate. Detailed discussions may be found in ISSC (2000) and Wang et al. (2000).

The calculated results are compared with the experiments on the S175 container ship performed by O'Dea et al. (1992), Watanabe et al. (1989) and Chen *et al* (1999), in order to demonstrate the non-linear behavior on rigid-body motion and structural response. The main particulars and body-plan of the S175 are given in Table 1 and Figure 1. Hereafter, a is defined as wave amplitude and subscript a represents amplitude of the full. The heave, pitch and acceleration amplitudes are defined as half the peak-to-peak value.

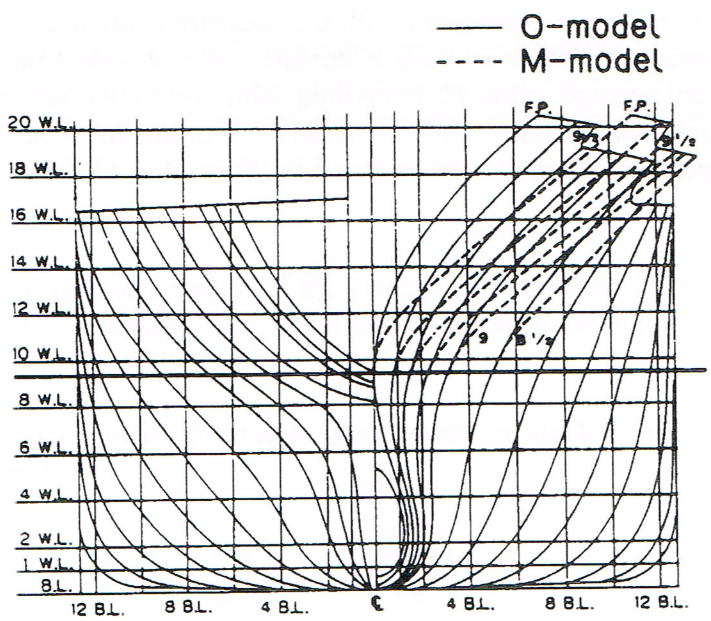

Figure 1: Body-plan of the original (solid lines) and modified (dashed lines) S175 container ship.

Table 1: Main particulars of the S175 container ship

Length between perpendiculars (*L*)	175 m
Breadth moulded (*B*)	25.4 m
Depth moulded (*D*)	15.4 m
Draft (*T*)	9.5 m
Displacement	24742 ton
LCG aft midship	2.5 m
Longitudinal radius of gyration	0.236 L
Two-node frequency	1.6 Hz
Structural damping (log)	0.051
Block coefficient (C_B)	0.572
Midship section coefficient (C_M)	0.98

Non-linear seakeeping and wave load methods applied

The non-linear seakeeping and wave load codes used in the comparisons are:

SHIPSTAR (Xia, Wang and Jensen, 1998) is a non-linear hydro-elastic strip-theory code in the time-domain. The hydrodynamic memory effect due to the free surface is approximated by a set of higher order ordinary differential equations. The hydrodynamic inertial and restoring forces are estimated exactly over the instantaneous wetted surface. The non-linear memory effect and the 'momentum slamming' force are automatically included. Green water loads can be included by a momentum approach (Wang *et al,* 1998).

THATS (Xia and Wang, 1997) is a non-linear hydro-elastic strip-theory code in the time-domain. The linear fluid forces are expressed by a time convolution, the non-linear hydrostatic restoring force and the Froude-Krylov force are calculated accurately, and the slamming action is included using momentum slamming theory.

SOST (Jensen and Pedersen, 1979) is a second-order strip-theory code formulated in the frequency domain. The theory is based on a perturbation expression of the hydrodynamic and the hydrostatic coefficients around the still water line and includes the incoming pressure field from Stokes' second order waves.

SRSLAM (Watanabe and Sawada, 1986), developed by the SR194 committee of the Japan Shipbuilding Research Association (JSRA), is based on a non-linear time-domain model directly extended from a frequency-domain strip-theory formulation, Yamamoto *et al* (1980).

NORS (Chen and Shen, 1990) is a non-linear time-domain method directly extended from a frequency-domain strip theory formulation. The hydrodynamic coefficients in the equations of motion are derived for a specific frequency, also in irregular waves. The hydrodynamic memory effects are neglected.

IST (Fonseca and Soares, 1998) is a non-linear time-domain strip theory code. The hydrodynamic memory effects are expressed by a time convolution, the non-linear hydrostatic restoring force and the Froude-Krylov force are calculated accurately over the instantaneous wetted surface of the ship hull.

LAMP-4 (Lin, Meinhold, Salvesen and Yue, 1994) is a 3D non-linear code based on time-domain Green function calculation. The LAMP-4 results included here are taken from Shin *et al* (1997).

SWAN-DNV (Adegeest *et al,* 1998) is a non-linear 3D Rankine source method.

An overview of the non-linear strip theory methods applied is given in Table 2. It should be mentioned that somewhat similar comparisons are available in the literature, see e.g. Watanabe and Guedes Soares (1999), providing in principle the same accuracy as found in the ISSC (2000) report.

Table 2: Non-linear strip theory methods applied in the study

Methods	Time-domain SHIPSTAR	Time-domain THATS	2^{nd}-order frequency-domain SOST	Time-domain SRSLAM	Time-domain NORS	Time-domain IST
Non-linear restoring and Froude-Krylov	Yes	Yes	Yes	Yes	Yes	Yes
Smith correction	Yes	Yes	Yes	Yes	Yes	No
Non-linear added mass and damping	Yes, by higher-order differential equations	No	Yes, by second order expansions	Yes, but with fixed frequency	Yes, but with fixed frequency	No
Slamming loads by momentum slamming	Yes	Yes	Yes	Yes	Yes	No
Green water on deck	Yes, by momentum approach	No	No	Yes, by static water height	Yes, by momentum approach	Yes, by static water height
Steady state forward speed potential included	No	No	No	No	No	No
Degrees-of-freedom	Vertical, elastic hull	Vertical, elastic hull	Vertical, elastic hull	Vertical, elastic hull	Vertical, elastic hull	Vertical, rigid hull
Computational efficiency	Fast	Fast	Very fast	Fast	Fast	Fast
Viscous effects	No	No	No	No	No	No

Comparison with the experimental results for ship motions

The experimental results obtained by O'Dea *et al* (1992) have been widely used as a database for validating non-linear seakeeping codes by several investigators and are also applied in present report. The model tests are performed for the S175 container ship. The body plan and the main particulars of the ship are given in Figure 1 and Table 1, identical to those also used in Watanabe *et al* (1989). Only head sea is considered. In the calculations, the still water response of the ship due to the difference of the distribution of the weight and buoyancy forces is omitted.

The references corresponding to the non-linear seakeeping codes in the comparison are as follows:

SHIPSTAR (Xia et al., 1998)
THATS (Xia et al., 1997)
SOST (Jensen et al., 1979)
SRSLAM (Watanabe et al., 1986)
IST (Fonseca et al., 1998)
NORS (Chen et al., 1990)
LAMP-4 (Lin et al., 1994) ◆
SWAN-DNV (Adegeest et al., 1998) ●
Exp (ODea et al., 1992) ◇

Figures 2 and 3 compare the first harmonics of heave and pitch motion of the original S175 container ship at three wavelengths, λ/L =1.4, 1.2, 1.0, and two Froude numbers, Fn= 0.2 and 0.275, whereas Figure 4 presents the bow acceleration under the same conditions. Both magnitude and phases (defined as leads relative to the wave elevation at the LCG, with wave, heave and acceleration defined as positive upward and pitch defined as positive bow down) are compared. Heave is defined as the vertical motion of the centre of gravity except for the SWAN-DNV results where it refers to the midship position. However, as LCG is only 1.4 per cent abaft amidships this will only marginally influence the results. It is seen that some of the non-linear strip theory codes, for example, SHIPSTAR and SRSLAM, are able to predict the variation in ship motion with wave slope *ka*. The exceptions are the second order procedure (SOST), which by nature cannot predict any change in the first harmonics and the procedure IST, yielding some peculiar results for high wave slopes. The limited amount of published results for the LAMP-4 code also fit well to the experimental results. The S175 container ship has a rather low freeboard and hence the relative bow motion exceeds the freeboard even for relatively low wave slopes. This behavior is also observed in experiments, see e.g. Watanabe *et al* (1989) and Chen *et al* (1999). Normally, the effect of the green water load on the global response is found to be small, O'dea and Walden (1984), Wang *et al* (1998), but for the SWAN-DNV calculations a very significant influence is observed. The SWAN-DNV results for wave bending moment are completely changed by the water on deck, resulting in a nearly vanishing of the wave-induced sagging bending moment for moderate wave slopes. Clearly, the load calculation due to shipping green water requires additional research in order to capture the pertinent details of this very complex flow pattern.

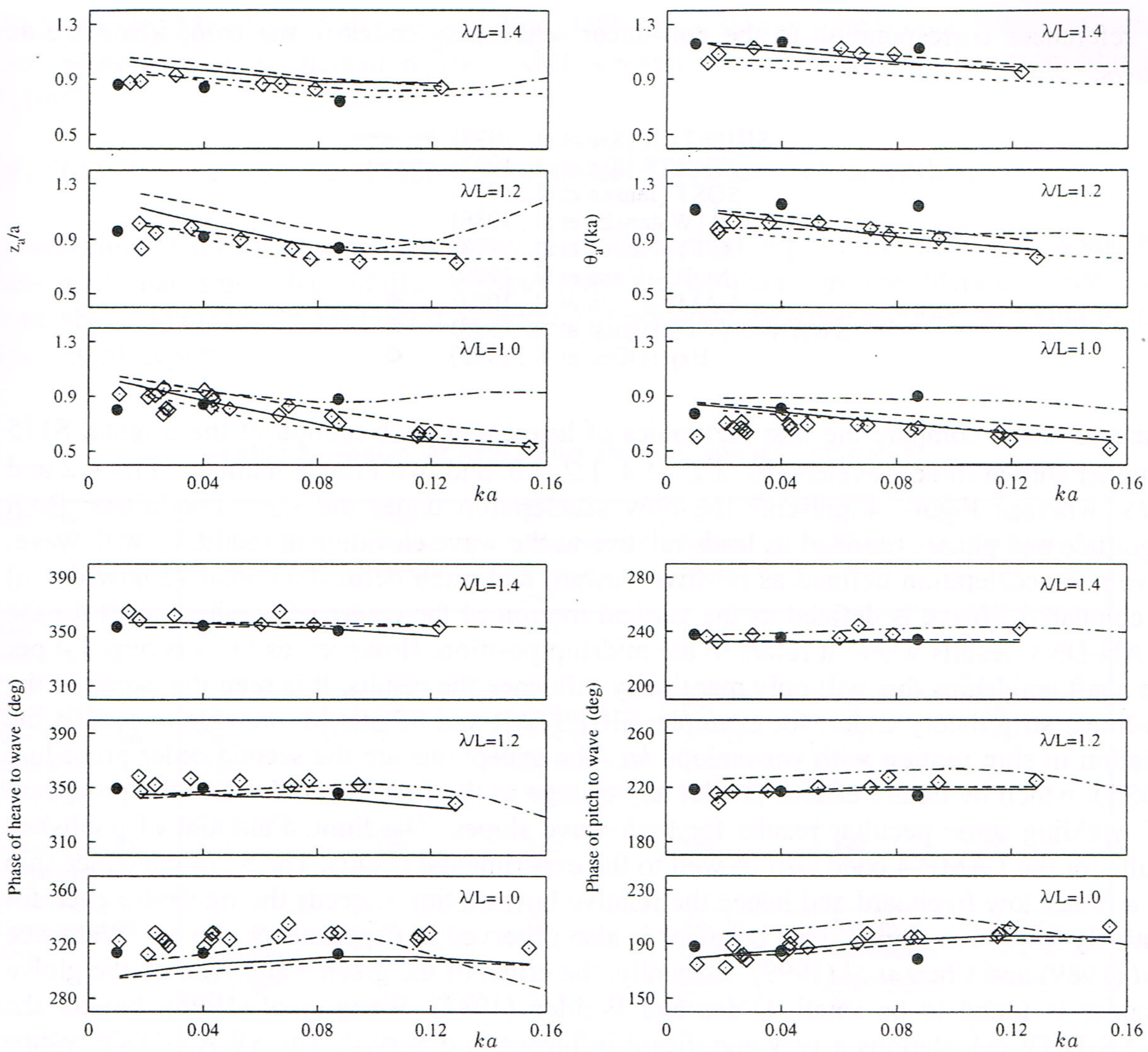

Figure 2: Comparison of the calculated and the experimental (hollow square points, O'Dea et al., 1992) magnitudes (above) and phases (below) of the heave (left) and pitch (right) of the original S175 container ship with respect to wave steepness, Fn=0.2. Solid lines for the present method, solid circle points for SWAN-DNV, dash lines for the partly non-linear simulation (Xia and Wang, 1997) and other strip methods (ISSC, 2000).

