Safety at Speed - S@S

FORMULATION OF MODELS
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CONTENTS

1. EXECUTIVE SUMMARY SUITABLE FOR PUBLICATION

The present report corresponds to the intermediate report (according to the WP2 storyboard revised on 2002-12-16) related to deliverables 2.2.2 and 2.2.3 of Work Package 2 – Ship Motions Hazards of Safety At Speed (S@S). It presents the works carried out in Sub-tasks 2.2.2 and 2.2.3 for the formulation of risk and cost models for hull design and operational practice respectively.

Following the list of hazardous events related to ship motions established in the previous task (Task 2.1), two basic models, related to the ship seakeeping and dynamic stability (in broaching situation) characteristics have been formulated and will be used to derive the risk models. Methodologies for their development and validation, based on the exploitation of existing data, on dedicated numerical seakeeping analyses, model tests and sea trials, have also been specified.

The next steps will consist in refining and validating these models according to the defined methodologies, to define cost models, together with the Shipyard and other work packages, and to implement the final models in a tool form, for integration in the Project Tool.

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2. INTRODUCTION

The present report corresponds to deliverable 2.2.2 of Work Package 2 – Ship Motions Hazards of safety At Speed (S@S). It presents the work carried out in Sub-tasks 2.2.2 and 2.2.3 for the formulation of risk and cost models for hull design and operational practice respectively.

The description of WP2 activities described in the work programme considers separately models related to hull design (Sub-task 2.2.2) and to operational practice (Sub-task 2.2.3), with corresponding separate deliverables (respectively D2.2.2 and D2.2.3). However, since the various models of risks related to ship motion hazards simultaneously involve both hull design and operational practice parameters, the activities of both sub-tasks are performed in common, and the two deliverables are merged in the present document.

3. DESCRIPTION OF WORK

The activities performed within the first Task 2.1 of WP2 allowed to identify the main hazards related to ship motions, and to relate them to the occurrence of various events. These hazards and events are listed in section 4.1.

The risk and cost models are meant to be implemented in the so-called Project Tool, and should be simple enough to be used at early design stage by the designers. Consequently, simple models have to be developed. In this context, the activities performed in Sub-tasks 2.2.2 and 2.2.3 concerned the investigation of simple mathematical models and the definition of the work to be carried out to develop and validate these models. The results of this work are presented in sections 4.2 and 4.3 respectively.

4. FORMULATION OF RISK MODELS

4.1 List of identified hazards related to ship motion

The work performed in Task 2.1 allowed to confirm and complement the main hazards related to ship motions identified in the work programme. These are :

- Crew and passengers disorientation and injury
- Large ship loading causing structural failure and foundering
- Loss of ship control

These hazards can be related to the occurrence of these events :

- Excessive ship motions and accelerations
- Excessive elastic ship vibrations
- Excessive local and global wave loads
- Dynamic capsize (broaching)
- Excessive noise, vibration levels and bad indoor climate

The risk prediction of the first three events requires first to model the seakeeping characteristics of the ship.

The risk prediction of dynamic capsize requires to model the dynamic stability of the ship in broaching situation.

The two above models are to be formulated within sub-tasks 2.2.2 and 2.2.3, and are described in the present report. The modelling of excessive elastic ship vibrations and excessive local loads will be carried out in co-operation with WP3, mainly through the delivery of experimental (model tests and sea trials) data collected within WP2.

The influence of excessive ship motions and accelerations on human is dealt with in Sub-task 2.2.1, with seakeeping models developed here as main input. The risk prediction of excessive noise, vibration levels and bad indoor climate, and of their effects on human (passengers and crew) is also studied in Sub-task 2.2.1. The results of this sub-task are presented in a separate report (deliverable 2.2.1).