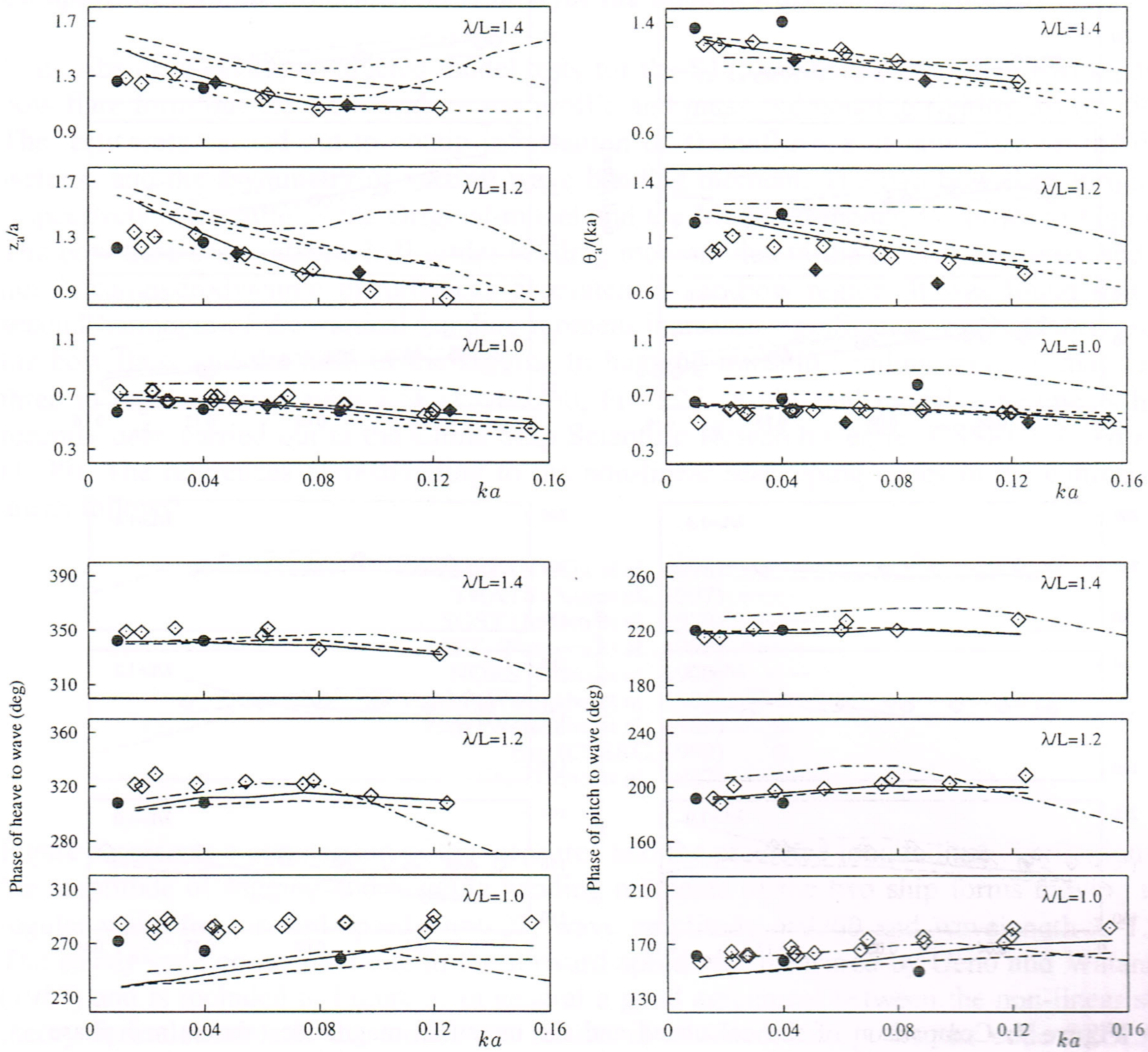

Figure 3: Comparison of the calculated and the experimental (hollow square points, O'Dea et al., 1992) magnitudes (above) and phases (below) of the heave (left) and pitch (right) of the original S175 container ship with respect to wave steepness, Fn=0.275. Solid lines for the present method, solid circle points for SWAN-DNV, solid square points for LAMP-4, dash lines for the partly non-linear simulation (Xia and Wang, 1997) and other strip methods (ISSC, 2000).

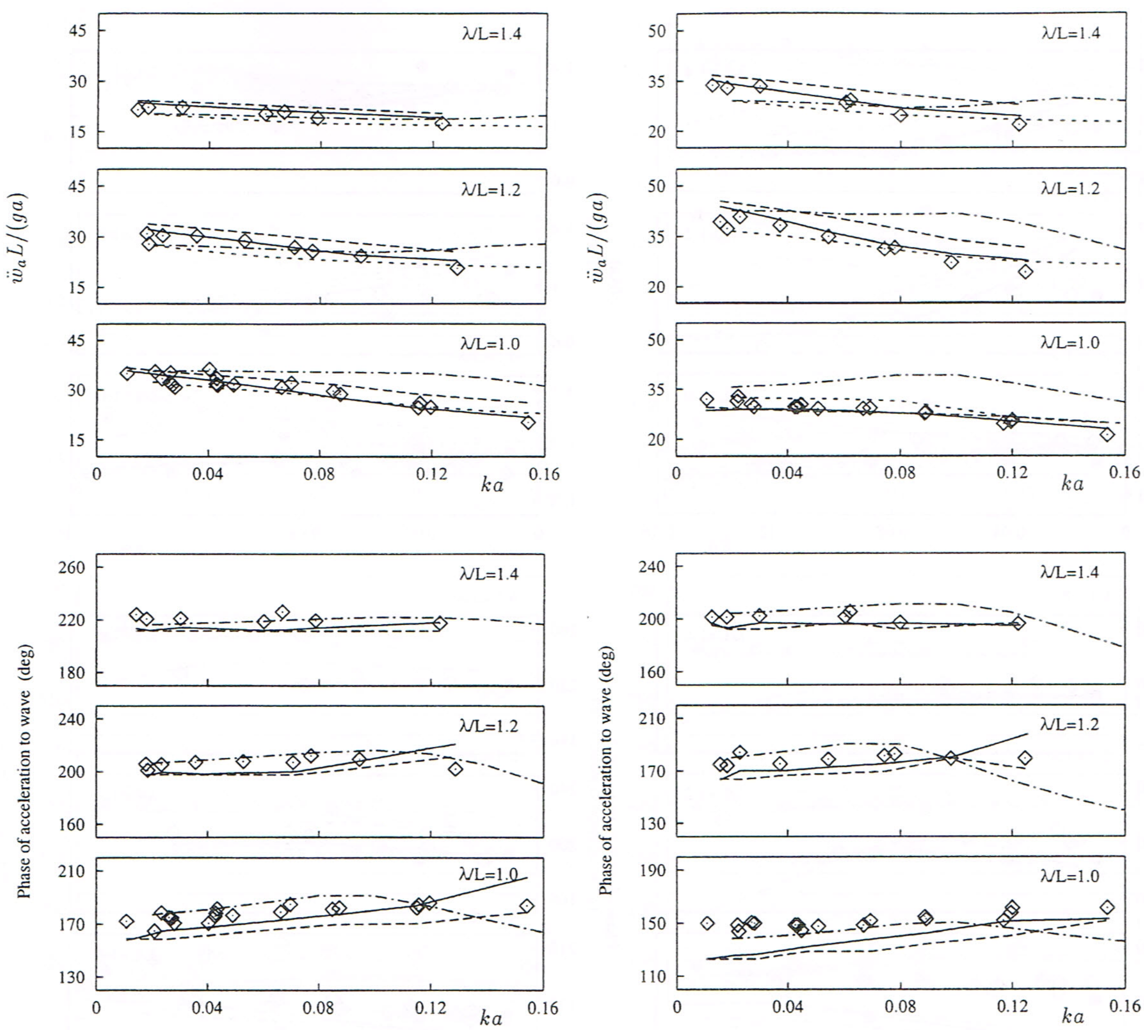

Figure 4: Comparison of the calculated and the measured magnitude (above) and phase (below) of the bow acceleration (15%L aft FP) of the original S175 container ship with respect to wave slope. Fn=0.2 (left); Fn=0.275 (right).

Comparison with the experimental results for the vertical wave bending moment

Watanabe *et al* (1989) conducted model tests for the S175 container ship using two kinds of bow flare form but the same underwater profile and mass and bending rigidity distributions. The tests were carried out to obtain information on the influence of bow flare on the deck wetness and the asymmetry of vertical wave bending moment. The two bow flare forms are respectively referred to as the Original-model and the Modified-model, as shown in Figure 1. The bow flare shape affects hull girder bending moment due to the reserve buoyancy and the non-linear hydrodynamic disturbances generated in the bow region. It was found that the second harmonic of the vertical bending moment increases significantly with the increase of the bow flare, and the ratio of the sagging to hogging midship bending moment may reach three in regular waves with λ=1.2*L*, *a*=*L*/60, Fn=0.25. Additional model experiments have recently been carried out at the China Ship Scientific Research Centre (CSSRC), Chen *et al* (1999). The references corresponding to the non-linear seakeeping codes in the comparison are as follows:

SHIPSTAR (Xia et al., 1998) ————
THATS (Xia et al., 1997) - - - - -
SOST (Jensen et al., 1979) ············
IST (Fonseca et al., 1998) —·—·—·
NORS (Chen et al., 1990) -- -- -- ·
SRSLAM (Watanabe et al., 1986) - - - - - -
Exp (Watanabe et al., 1989) ◇
Exp (CSSRC, 1999) ○
Exp (Ueno et al., 1987) —×—

Figure 5 presents a comparison of the measured and the predicted longitudinal distributions of the amplitude of sagging and hogging bending moments of the two ship forms moving in a regular wave for forward speed Fn=0.25, wave amplitude *a*=*L*/60 and wavelength λ=1.2*L*. The steady state component due to the forward speed was measured by Ueno and Watanabe (1987) and is included in Figure 5. In general a good agreement between the non-linear strip theory formulations and the model test results is obtained, except for the code SRSLAM which seems to over-predict the hogging moment. Clearly the wave-induced sagging bending moment depends rather strongly on the degree of bow flare.

Finally, Figure 6 gives the comparison of the experimental and the predicted significant values of sagging and hogging bending moments, M_s, of the original and the modified S175 container ship moving in irregular waves for Fn=0.25. The significant values are defined as the average of the one-third highest peaks in time history. A very good agreement is seen except for SRSLAM, but it should also be recognized that this comparison does not tell much about the accuracy of the extreme values. A discussion of this topic is given in the ISSC Committee VI.1 report (ISSC, 2000).

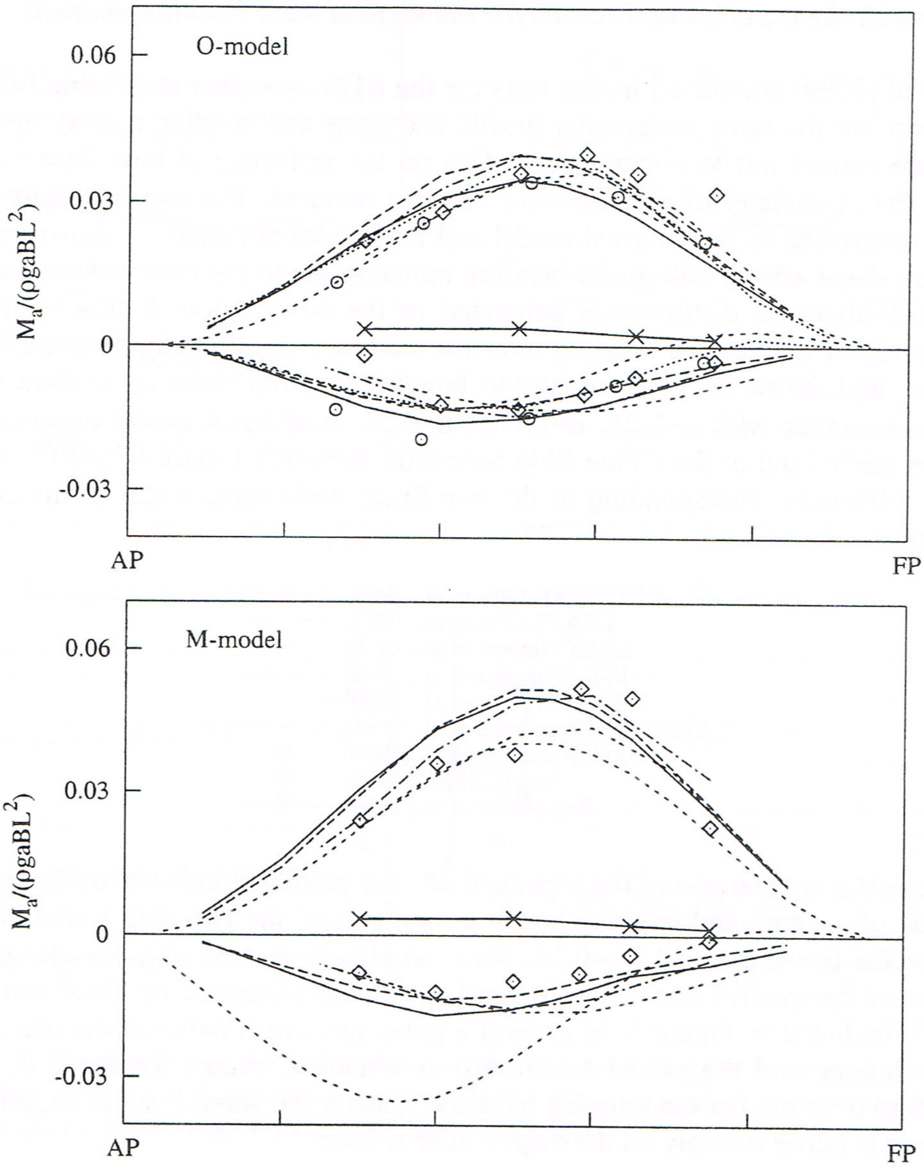

Figure 5: Non-linear sagging (positive) and hogging (negative) bending moments of the original and the modified S175 container ship moving in the regular wave, λ=1.2L, a=L/60; Fn=0.25. O-model (above) and M-model (below).

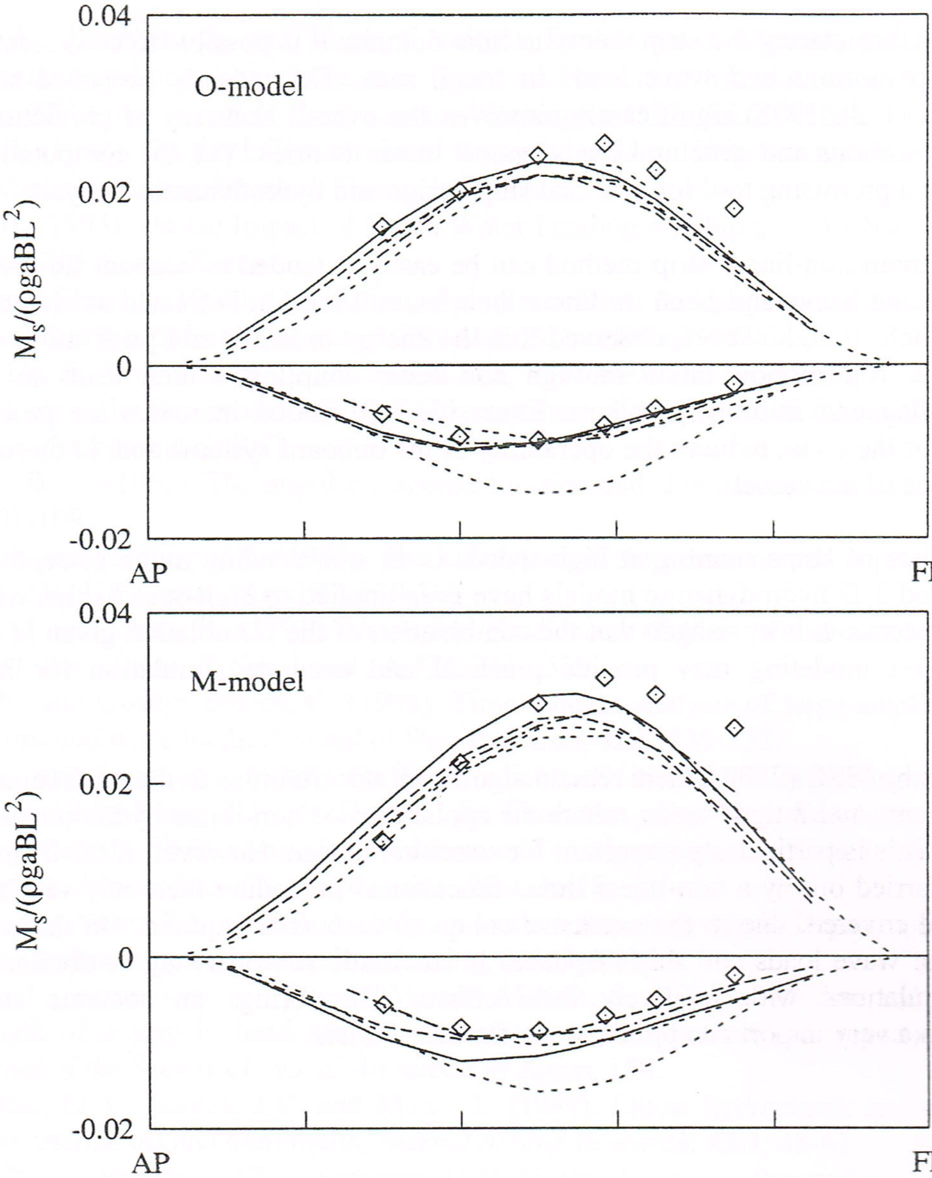

Figure 6: Comparison of the measured and calculated longitudinal distribution of the non-dimensional significant sagging (positive) and hogging (negative) bending moments for the original and the modified S175 container ship moving in irregular waves, H_s=L/21, T_0=11.1sec; Fn=0.25. O-model (above) and M-model (below).

4. Concluding remarks

By rationally formulating the strip theory in time-domain, it is possible to consistently predict accurate ship motions and wave loads in rough seas. The recently proposed strip-theory method (Xia et al., 1998) significantly improves the overall accuracy of prediction of both heave, pitch motions and structural loads against linear theories. Yet, the computation is very fast, yielding a promising tool for practical ship design and hydrodynamic analysis.

The time-domain non-linear strip method can be easily extended to account for the coupling between roll and heave and pitch. In linear theories, roll motion is treated as uncoupled with heave and pitch. It is, however, observed that the energy in heave and pitch motions may be transferred to roll motion mode through non-linear coupling, which leads to excessive resonant rolling and stability problems. Excessive roll motion increases the probability of seasickness of the crew, reduces the operability of the onboard systems and, in the worst case, causes capsize of the vessel.

Hydrodynamics of ships running at high-speeds is an area needing much in-depth research. Many 2-D and 3-D hydrodynamic models have been applied to high-speed ships with certain degrees of success. It is envisaged that the combination of the formulation given in this paper with the 2D+t modeling may provide practical and accurate simulation for high-speed monohulls.

As discussed by ISSC (2000), there remain significant uncertainties in the prediction of wave-induced extreme and fatigue loads, where the application of non-linear hydrodynamic tools is a necessity. This is particularly important for structural design. However, if the hydrodynamic analysis is carried out by a non-linear three-dimensional procedure then only very short time series can be covered, due to the extensive computer resources required. On the other hand, time series of wave loads and ship responses in stochastic seaways may be obtained by strip method simulations with relatively little efforts. This brings an accurate strip-theory simulation to a very important application in the near future.

References

Adeegest, L., Braathen, A. and Vada, T. (1998). Evaluation of Methods for Estimation of Extreme Non-linear Ship Responses Based on Numerical Simulations and Model Tests, *Proc. 22nd Symp. Naval Hydrodynamics*, Washington D.C. Vol.1, 70-84.

Buchner, B. (1994). On the Effect of Green Water Impacts on Ship Safety, *Proc. Inter. Symp. on Ship and Marine Research*, Rome, Italy, Vol. 2.

Buchner, B. (1995). On the Impact of Green Water Loading on Ship and Offshore Unit Design, *PRADS'95*, Seoul, Korea.

Chen, C. and Shen, J. (1990). Calculation of ship non-linear bending moment in regular waves in time domain, *Selected papers of CSNAME*, **5**, (published in China).

Chen, R., Du, S., Wu, Y., Lin, J., Hu, J. and Yue, Y. (1999). The experiment on extreme wave loads on a flexible model of the S175 container ship, Report of *China Ship Scientific Research Centre* (CSSRC).

Cummins, W. E. (1962). The impulse response function and ship motions, *Schiffstechnik*, Band **9**, 101-109.

de Kat, J.O. and Paulling, J.R. (1989). The simulation of ship motions and capsizing in severe seas, *SNAME Transactions*, **97**, 139-168.

Faltinsen, O. and Zhao, R. (1991). Numerical Predictions of Ship Motions at High Forward Speed, *Phil. Trans. R. Soc. Lond.*, **A 334**, 241-252.

Fonseca, N. and Guedes Soares, C. (1998). Time-domain analysis of large-amplitude vertical motions and wave loads, *Journal of Ship Research*, **42:2**, 139-153.

Fontaine, E. and Tulin, M.P. (1998). On The Prediction Of Non-linear Free-Surface Flows Past Slender Hulls Using 2D + t Theory: the Evolution of an Idea, *Proceedings RTO AVT Symposium on Fluid Dynamics Problems of Vehicles Operating Near or in the Air-Sea Interface,* Amsterdam, The Netherlands, 26-1-26-10.

Fujino, M. and Yoon, B.S. (1986). A study on loads acting on a ship in large amplitude waves, Naval Architecture and Ocean Engineering, SNAJ, Vol.24.

Fukasawa, T., Yamamoto, Y., Fujino, M. and Motora, S. (1981). Motion and longitudinal strength of a ship in head sea and the effects of non-linearity(4th report)-Experiment. *Journal of the Society of Naval Architects of Japan*, **150**.