4.2 Description of models

4.2.1 Seakeeping models

Based on comparisons, documented in Document ID Code: S102.00.13.050.001c a rational procedure able to predict the wave-induced motions and accelerations with sufficient engineering accuracy is formulated. The procedure takes into account all pertinent parameters: length, breadth, draught, block coefficient, forward speed and operational profile. The formulas derived are semi-analytical so that the calculations can be easily implemented in a decision support system.

4.2.1.1 Frequency response functions

The frequency response functions Φ_w, Φ_θ for heave (w), and pitch (θ), for the vertical wave-induced motions of a homogeneously loaded box-shaped vessel can be derived analytically using the linear strip theory proposed by Gerritsma and Beukelman (1964). Neglecting the coupling terms between heave and pitch and assuming a constant sectional added mass equal to the displaced water the equations of motions in regular waves with amplitude *a* can be written, Jensen (2001) :

$$2\frac{kT}{\omega^2}\ddot{w} + \frac{A^2}{kB\alpha^3\omega}\dot{w} + w = aF\cos(\varpi t) \tag{1.1}$$

$$2\frac{kT}{\omega^2}\ddot{\theta} + \frac{A^2}{kB\alpha^3\omega}\dot{\theta} + \theta = aG\sin(\varpi t) \tag{1.2}$$

Where k is the wave number, ω the wave frequency ($\omega^2 = kg$) and B, T the breadth and draft of the box. Differentiation with respect to time t is denoted by a dot. Furthermore, the frequency of encounter ϖ is given by

$$\varpi = \omega - kV\cos\beta = \alpha\omega \tag{1.3}$$

where V is the forward and β the heading angle (180 degrees corresponding to head sea). In term of the Froude number $Fn = V/\sqrt{gL}$ and L the length of the box,

$$\alpha = 1 - Fn\sqrt{kL}\cos\beta \tag{1.4}$$

The sectional hydrodynamic damping is modelled by the dimensionless ratio between the incoming and diffracted waves through the approximation, Yamamoto et al. (1986):

$$A = 2\sin(\frac{\varpi^2 B}{2g})\exp(-\frac{\varpi^2 T}{g}) = 2\sin(\frac{1}{2}kB\alpha^2)\exp(-kT\alpha^2) \tag{1.5}$$

The forcing functions F, G are

$$F = \kappa f\frac{2}{k_e L}\sin(\frac{k_e L}{2}) \tag{1.6}$$

$$G = \kappa f\frac{24}{(k_e L)^2 L}\left[\sin(\frac{k_e L}{2}) - \frac{k_e L}{2}\cos(\frac{k_e L}{2})\right] \tag{1.7}$$

where

$$k_e = |k\cos\beta| \tag{1.8}$$

is the effective wave number and

$$f = \sqrt{(1-kT)^2 + (\frac{A^2}{kB\alpha^3})^2} \tag{1.9}$$

The Smith correction factor κ is approximated by

$$\kappa = \exp(-k_e T) \tag{1.10}$$

Solution of (1.1) and (1.2) yields the frequency response functions

$$\Phi_w = \eta F \tag{1.11}$$

$$\Phi_\theta = \eta G \tag{1.12}$$

where

$$\eta = \left(\sqrt{(1-2kT\alpha^2)^2 + (\frac{A^2}{kB\alpha^2})^2} \right)^{-1} \tag{1.13}$$

Finally, the frequency response functions for the vertical motion $u = w - x\theta$ and acceleration $v = -\varpi^2(w - x\theta)$ at a longitudinal position x from amidships become

$$\Phi_u = \sqrt{\Phi_w^2 + x^2\Phi_\theta^2}; \quad \Phi_v = \varpi^2\Phi_u = \alpha^2 kg\Phi_u \tag{1.14}$$

4.2.1.2 Relative motions

Relative vertical motion *r(t)* with respect to the wave elevation *h(x,t)* at a position x measured positive from amidships can also be derived

$$r(t) = w(t) - x\theta(t) - h(x,t) \tag{2.1}$$

Its frequency response function Φ_r becomes

$$\Phi_r = \sqrt{(\Phi_w - \cos\varphi(x))^2 + (x\Phi_\theta - \sin\varphi(x))^2} \tag{2.2}$$

where

$$\varphi(x) = \xi - k_e x \tag{2.3}$$

with ξ defined by

$$\cos\xi = \frac{1-kT}{f}; \quad \sin\xi = -\sqrt{1-\cos^2\xi} \tag{2.4}$$

using equation (1.9). The frequency response function for the relative velocity is obtained by multiplication with ϖ.