Hermundstad, O.A., Aarnes, J.V. and Moan, T. (1999). Linear hydroelastic analysis of high-speed catamarans and monohulls, *Journal of Ship Research*, **43:1**, 48-63.

ISSC (2000). Committee VI.1: Extreme Hull Girder Loading, Proceedings of the 14th International Ship & Offshore Structures Congress, Nagasaki, Japan.

Jensen, J. J. and Pedersen, P. T. (1979). Wave-induced Bending Moments in Ships - a Quadratic Theory, *Trans. RINA*, **121**, 151-165.

Jensen, J.J., Petersen, J.B. and Pedersen, P. T. (1991). Prediction of non-linear wave-induced loads on ships, Dynamics of Marine Vehicles and Structures in Waves, Elsevier Science Publishers, 279-289.

Lin, W.-M., Meinhold, M., Salvesen, N., and Yue, D. (1994). Large-Amplitude Motions and Wave Loads for Ship Design, *Proceedings 20th Symposium on Naval Hydrodynamics*, Santa Barbara, CA, 205-226.

Meyerhoff, W.K. and Schlachter, G. (1980). An approach for the determination of hull girder loads in a seaway including hydrodynamic impacts, *Ocean Engineering*, **7**, 305-326.

O'Dea, J.F., Walden, D.A. (1984). Effect of Bow Shape and Non-Linearities on the Prediction of Large Amplitude Motions and Deck Wetness, *Proc. 15th ONR Symp on Naval Hydrodynamics*, Hamburg.

O'Dea, J., Powers, E. and Zselecsky, J. (1992). Experimental Determination of Non-linearities in Vertical Plane Ship Motions, *Proceedings 19th Symposium on Naval Hydrodynamics*, Seoul, Korea.

Petersen, J.B. (1992). Non-linear strip theories for ship response in waves, Dep. of Ocean Eng., Technical University of Denmark.

Schlachter, G. (1989). Hull girder loads in a seaway including non-linear effects, Schiffstechnik, 36:169-180.

Shin, Y.-S., Chung, J.S., Lin, W.M., Zhang, S., and Engle, A. (1997). Dynamic Loadings for Structural Analysis of Fine Form Container Ship Based on a Non-linear Large Amplitude Motions and Loads Method, SNAME *Transactions*, **105**, 127-154.

Soares, C.G. (1989). Transient response of ship hulls to wave impact, *Int. Shipbuilding Prog.*, **36**, 137-192.

Soeding, H. (1982) Lekstabilitat im seegang, Report 429 of the Institut fur Schiffbau Hamburg.

Tick, J.L. (1959) Differential equations with frequency-dependent coefficients, *J. of Ship Research*, Oct., 45-46.

Ueno, M. and Watanabe, I. (1987). On asymmetry of vertical bending moment on ships, *SNAJ*, **162.**

Yamamoto, Y., Fujino, M. and Fukasawa, T. (1980). Motion and longitudinal strength of a ship in head sea and the effects of non-linearities, *Naval Architecture and Ocean Engineering*, JSNAJ, **18**.

Wang, Z. (1992). Hydroelastic analysis of ships at rough seas, M.Sc. thesis, (in Chinese), *China Ship Scientific Research Center* (CSSRC).

Wang, Z. (2000). Hydroelastic analysis of high speed ships, Ph.D. Thesis, Department of Naval Architecture and Offshore Engineering, Technical University of Denmark

Wang, Z., Jensen, J. J. and Xia, J. (1998). On the Effect of Green Water on Deck on the Wave Bending Moments, *Proc. 7th Int. Symposium on Practical Design of Ships and Mobile Units, PRADS'98*, The Hague.

Wang, Z. and Xia, J. (1992). Differential method for the nonlinear hydroelastic analysis of ships, (in Chinese), Theories and Applications of Computational Mechanics, Science Press, China.

Wang, Z., Xia, J., Jensen, J. J. and Braathen, A. (2000). Prediction of vertical-plane non-linearities in wave loading and ship responses, Proceedings of the 23rd *Symposium of Naval Hydrodynamics*, France.

Watanabe, I. and Guedes Soares, C. (1999). Comparative Study on the Time Domain Analysis of Non-linear Ship Motions and Loads, *Marine structures*, **12:3**,153-170.

Watanabe, I. and Sawada, H. (1986). Effects of elastic responses to the longitudinal bending moment in two-directional irregular waves, *Naval Architecture and Ocean Engineering*, **24**, 91-102.

Watanabe, I., Ueno, M., Sawada, H. (1989). Effects of Bow Flare Shape to the Wave Loads of a Container Ship, *JSNAJ*, **166**, 259-265.

Wu, M.K. and Moan, T. (1996). Linear and non-linear hydroelastic analysis of high-speed vessels, *Journal of Ship Research*, **40:2**, 149-163.

Xia, J. Gu, M. and Wu, Y. (1987). Linear and nonlinear wave-induced responses of travelling elastic ships, (in Chinese), *Ship Behavior Research*, No. 4, CSSRC.

Xia, J. and Wang, Z., (1997). Time-domain hydro-elasticity theory of ships responding to waves, *J. of Ship Research*, **41:4**.

Xia, J., Wang, Z. and Jensen, J.J. (1998). Non-linear Wave Loads and Ship Responses by a Time-Domain Strip Theory, *Marine Structures*, **11:3**,101-123.

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He is now a Research Fellow with the Centre for Oil and Gas Engineering, The University of Western Australia. Before taking this position in February 1999, he was employed by the Danish Maritime Institute as a consulting engineer. Prior to that, he was employed at the Dept. of Naval Architecture and Offshore Engineering, the Technical University of Denmark as a Research Associate and an Assistant Research Professor. He earned a Ph.D. in Naval Architecture and Offshore Engineering in 1994.

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PAPER FOUR

PREDICTION OF CATAMARAN WAVE LOADS

Presented by Stephen Cook,
Austal Ships / Curtin University, Western Australia

Paper presented at

Conference

HYDRODYNAMICS WITHOUT INTEGRALS

2 November 2000, Fremantle, Western Australia

Prediction of Catamarans Wave Loads

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AME CRC, Curtin University.*
Patrick Couser,
AME CRC.**
Kim Klaka,
AME CRC, Centre for Marine Science & Technology, Curtin University.

SUMMARY

Catamaran designs, whilst fundamentally unchanged from their historic predecessors, have developed rapidly over the last decade. As these vessels become larger and faster, accurate prediction of hull loads becomes increasingly important. Unlike more traditional vessels, high speed craft must have efficient, light-weight structures to maximise their payload and/or operating speed; however, the safety and structural integrity of the vessel must not be compromised.

To obtain a better understanding of the behaviour and response of a catamaran in a seaway, an eight metre research catamaran, "Educat", has been built and instrumented with strain gauges and motion sensors. The research and findings to-date are presented in this paper, including: calibration of "Educat"; results of the sea trials and towing tank tests; and correlation with the numerical models.

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Stephen Cook graduated with a Bachelor of Engineering (Naval Architecture) from the Australian Maritime College in 1996. He is currently working for Austal Ships and is completing his Masters at the Curtin University of Technology part-time on the topic of global wave loads and catamarans.

Patrick Couser is currently employed by Formation Design Systems as a Naval Architect / Software Engineer. He graduated from the University of Southampton in 1990 with a masters degree in Naval Architecture; in 1996 he completed his PhD in the field of high-speed catamaran resistance and seakeeping. His current interests lie in the fields of high-speed commercial catamarans, computational methods and yacht technology.

Kim Klaka is a naval architect with particular interests in hydrodynamics and small craft. He joined the Centre for Marine Science and Technology at Curtin University, Perth, in 1985 and is currently the Director of the Centre. He is the "Seakeeping of Catamarans" Task Leader for AME CRC.

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now employed by Austal Ships, ** now employed by Formation Design Systems

NOMENCLATURE

AME CRC	Australian Maritime Engineering Co-operative Research Centre
BOA	Vessel Beam Over All
CMST	Centre for Marine Science and Technology
Fn	Froude Number
g	Acceleration due to gravity
L	Length of Vessel
LOA	Length Over All
LWL	Length on Water Line
o/b	Outboard
RMS	Root Mean Square
Δ	Displacement of Vessel
∇	Immersed Underwater Volume of Vessel

1. INTRODUCTION

1.1. BACKGROUND

Loads imposed on a monohull operating in a seaway have been the topic of much research and are relatively well understood compared with catamaran loading. With a catamaran a whole new series of structural problems arise. The primary additional problem is related to predicting the size of the loads acting on the structure connecting the two hulls.

In the past cross-structure loads have been predicted mainly from experience with similar vessels with inbuilt safety factors or error margins. Classification society rules have for the most part been empirically derived from structural designs that have stood the test of time. The work of Disenbacher (1970) is still the basis for obtaining most operational design loads for catamarans.

Smaller catamarans, less than 50 m, are essentially designed to satisfy local load conditions, Fan & Pinchin (1997). With these smaller catamarans it is inherent in the design that they will be able to withstand the global loads applied to them if they satisfy local load conditions. As the size of catamarans increase the global loads will tend to dominate the design of the overall structure, Speer (1996).

The primary global loads that are being investigated in this research, under guidance from industry, are listed below in order of priority.

1. Longitudinal bending moment of demihulls, Figure 1-1.

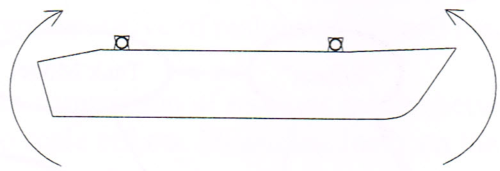

Figure 1-1: Longitudinal Bending Moment

2. Transverse vertical bending moment in cross-structure, Figure 1-2.

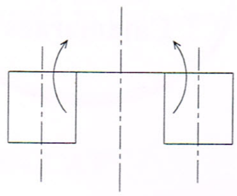

Figure 1-2: Transverse Vertical Bending Moment

3. Torsional moment in cross-structure (Pitch connecting moment), Figure 1-3.

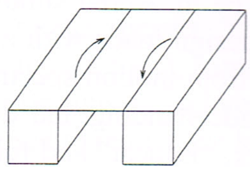

Figure 1-3: Pitch Connecting Moment

For the transverse vertical bending moment the most severe condition has been found to be at zero speed in beam seas. Lankford (1967) was the first to discover this somewhat surprising result with his work on the ASR catamaran.

Faltinsen et al. (1992) presented numerical and experimental results of global wave loads at a Froude number of 0.49. The results showed satisfactory agreement except for transverse vertical shear. At high vessel speeds, the numerical model shows that the transverse vertical bending moments and shear force are generally largest in beam seas, while the largest value for pitch connecting moment occurs at a wave heading angle of 60 degrees for most wave periods. Roll motion is important for vertical shear and pitch connecting moment, while heave and pitch acceleration influence the vertical bending moment.

1.2. METHODOLOGY

The aim of this project is to develop an improved understanding of wave induced loads on catamarans by developing techniques for their prediction and investigation.

To achieve this overall aim various tools are available. They are outlined in Figure 1-4 along with their associated contributions.

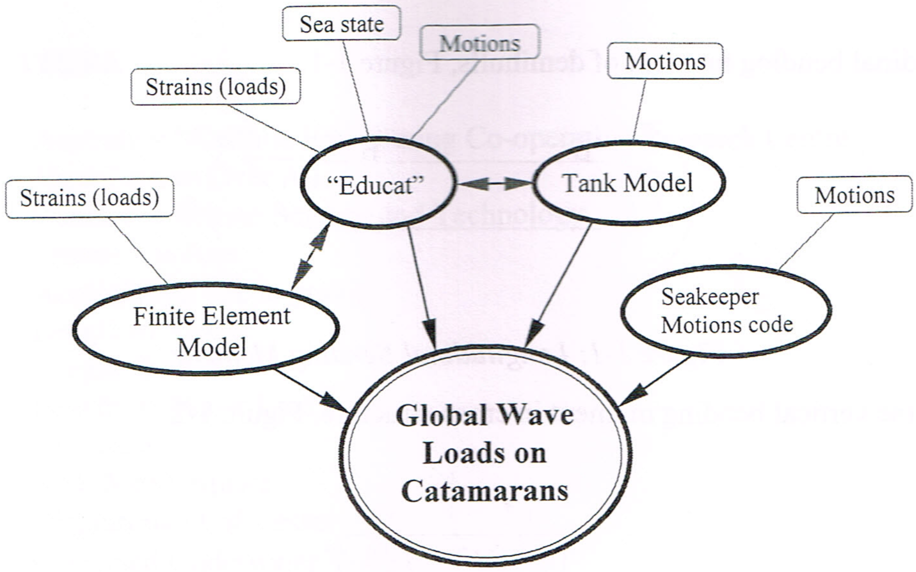

Figure 1-4: Overview of Project

The 8m research vessel "Educat", described in detail in Section 2.1, is central to the goals of this research. It is able to operate in real ocean waves at specified heading and speed. The sea state, vessel motions and loads can be measured simultaneously. It is common practice for regulatory bodies to associate their design loads with a maximum expected value of vertical acceleration, which can be obtained from motion spectra. Thus evaluating the motions of the "Educat" is an important step in understanding and developing a link to the global loads. Comparing motions between the tank, "Educat" and full scale will also establish the validity of comparing loads determined from these methods. Due to the small physical size, and relatively simple construction of "Educat", the strain gauges have been calibrated by applying known loads to the vessel while out of the water. This procedure is outlined in Section 2.3. These known load conditions are then applied to the finite element model to correlate the predicted strains with those measured on the "Educat". This will allow the global loads experienced during operation to be determined more accurately. The "Educat" has a very simple structure and thus the combination of load cases which can produce a given strain combination is greatly reduced. This is a distinct advantage over strain gauging full scale vessels in which local strains can be easily obtained, however relating these strains to a global load requires an in depth knowledge of how the structure reacts to different load case combinations. It is also logistically difficult to obtain wave magnitude and direction, motions, and loads data simultaneously for full scale vessels, thus only statistical predictions can be derived rather than establishing transfer functions which will allow different operational conditions to be tested and to evaluate the most severe load conditions.

Finite element modelling is currently commonplace in high speed vessel design. The fundamental problem with finite element modelling is knowledge of the loads to be applied. Classification Societies impose loading conditions which they feel are representative cases which will allow for safe operation of the vessel. These load cases while providing a structurally sound design are not necessarily representative of real operational conditions experienced in a seaway and thus may produce a somewhat conservative design.

By comparing the strains predicted by finite element modelling of these imposed load cases with those measured on the "Educat" during operation it will be possible to establish if these load cases are realistic and representative of real wave induced loadings.

The towing tank model gives comparison of motions results between the tank model and full scale tests and an insight into scale effects. Measuring loads on the model was not an available option in this project.

The strip theory based Seakeeper motions prediction program, outlined in Section 4, provides another tool for validating the motions of the "Educat".

All of these tools mentioned provide input to the overall goal of further understanding global loads on catamarans.

2. SEA TRIALS AND THE "EDUCAT"

2.1. VESSEL DESCRIPTION

The "Educat" is an 8m research vessel, the hulls constructed of aluminium with a plywood/fibre glass accommodation pod. It is shown below in Figure 2-1. It essentially consists of two hulls joined by two tubular beams. The accommodation pod is resiliently mounted to the top of these beams to reduce the dynamic loads transferred to the hulls. This practice is used in various catamaran designs to isolate the accommodation decks from the main structure. A system of cables is used to reduce the lateral motion of the pod.

Figure 2-1: "Educat" Research Vessel

The cross-structure is shown below in Figure 2-2.

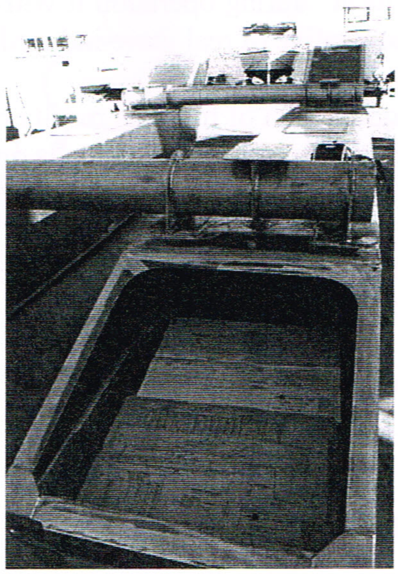

Figure 2-2: "Educat" Cross-structure

Particulars of the "Educat" are shown below in Table 2-1.

LOA	7.80m
LWL	6.80m
BOA	2.79m
Demi-Hull Beam	0.74m
Draught	0.28m
Separation between centre lines	2.05m
Power	twin 70hp o/b
Max. speed	25 kts

Table 2-1: "Educat" Particulars

2.2. INSTRUMENTATION

The "Educat" is instrumented to record loads and motions while operating in a seaway. The vessel motions are obtained from a TSS 335 motion sensor which provides real time heave, roll and pitch, as well as vertical and horizontal accelerations, at a rate of 21.333 Hz.

The sea state is evaluated using CMST wave recorders. These are normally free floating, suspended beneath buoys and the pressure fluctuations, sampled at 4 Hz, are used to calculate the sea surface elevation spectra.

The speed and heading of the vessel are obtained by visual observation of the ship speed display, derived from a water wheel, and a simple compass respectively.

Strain gauges are applied throughout the "Educat" in locations specific to particular load cases. The positions of the strain gauges for measurement of longitudinal bending are shown below in Figure 2-3, with their associated identification. All gauges mounted in the hulls are aligned longitudinally.

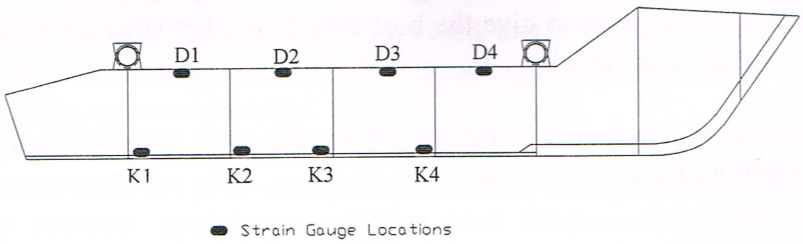

Figure 2-3: Strain gauge positions in starboard hull

Positions of gauges for measurement of cross-structure loads are shown in Figure 2-4.

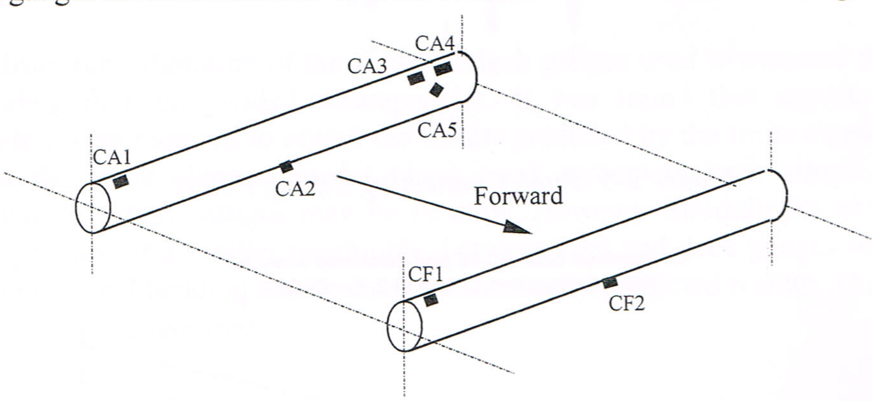

Figure 2-4: Cross-structure strain gauge positions

The strain gauge gives a time varying voltage signal due to a change in resistance of the gauge as it is strained. The change in resistance is quite small and hence the voltage change is also quite small so the signal is amplified up to 5000 times before recording. A half bridge configuration for the strain gauges is also used to increase the output voltage per unit of strain. The analogue signal from the strain gauge is converted to digital using a data acquisition unit (IOTech DAQ book) which also allows filtering of the signal and it is then logged on a PC. Strain gauge data is recorded at a sampling frequency of 100 Hz with a 20 Hz low pass filter.

2.3. CALIBRATION

Due to the small physical size, and relatively simple construction of "Educat", the strain gauges have been calibrated by applying known loads to the vessel while out of the water. These known load conditions have then been applied to the finite element model to correlate the predicted strains with those measured on the "Educat" during calibration.

Known loads were applied by supporting the vessel in various configurations and changing the loads at the desired location using hydraulic jacks. The loads applied by these jacks were measured with calibrated load cells. The physical deflections were also measured using a theodolite. The deflection results provided independent confirmation of the finite element models validity. The measured deflections showed good agreement with those obtained from finite element analysis.

When converting the strains measured during the calibration procedure to stresses, the value of Young's modulus was chosen to give the best correlation between the measured and finite element results; "tuning" the model.

An example of this is shown below in Figure 2-6 for the pitch connecting calibration. Figure 2-5 shows the location of the lifting point.

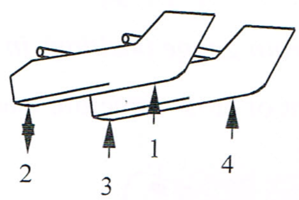

Figure 2-5: Pitch Connecting Lifting Points

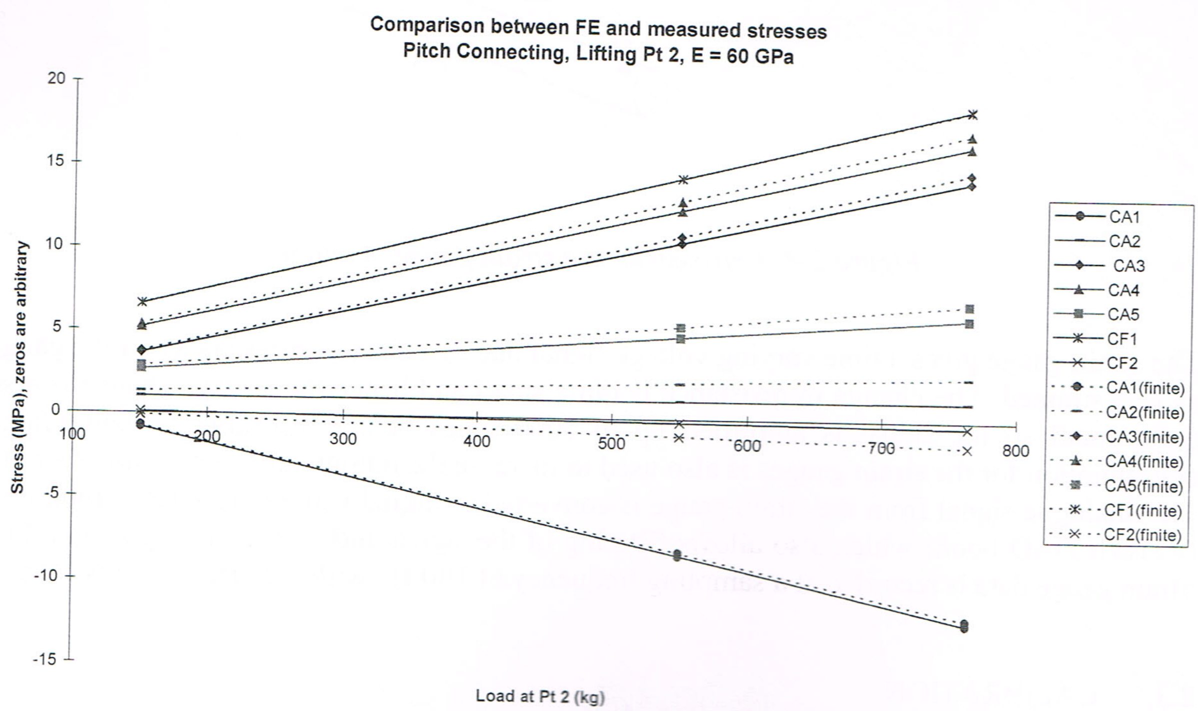

Figure 2-6: Pitch Connecting Calibration Results

Good correlation between experimentally measured strains and those obtained from the finite element model are obtained by applying a young's modulus of 60 GPa to the experimental strains. The standard value assumed for Young's Modulus of Aluminium is 70 GPa. This is used throughout the finite element model implying that the strains obtained from the finite element model must be multiplied by a factor of 70/60 to obtain the strains measured during the calibration. Thus the finite element model is "stiffer" than the actual "Educat", as might be expected.