4.2.1.3 Vertical wave-induced bending moment

An equivalent formula for frequency response function Φ_M for the wave-induced vertical bending moment amidships has been derived in Jensen and Mansour (2002). In the present notation it reads

$$\frac{\Phi_M}{\rho g B_o L^2} = \kappa \frac{1-kT}{(k_e L)^2}\left[1-\cos(\frac{k_e L}{2}) - \frac{k_e L}{4}\sin(\frac{k_e L}{2})\right] F_V(Fn) F_C(C_B) \tag{3.1}$$

The correction factor for block coefficient is taken from Jensen and Mansour (2002):

$$F_C(C_B) = \left[(1-\vartheta)^2 + 0.6\vartheta(2-\vartheta)\right]; \quad \vartheta = 2.5(1-C_B); \quad C_B = \max(0.6, C_B) \tag{3.2}$$

as also the speed correction factor:

$$F_V(Fn) = 1 + 3Fn^2 \tag{3.3}$$

In this reference a validation of the approach is given. Corrections for non-linearities are done as suggested in Jensen and Mansour (2002).

4.2.1.4 Short-term statistics

The standard deviation s_R of a linear wave-induced response R (e.g. heave, pitch, motions, accelerations, relative motions, wave bending moment) is given as

$$s_R^2 = \int_0^\infty \Phi_R^2(\omega) S(\omega) d\omega \tag{6.1}$$

where Φ_R is the frequency response function of the response. The stochastic properties of ocean waves is here modelled by the generalized JONSWAP wave spectrum $S(\omega)$ formulated in the wave frequency ω, the significant wave height H_S, the mean period T_S and the peak enhancement factor γ:

$$S(\omega; H_S, T_S, \gamma) = S_{PM}(\omega; H_S, F_2 T_S) \frac{1}{F_1} \gamma^{\exp\left\{-\frac{1}{2}\left(\frac{0.206 F_2 T_S \omega - 1}{\sigma}\right)^2\right\}} \tag{6.2}$$

where the Pierson-Moskowitch wave spectrum

$$S_{PM}(\omega; H_S, T_S) = 173 H_S^2 T_S \left(\omega T_S\right)^{-5} e^{-692(\omega T_S)^{-4}} \quad (6.3)$$

has been introduced. The coefficients F_1 and F_2 depend on γ as shown below:

$$\begin{aligned}
1 \le \gamma \le 2 &: F_1 = 0.24\gamma + 0.76;\ F_2 = -0.05\gamma + 1.05 \\
2 \le \gamma \le 3 &: F_1 = 0.22\gamma + 0.80;\ F_2 = -0.02\gamma + 0.99 \\
3 \le \gamma \le 4 &: F_1 = 0.20\gamma + 0.86;\ F_2 = -0.02\gamma + 0.99 \\
4 \le \gamma \le 5 &: F_1 = 0.20\gamma + 0.86;\ F_2 = -0.01\gamma + 0.95 \\
5 \le \gamma \le 6 &: F_1 = 0.18\gamma + 0.96;\ F_2 = -0.01\gamma + 0.95
\end{aligned} \quad (6.4)$$

whereas

$$\begin{aligned}
\sigma &= 0.07 \quad if \quad 0.206 F_2 T_S \omega \le 1 \\
\sigma &= 0.09 \quad if \quad 0.206 F_2 T_S \omega > 1
\end{aligned} \quad (6.5)$$

The standard JONSWAP spectrum is obtained by taking γ = 3.3. Often the zero-upcrossing period T_Z is given rather than T_S. Hence the relation, Gran (1992)

$$T_S = T_Z \frac{\sqrt{(10.89+\gamma)(5+\gamma)}}{6.774+\gamma} \quad (6.6)$$

is useful.