It can be seen that gauges CA2 and CF2 (located at the transverse centre of each cross structure beam) show little response to Pitch Connecting Moment and thus are used to obtain the transverse bending moment directly.

Calibration was conducted for pitch connecting moment, lifting at points 1,2 and 3; transverse vertical bending moment, applying a splitting load between the hulls; and longitudinal bending by applying a point load at the midships of the instrumented hull.

The linearity of the results shown in Figure 2-6 is typical for the other load cases calibrated.

The results from the calibration of the keel and deck gauges used to measure the longitudinal vertical bending moment yielded discrepancies. It was found that significantly different Young's moduli were required to obtain the results predicted by the finite element model. The results from the finite element model show good agreement with simple beam theory, suggesting that the strain gauges may be in error. However Stredulinsky et al (1999) also found discrepancies, of a smaller magnitude, between keel and deck gauges when comparing measured longitudinal bending moments and numerically predicted results. This suggests that further investigation is required.

2.4. TRIALS CONDITIONS

Trials were conducted in Cockburn Sound, off Fremantle, Western Australia, for various headings, speeds and sea conditions. The water depth where the trials were conducted is generally between 15 and 20m.

A list of trials conducted to date is shown below in Table 2-2. Each run was conducted for five minutes duration and a new data set recorded for each different heading and speed. The speeds chosen correspond approximately to those used in model testing.

Type	Heading (degrees)	Speed (kts)
Head Seas	180	5
	180	9
	180	12
	180	15
Beam Seas	90	0
	90	8
	90	12
	270	0
	270	15
Quartering Sea	145	8
	145	15

Table 2-2: Trials Conducted

Figure 2-7 show the trials course and positioning of wave buoys. One wave buoy was located at the top and one at the bottom of the trials course to measure the spatial variation in the wave field.

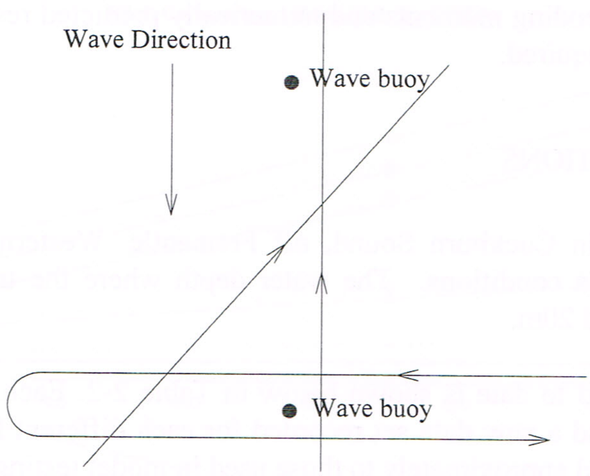

Figure 2-7: Trials Course

The results of these trials are presented in Section 6 and compared with towing tank and finite element results.

3. MODEL TESTS

Captive model tests were conducted at the Australian Maritime College (AMC) in Launceston, Tasmania, in June 1999. The tank is approximately 60 m long, 3.5 m wide, and 1.5 m deep, allowing testing in head seas and zero speed beam seas. The objective of the model test program was to measure the motions of the Educat operating in head and beam seas for regular waves. These results would enable comparisons to be drawn between full scale trials results, theoretical predictions and model measurements, and also an assessment of scale effects.

For head seas five conditions were tested. For one vessel speed three different wave heights were tested to check the linearity of the motion responses. At constant wave height two more speeds were tested.

Beam seas tests were conducted at zero speed for two different wave heights.

The speeds tested and their scaled equivalents are outlined in Table 3-1.

	Model		Educat
LWL (m)	1.7		6.8
Velocity	m/s	Fn	kts
	2.15	0.526	8.4
	2.99	0.732	11.6
	3.27	0.801	12.7

Table 3-1: Summary of Vessel Speeds

The model was a 1/4 scale and the ballast condition tested corresponded to the standard trials operating condition of the "Educat".

The pitch gyradius was set using a roll frame. Difficulties where encountered with this method and it was later concluded that the desired pitch gyradius was not achieved for the tests. This resulted in poor agreement between tank results, and both "Educat" and Seakeeper pitch response functions.

The results of towing tank tests are presented in Section 6.1.

4. STRIP THEORY PREDICTION

The Seakeeper motions program has been used to evaluate the motions of the "Educat" and investigate the effects of varying displacement etc.

Seakeeper is a two dimensional strip theory method, based on the work of Salvesen et al. (1970). Conformal mapping techniques are used to calculate the section added mass and damping for the vessel. These are then integrated to obtain the global vessel added mass, damping and cross-coupling terms. The coupled heave and pitch equations are solved to obtain the vertical plane RAOs. The motions are calculated for one hull in isolation, previous work (Wellicome et al. 1995) found that little interference between the hulls was found above a Froude number of 0.2. In addition, Seakeeper has been validated against a wide variety of catamaran data (Couser 1999) and found to give acceptable results up to a Froude number of 0.8.

For a user specified wave spectrum, heading and vessel speed, Seakeeper is able to calculate the response of the vessel. Response amplitude operators (RAOs) are computed as well as the added resistance, significant absolute and relative motions, velocities and accelerations of the vessel in the specified sea spectrum. Motion, velocity and acceleration spectra are computed at the centre of gravity and vertical motions may also be computed for other positions on the vessel.

5. FINITE ELEMENT MODEL

A global finite element model has been developed using MSC/NASTRAN. The model is constructed primarily of QUAD4 elements with CBAR elements use to model the stiffeners. Triangular elements are used very sparingly and only in areas where the number of nodes did not allow for a QUAD4 element.
Details are contained in Kastak (1998). Figure 5-1 shows the finite element model of the "Educat".

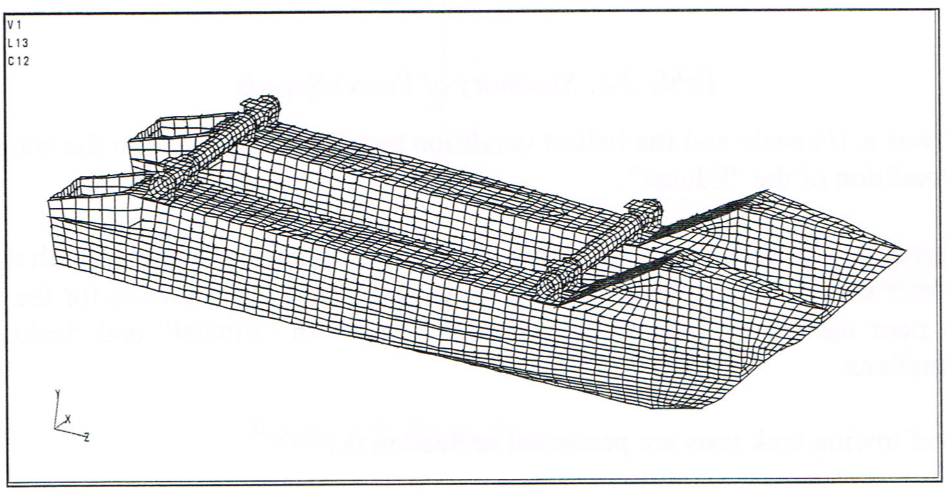

Figure 5-1: "Educat" Finite Element Model

6. RESULTS COMPARISON AND DISCUSSION

The following section presents a small sample of the data which has been obtained for the "Educat" from the various tools mentioned in the preceding sections. At present, analysis of this data is at an early stage and the full implications of this research have yet to be assessed.

6.1. MOTIONS

A typically encountered wave spectrum during "Educat" trials is shown below in Figure 6-1.

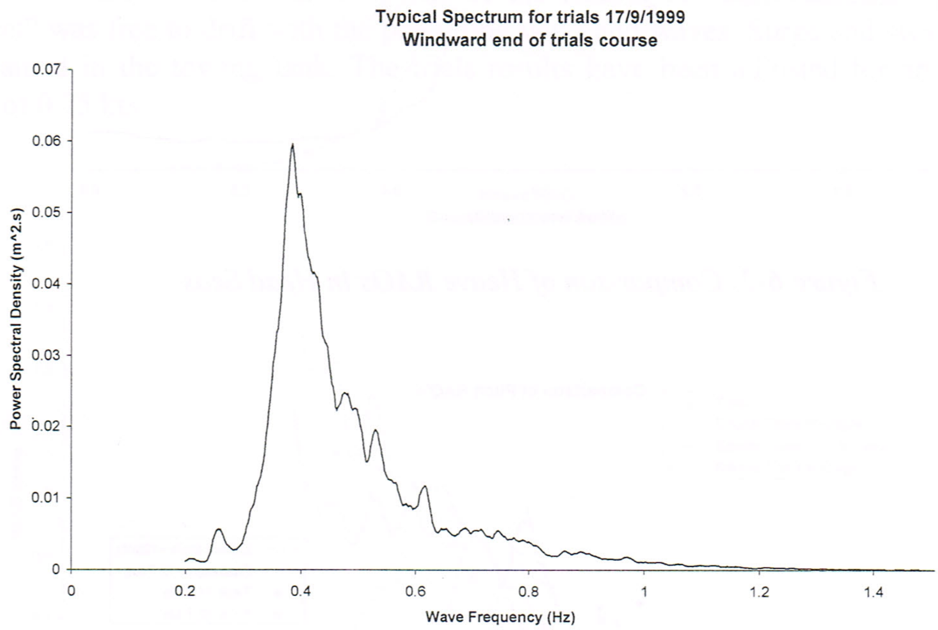

Figure 6-1: Typical Wave Spectrum

Analysis of the TSS motion sensor output gives the spectral densities for the roll, pitch and heave motions. Dividing the spectral densities and taking the square root gives the response amplitude operator (RAO) for the given motion. Throughout the duration of the trials the variation in the wave field is accounted for by calculating the wave spectrum in 40 minute windows and for each run the corresponding wave spectrum in time is used to calculate the RAO. The spatial variation is generally small and the spectrum from the closest buoy is used in the RAO calculations. The measured RAOs are compared to the RAOs predicted from the Seakeeper motions code and towing tank results in Figures 6-2, 6-3, 6-4, and 6-5. A cut-off frequency of 0.25 Hz has been applied to the trials results. Below this frequency there is very little energy in the wave spectrum, as can be seen in Figure 6-1.

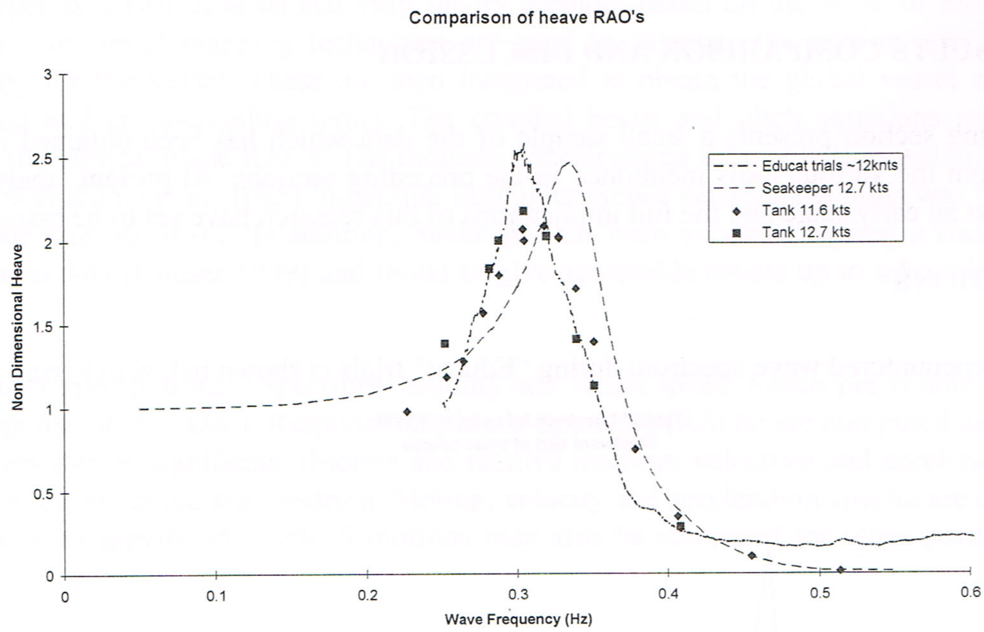

Figure 6-2: Comparison of Heave RAOs in Head Seas

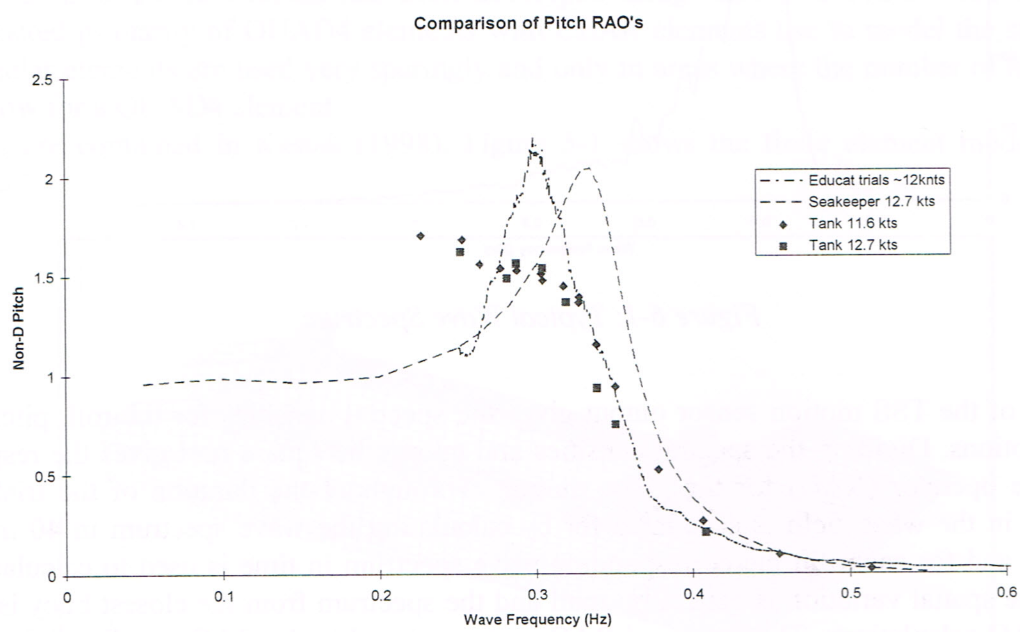

Figure 6-3: Comparison of Pitch RAOs in Head Seas

As previously mentioned difficulty was encountered in setting the pitch gyradius for the model. Using the Seakeeper motions package it was found that increasing the pitch gyradius to 34% LOA, from 25% LOA (desired pitch gyradius), does produce similar results to the model tests.

It is interesting to note that the magnitudes of the motion peaks from full scale trials and those predicted by Seakeeper are in good agreement, though Seakeeper tends to overpredict the resonant frequency. Both these sets of data show a resonant peak significantly greater than that measured in the model tests; though the effective scaled wave height used in the model tests and that present during the sea trials were very similar. In addition the motions measured during the model tests showed good linearity with wave amplitude. Similar results were found by Maggi et al (1998), where the peak amplitude of tank tests at higher speeds were somewhat smaller than those measured at full scale.

These findings suggest that there may be significant scale effects for vessels of this type, and this is an area worthy of further investigation.

Figures 6-4 and 6-5 show the response of the Educat in beam seas. During the trials the "Educat" was free to drift with the prevailing wind and waves. Surge and sway motions were constrained in the towing tank. The trials results have been adjusted for an estimated drift speed of 0.75 kts.

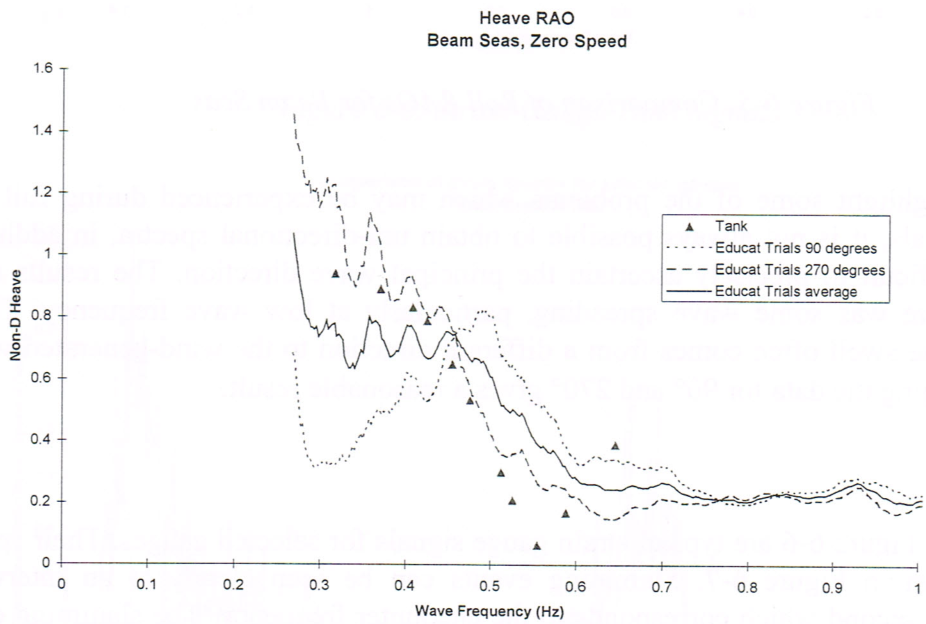

Figure 6-4: Comparison of Heave RAOs for Beam Seas

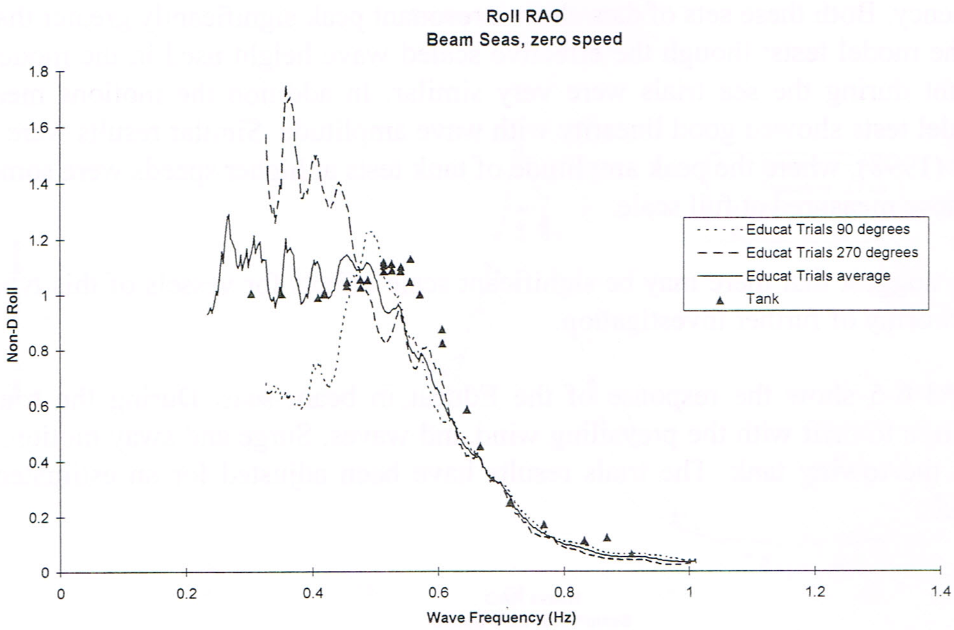

Figure 6-5: Comparison of Roll RAOs for Beam Seas

These results highlight some of the problems which may be experienced during full scale trials. During trials, it is not always possible to obtain uni-directional spectra, in addition it may also be difficult to precisely ascertain the principal wave direction. The results above suggest that there was some wave spreading, particularly at low wave frequency. This is possible since the swell often comes from a different direction to the wind-generated waves. However, averaging the data for 90° and 270° gives a reasonable result.

6.2. LOADS

Shown below in Figure 6-6 are typical strain gauge signals for selected gauges. Their spectral content is shown in Figure 6-7. Slamming events can be seen clearly at an interval of approximately 1 second which corresponds to the encounter frequency. The slamming events are most exaggerated at the end of the cross structure beams (CF1) and result in quite high stresses in the cross-structure in direct head seas. This phenomenon has not been fully investigated but it is thought that because the vessel is not operating in perfect head seas there will be some pitch-connecting moment producing a large bending moment at the beam ends, since the cross beam has to follow the flat deck surface at its' end. This could induce unusual vibration modes - walking mode, longitudinal mode and split mode. The effect of the slamming can also be seen throughout the rest of the structure. The underlying global load can be obtained by spectrally filtering out higher frequencies where the slamming phenomenon occurs. In Figure 6-7 the main peak at approximately 1 Hz corresponds to the underlying global load occurring at encounter frequency. The less defined peak at around 10 Hz is probably a result of the slamming events on the structure.