Finally, short-crested waves could be applied. This is easily done taking

$$S(\omega, \mu) = S(\omega) f(\mu) \quad (6.7)$$

where μ is the angle between the wind direction and the wave component and $f(\mu)$ is a suitable wave spreading function, expressed by

$$f(\mu) = A\cos^n \mu \qquad (n\ even; |\mu| \le \frac{\pi}{2}) \quad (6.8)$$

Thereby, the integration in equation (6.1) should be supplemented with an integration over $-\pi/2<\mu<\pi/2$ and, furthermore, the argument β (the heading angle) in the frequency response function must be replaced by $\upsilon-\mu$, where υ is the angle between the ship and the wind.

4.2.1.5 Extreme values

From the standard deviations the most probable largest responses $\tilde{s}_R$ within a given time period T can be estimated as

$$\tilde{s}_R = s_R\sqrt{2\ln(T/T_R)} \tag{7.1}$$

where T_R is the peak period for the response (estimated or measured), often taken as T_Z. The number N_{slam} of slams during the time T at the longitudinal position x can be estimated as

$$N_{slam} = \frac{T}{T_Z}e^{-\frac{1}{2}\left[\left(\frac{d}{s_r}\right)^2+\left(\frac{v_{cr}}{s_{rv}}\right)^2\right]} \tag{7.2}$$

where s_r and s_{rv} are the standard deviations of the relative vertical motion and relative vertical velocity at x. Furthermore, d is the draught at x and V_{cr} is a lower bound on the velocity necessary to yield a slamming force, usually taken to be $0.093\sqrt{Lg}$. Similarly, the number N_{green} of deck wetness during the time T at the longitudinal position x can be estimated as

$$N_{green} = \frac{T}{T_Z}e^{-\frac{1}{2}\left(\frac{f}{s_r}\right)^2} \tag{7.3}$$

where f is the freeboard at x. It should, however, be recognized that the predictions suggested in here should be carefully evaluated with measured data as they are subjected to large uncertainties.

4.2.1.6 Transverse motions

If the rolling motion is assumed to be decoupled from the other transverse motions the equation of motion for roll is

$$\left(I_4 + A_{44}\right)\ddot{\varphi} + B_{44}\dot{\varphi} + C_{44}\varphi = \xi_a F_4$$

where

φ	is the roll angle
I_4	is the mass moment of inertia
A_{44}	is the added mass moment
B_{44}	is the added damping
C_{44}	is the restoring moment coefficient
F_4	is the roll excitation moment
ξ_a	is the wave amplitude

The above equation is an ordinary differential equation where the coefficients are functions of the encounter frequency. The solution to the equation gives the transfer function for roll

$$\Phi_\varphi = \frac{F_4}{\left(\left[-\varpi^2\left(I_4 + A_{44}\right) + C_{44}\right]^2 + \varpi^2 B_{44}^2\right)^{1/2}}$$

where ϖ is the encounter frequency.

The mass moment of inertia, I_4, is assumed to be known for the ship and is given as input. The restoring moment coefficient, C_{44}, is calculated in the same manner as in strip theory. Hence the coefficient is given by :

$$C_{44} = g\Delta GM_T$$

where Δ is the displacement of the ship and GM_T is the transverse metacentric height.

The added mass moment, A_{44}, is assumed to be a function of the length and the breadth of the ship only, and constant in the wave frequency. From calculations for a wedge with different breadth to draft ratios it is found that a good approximation of A_{44} is given by $0.014\rho L_{pp} B^4$. The breadth B is given by $B = C_{WP} B_0$, where B_0 is the maximum waterline breadth, in order to account for the longitudinal variation of the breadth.