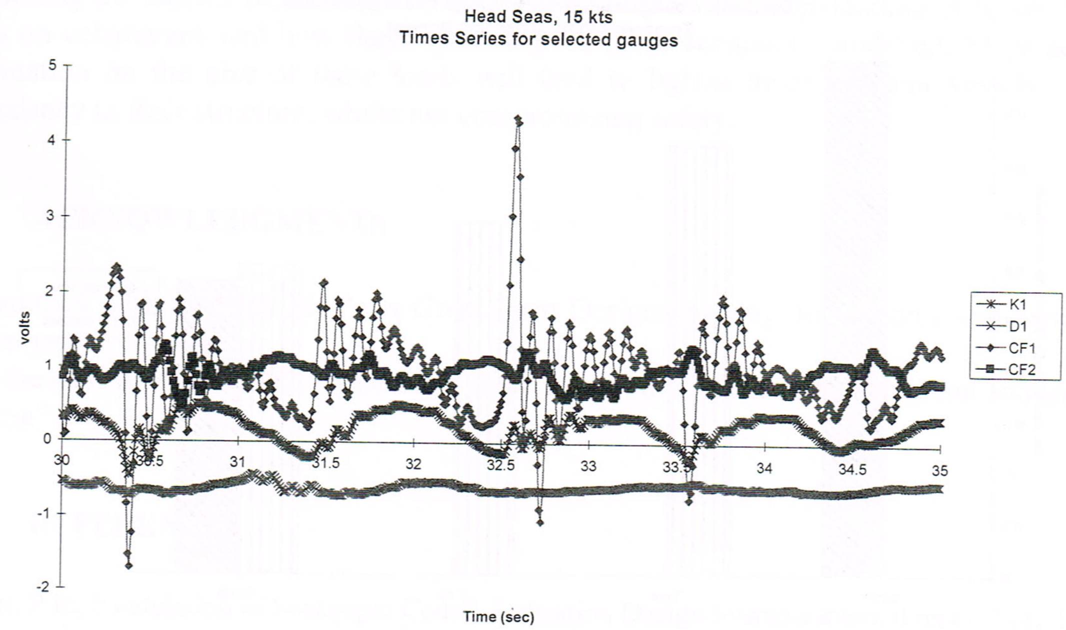

Figure 6-6: Strain Gauge Time Signals

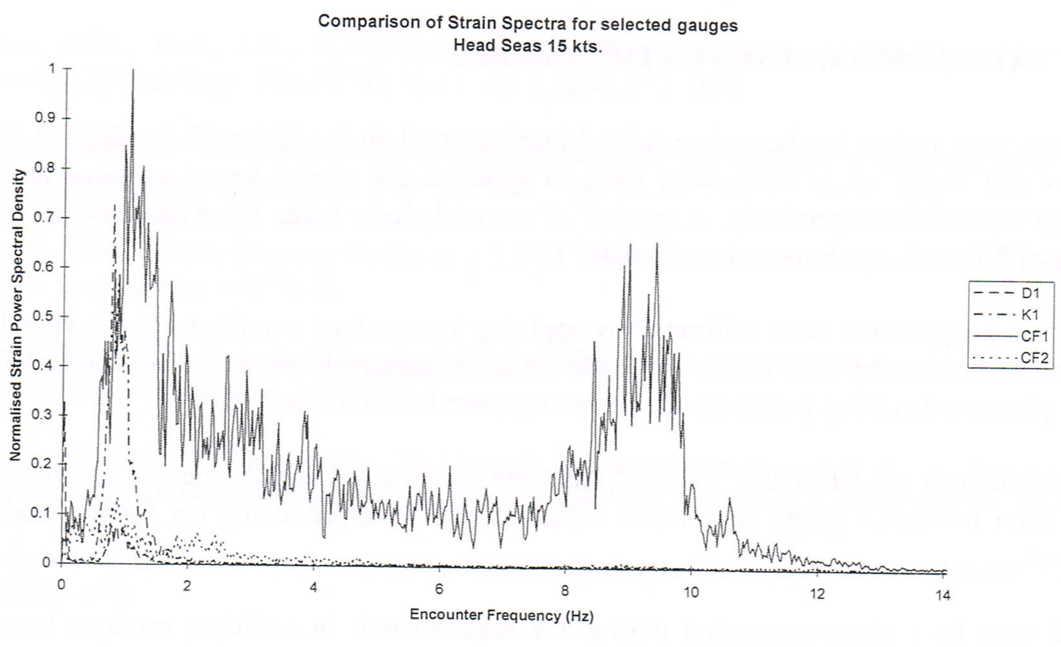

Figure 6-7: Spectral Analysis of Time Signals

Figure 6-8 shows a comparison of maximum strain amplitude (spectral peak) for cross structure gauge CA2 in beam seas for varying speeds. The strain at CA2 is a direct measure of the transverse vertical bending moment in the cross structure. It can be seen that maximum strain amplitude decreases as the speed increases. This agrees qualitatively with findings by Lankford (1967). Interestingly RMS values and energy in the spectrum remain reasonably constant.

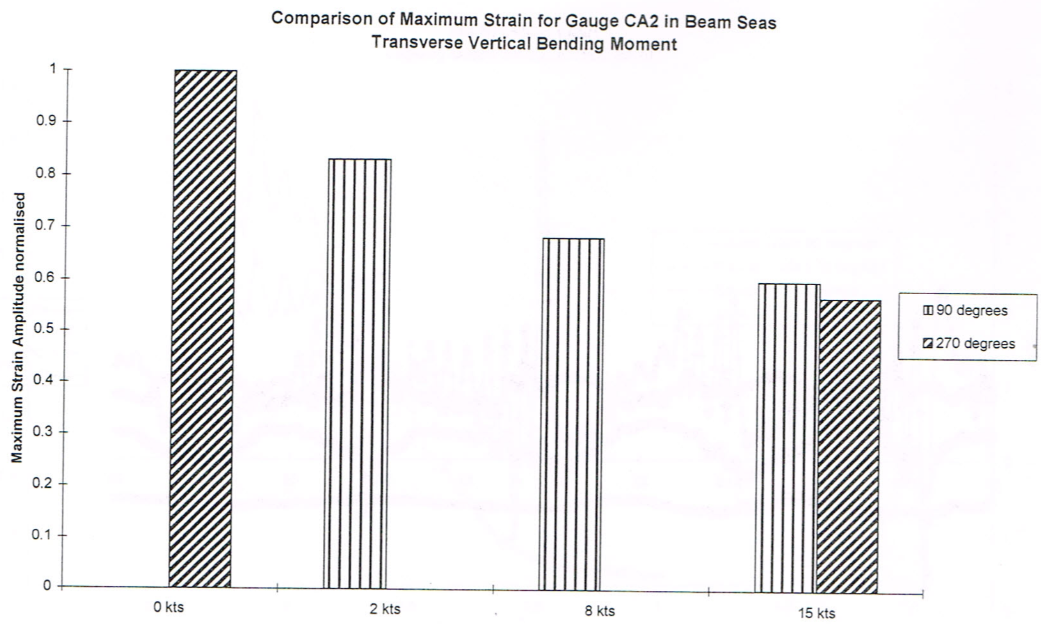

Figure 6-8: Transverse Vertical Bending Moment response to speed.

7. CONCLUSIONS AND FURTHER WORK

A strain gauge system has been devised and implemented on the "Educat". In addition, motion sensors and wave buoys have been used to measure the global vessel motions and local incident wave field respectively. A number of sea trials have taken place and have provided a wealth of full scale motions and strain data.

The strain gauges have been calibrated by applying known load conditions to the vessel. This was carried out on dry land where the loads could be accurately measured with load cells and the displacements of key points on the vessel measured with a theodolite.

A finite element model of the "Educat" has been constructed and the results compared with those from the strain gauge calibration. Initial results look promising but further analysis is required.

Model tests have been completed with a 1/4 scale model. In addition, motions have been predicted with a strip theory code. This data has provided useful validation of the full scale motions measurement systems and provides an insight into scale effects.

Work on this project is ongoing: the strain gauge data collected will be used to statistically predict life time operational loads for the "Educat". Having established a procedure for calculating the global loads from local strain measurements it will then be possible to compare results with the finite element model for different global load cases. This will give an insight into developing more operational based design loads for catamarans.

Continuing the current research it will be possible to gain an understanding of wave induced loads on catamarans and how these loads can be more accurately modelled. More accurate information on the size of these loads will lead to lighter more efficient vessels without redundancy in their structure, whilst not compromising safety.

8. ACKNOWLEDGMENTS

The authors wish to thank Mr Peter Goss, Incat Designs-Sydney, for his on going support of this project.
Also the staff of AME CRC, particularly those involved with the instrumentation and trials on "Educat".

9. REFERENCES

Couser, P.R., "Validation of Seakeeper Code", Formation Design Systems internal report May 1999.

Dinsenbacher, A.L., "A Method for Estimating Loads on Catamaran Cross-Structure", Marine Technology, Volume 7, No.4, October 1970.

Faltinsen, O.M., Hoff, J.R.., Kvalsvold, J., Zhao, R., "Global Wave Loads on High-Speed Catamarans", Proceedings, PRADS'92, Vol.1, pp. 1.360-1.375, 1992.

Fan, M., Pinchin, M.. "Structural Design of High Speed Craft - A Comparative Study of Classification Society Requirements", Proceedings, Fast '97, Vol. 1, pp. 27-33, 1997.

Kastak, D., 1998 "Finite Element Model of a 7.9 M Training/Research Vessel - Educat, Task B6", AMECRC Internal Report IR 98/2.

Lankford, B.W., "The Structural Design of the ASR Catamaran Cross-Structure," American Society of Naval Engineers Journal, August 1967.

Maggi, A., Thomas, G., McRae, B., "Seakeeping: A Comparison of Numerical and Towing Tank Predictions with Full Scale Measurements", Proceedings, Ship Motions & Manoeuvrability, 1998.

Salvesen, N., Tuck, E.O. & Faltinsen, O.M., "Ship Motions and Sea Loads", Trans. SNAME, vol. 78, pp.250-287, 1970.

Speer, T.D., "High Speed Craft and the Design Learning Curve" Proceedings, Ausmarine '96, pp. 85-92, 1996.

Stredulinsky, D. C., Pegg, N.G., Gilroy, L.E., "Motion, Load and Structural Response Predictions and Measurements on CFAV Quest", Trans. RINA, Vol. 141, Part B, pp. 117-123, 1999.

Wellicome, J.F., Temarel, P., Molland, A.F., Couser, P.R., "Experimental Measurements of the Seakeeping Characteristics of Fast Displacement Catamarans in Long-crested Head-seas". Ship Science Report No. 89, December 1995.

PAPER FIVE

DO WE STILL NEED EXPERIMENTS?

Presented by Jorgen Krokstad, Curtin University / Marintek

Paper presented at

Conference

HYDRODYNAMICS WITHOUT INTEGRALS

2 November 2000, Fremantle, Western Australia

{g:\jmb\RINA\RINA Forum 2000}

Do We Still Need Experiments?

by
Dr. Jørgen R. Krokstad
CMST/MARINTEK

After many years of an almost unforeseen development within computer hardware - RAM capacity, disk capacity, processor speed, vectorizing, parallel processing - leading to extremely reduced cost performance, naval architects, offshore engineers and scientists ask themselves; ***Do we still need these expensive model tests carried out in a test tank, cavitation tunnel or ocean basin?*** The scientific organisations with huge investments in laboratories have a professional obligation to make strategic decisions between numerical and experimental investments. They must regularly update and review their knowledge. For an institute like MARINTEK with one of the world's most advanced hydrodynamic laboratories, the market aspect and the world wide need for experimental services are obviously of large economic importance. The need for any investments in Australian experimental facilities will be dependent on rational answers to this question.

It is not the intention of this presentation to give an explicit answer to this question but rather to illustrate the status of numerical tools and the variety of numerical approaches applied within a commercial research institute like MARINTEK. Examples from frontier numerical developments internationally will be illustrated as well. The ongoing and prospective need for combining model tests and numerical analysis is still valid. The dramatic increase in cheap computer power does not change the fact that the numerical approaches are not only dependent on "increased numerical resolution" but also on "validity" of mathematical approaches. To illustrate, the complex phenomenon of vortex generation from a fixed cylinder in a viscous fluid is perhaps not a "deterministic process" captured by direct numerical methods but rather of stochastic or chaotic nature. This is a problem that needs more than just powerful computers to solve in the future.

The presentation will in particular show the need for close interaction between numerical and experimental analysis. There is a shift from using model tests for final verification of a design or a concept, to calibration of a numerical model. The numerical model will instead be used for a large range of parametric variations not possible to cover within a restricted experimental testing period. Many of MARINTEK's clients have shown a need for this service. In addition, so called hybrid modelling techniques have been developed. Here parts of the system, for example the mooring or riser system, are modelled numerically while the vessel parameters are experimentally obtained.

Looking at the developments of numerical methods within fluid dynamics and other related numerical fields we identify many generic elements which will be discussed. The past experience within the development of numerical methods has clearly shown that there is no generic way to solve all the complexities involved in naval architecture or offshore hydrodynamic problems. Different computational strategies will be discussed and the maturity of the different numerical approaches will also be reviewed.