The excitation moment can be found using the Haskind relation, see e.g. Newman (1977), hereby the amplitude of the excitation moment per unit length in beam sea and zero forward speed is given by

$$|F_4| = \xi_a \left(\frac{\rho g^2}{\omega} b_{44} \right)^{1/2}$$

where ξ_a is the wave amplitude, b_{44} is the two dimensional added damping, and ω is the wave frequency. The damping is assumed to be a function of the wave frequency and the breadth to draft ratio, B/T, for the ship, in a later section it is described how b_{44} is calculated. The variation of the moment with ship heading can be included by multiplying the amplitude of the moment with the mean slope of the wave along the breadth of the ship, hence

$$F_4' = -|F_4| \frac{\sin(0.5kB\sin\beta)}{0.5kB\sin\beta} k\sin\beta \sin(k_e x - \varpi t)$$

where F_4' is the moment per unit length, and $k_e = |k\cos\beta|$ is the effective wave number. The expression for the mean wave slope is given in Bishop and Price (1979).
The total excitation moment is found by a longitudinal integration

$$F_4 = \int_0^{L_{pp}} F_4' dx = 2|F_4| \frac{\sin(0.5kB\sin\beta)}{\sin(0.5kB)} \frac{\sin(k_e L_{pp}/2)}{k_e} \sin\varpi t \qquad (1)$$

where the term $\sin(0.5kB)$ in the denominator is introduced in order to get the right amplitude in beam sea. Making $|F_4|$ a function of the encounter frequency ϖt instead of the wave frequency includes the forward speed effect.

It has not been possible to find a simple closed form solution for the sectional damping coefficient, b_{44}, in the literature and therefore it is necessary to determine it in an approximate manner. By the Frank close fit method the two dimensional damping coefficients was calculated for triangular sections with different B/T ratios thereafter a curve was fitted through the points and it was possible to find an relatively simple expression which can account for the different breadth to draft ratios and the variation of cross sectional area. The method gives the following expression for the non-dimensional damping

$$\frac{b_{44}}{\rho A B^2}\sqrt{\frac{B}{2g}} = a(B/T)\exp\left(b(B/T)\varpi^{-1.3}\right)\varpi^{d(B/T)} \tag{2}$$

where A is the cross sectional area of the submerged part of the section. The three functions a, b and d are assumed to be linear in B/T

$$a(B/T) = 0.256B/T - 0.286$$

$$b(B/T) = -0.11B/T - 2.55$$

$$d(B/T) = 0.033B/T - 1.419$$

An averaged value for the cross sectional area is found from the block coefficient

$$\begin{gathered} C_B = \frac{LA}{LBT} = \frac{A}{B^2}\frac{B}{T} \\ \Updownarrow \\ A = C_B B^2 \left(\frac{B}{T}\right)^{-1} \end{gathered} \tag{3}$$

It is now possible to calculate the excitation moment from Equations (1), (2) and (3)

The total damping coefficient B_{44} is simply found by

$$B_{44} = L_{pp} b_{44}$$

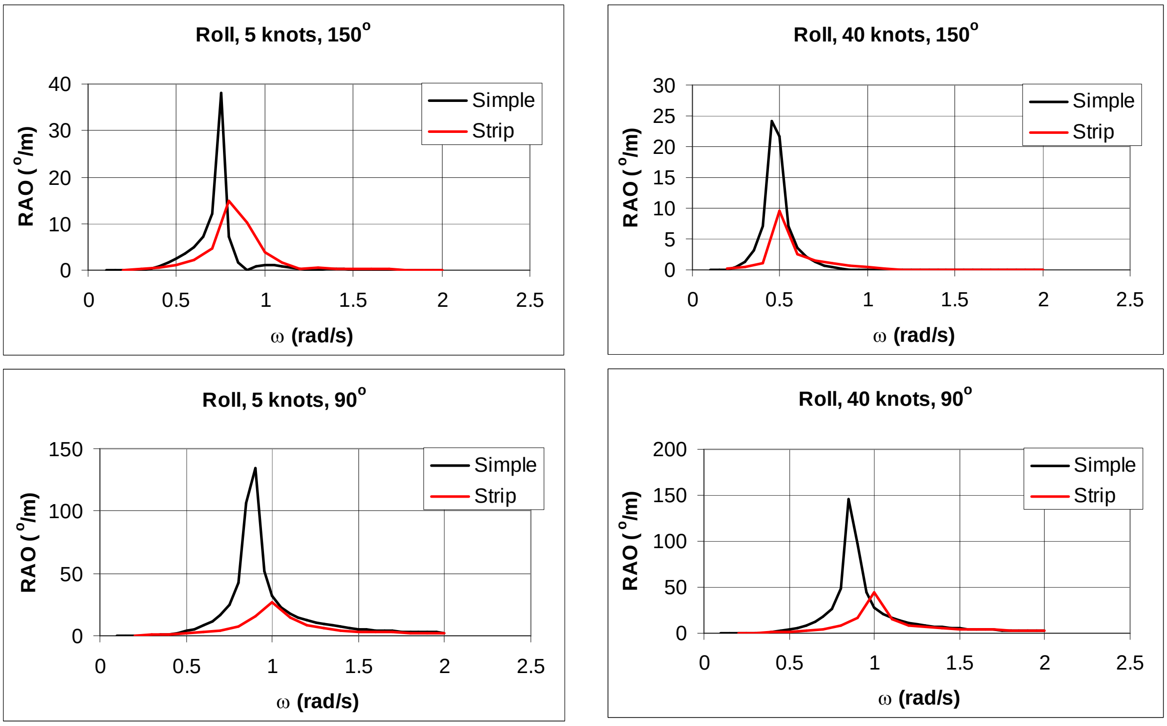

Figure 1: Comparison of transfer functions for roll. Top row 150° heading, bottom row 90° heading. Left column 5 knots speed, right column 40 knots speed.

Figure 1 shows a comparison of the results from a strip theory calculation and the simple expression. It is seen that the roll amplitudes for the simple expression is much larger than predicted by the strip theory, especially at resonance frequency. The simple theory also tends to under predict the frequency of resonance. An encouraging thing is that the simple expression is capable of predicting the right variation of the resonance frequency as a function of heading and speed. Thus for higher speed the variation of the peak frequency with heading is larger than for the lower speed.

The way to get a better agreement between the peak frequencies is to change the prediction of A_{44}. This will be investigated by looking for a better prediction of the added mass. Furthermore an additional damping will be introduced in order to get a better agreement at the peak frequency.

4.2.2 Dynamic stability models

Intact ship stability is of prime importance to Naval Architects since it is essential in order to ensure safe and efficient operation of ships. Although ship stability is a feature of paramount importance for the design of a ship, it still remains one of the most difficult aspects of Naval Architecture. Despite an immense amount of research, a universally acceptable solution of the problem of ship stability has not been achieved. The reason for the existence of this state of affairs is the complexity of the problem.

In order to fulfil the immediate needs to treat the problem of ship stability in an engineering context a simplified approach has been adopted and used over many years. However, the increased marine activities of both ships and other ocean going structures as well as the demand for safer and more efficient vessels raised the need for a better treatment of ship stability.

To assess ship stability more precisely, rationally and logically the ship's motions in a seaway should be taken into account and related to stability. With the advent of computers such an approach has become more realistic, making theoretical models and numerical simulations useful and fashionable.

Capsize in severe weather is a long standing problem in Naval Architecture. Even though ship capsizing is an extreme event, most ships run the danger of capsize in either intact or in a damaged condition. Small vessels are usually more at risk of intact capsize. Capsizing of vessels does not only represent material losses but also a great risk for human lives and depending on the nature of the cargo, considerable environmental damage.

One of the most severe modes of wave induced capsize of intact ship is broaching. It is considered the most dynamic mode of ship capsize, resulting from loss of controllability in severe following to quartering seas. The vessel experiences a sudden yaw motion that can take it off course and it may end up beam on to the wave direction despite application of maximum opposite rudder. This is frequently accompanied by large heel angles and can easily result in capsizing.

Since this is a very dangerous situation for most ships, the occurrence of this specific mode of capsize needs to be treated in a rigorous way. The actual capsizal mechanisms and vessel's behaviour at a broaching situation has to be clarified and the onset of the phenomenon has to be revealed. This is a laborious task because the vessel is a very extreme condition which makes studies of physics, either theoretical or experimental, difficult.

The dynamic motion of ships during broaching has excited many researchers to investigate the phenomenon, however broaching was not examined as a mode of capsize but as a loss of direction mechanism.

Attempt to study the phenomenon of broaching were mainly based on experimental evidence. The complexity in describing the dynamic behaviour has lead many researchers to experimentally modal various contributions separately and using a modular approach to simulate the phenomenon. This approach guarantees the accuracy of the concerned forces. However, it is not practical to implement. Experimental results are required for a very large range of conditions and are restricted to the investigated vessel. Furthermore, in the related studies investigation does not extend to capsize as experimental data for extreme motions are often unavailable.

Numerical models on the other hand, based on semi-empirical approaches are limited to three or four degrees-of-freedom neglecting the coupling effect of vertical motions which will greatly influence both the ability of the rudder and the phasing between heave and roll motion to determine broaching and capsize respectively. This is indeed a serious omission in mathematical models describing the dynamic turning of ships. A static equilibrium position has been used in some models to compensate for the vertical motion, however this does not represent realistically the dynamic acceleration and cannot model capsize. Six-degrees-of freedom numerical models have been developed, aiming to describe the ship motion in an extreme environment, but have not been used for an investigation of broaching induced capsize. The present research focus is in establishing a complete model to describe capsize by broaching. Emphasis will be placed on incorporating coupling of the planar and vertical motions to model capsize. Adopting such an approach would allow for the development of a mathematical model unique to the case of broaching and capsize in astern seas. Based on time domain simulation, this model will incorporate the hydrostatic and hydrodynamic complexities arisen when considering a ship manoeuvring in a wave environment. The increased computational power will enable a systematic parametric investigation to identify environmental conditions that are likely to induce a broach and therefore help enhance knowledge and insight concerning vessel safety.

Furthermore, the potential of recent advances in non-linear dynamics as practical tools is also investigated. Chaotic dynamics of simple deterministic systems have become a fashionable subject for applied engineers and high expectations are raised in achieving a global understanding in the behaviour of physical systems. However their relevance to ship motion dynamics has not yet been fully demonstrated. New concepts and ideas based on geometric representations of ship dynamics have been put forward but these are usually condensed with transient analysis.

4.2.2.1 The Definition of a Broach

It has long been appreciated that considerable difficulties often arise in vessels operating in following and quartering seas. Ships have been frequently described to behave in a disconcerting manner and mariners are often impressed by the liability of vessels to unpleasant and dangerous behaviour when steering in oblique seas.

Broaching describes the phenomenon of loss of directional stability of a ship in a wave environment and is related to both manoeuvring and stability properties of the vessel. During a broach the ship veers off course and in spite of the opposite rudder action finds itself broadside to the wave. This behaviour is encountered in following to quartering seas and is characterised by the large yaw deviations the vessel undertakes. The large rotational acceleration in yaw, allied by the unbalanced roll moments, forces the ship to large roll angles and eventually to capsize. The phenomenon is more severe when the vessel obtains a considerable forward velocity and is trapped in the front slope of a wave. Broaching is a highly non-linear phenomenon often leading to capsize.

The loss of directional control and the ensuing broaching behaviour largely depend on the velocity the vessel encounters the waves. In general, when the vessel is advancing at a speed near the wave celerity, it will be forced to move along with the wave. More particularly if the ship runs faster than the waves but with a low encounter frequency, it will be slowed down by climbing up a wave to reach the wave speed before it is accelerated on the down slope of the wave where broaching occurs. However, more commonly the ship travelling slower than the wave will be hit from astern and accelerated to the wave speed. It will then be caught by a wave and situate near the wave crest. This is the state of surf riding which may be experienced only momentarily. If situated on the down slope, broaching is most likely to happen as the vessel will be accelerated with a large downward speed on the front slope of the wave. At this instant the sudden wave induced yaw moment can turn the vessel beam on the waves. The wave yaw moment will depend on the ship heading and wave height and it will undergo variations as the relative position of the ship to the wave changes.

Of profound importance for the occurrence of broaching is the magnitude of yaw moment exerted by the rudder. If, despite the maximum rudder action, the wave moment cannot be rectified by the applied moment by the rudder the yaw angle will increase monotonically. The yaw moment can become so large that the vessel could turn round during its run on the down slope to find itself almost beam on to the wave direction. As the vessel turns rapidly, large roll angles are developed that most probably will lead the vessel capsize. This phenomenon occurs in a short time span by the action of one wave alone in contract to the cumulative yaw motion or pseudo broaching.

The complicated nature of broaching has encouraged researchers to investigate the principal causes leading to its occurrence. Three were the main causes examined: loss of directional stability, decrease of rudder force, and wave exciting yaw moment formed by the cross-flow drag acting on the bow and the stern of the ship in opposite directions. The assumption that broaching is caused by the loss of its inherent stability when the ship is travelling on the down slope of a wave, measured by the calm water stability criteria has proven to be incorrect. The decrease of the rudder force is dependent on the emergence of the rudder and will vary for different hull forms and rudder configurations. The primary cause of broaching is the sudden increase of the wave induced yaw moment, which the rudder is enable to counteract.

4.2.2.2 Mathematical Modelling

Considerable expertise has been gained and experience accumulated over the years on the phenomenon of broaching. Broaching itself however is neither a necessary nor a sufficient condition for a vessel to capsize. Because of this fact broaching is normally studied by means of planar motion models while the phenomenon of actual ship capsize following a broach is usually ignored.

Capsize by broaching requires the treatment of both problems of loss of directional control and dangerous roll motions intensified by the non-linearities of large amplitude motions, indicating that the coupling of longitudinal plane motions with the transverse plane motions is necessary. This coupling effect has been shown to be significant in steep waves and large amplitude motions [6]. In quartering seas, the yaw moment developed from the wave force acting in the transverse direction will be affected if the vessel is caused to heave, pitch and roll at the same time due to the change in the underwater volume. Surge and sway motion can alter the vessel position in respect with the wave and enhance the hydrodynamic cross flow effects causing the ship to yaw even further. Due to the astern seas environment, the emphasis is on hydrostatic forces rather than hydrodynamics. Froude-Krylov components need to be calculated accurately considering the instantaneous position of the vessel. Diffraction wave forces for the case of low encounter frequency need to be included in the formulation as their effect has been found to be considerable for the yaw motion [4, 2]. While hydrodynamic coefficients can be obtained approximately through the published literature, the need to obtain accurate coefficients is also realised. Though ideally hydrodynamic reaction forces could be computed in a more rigorous way, the work involved and the uncertainty whether the existing computational approaches are sufficient for the examined phenomena, does not justify their implementation in the present scope of work. Experimental procedures for their evaluation require extensive investigations while undertaken research has produced contradicting results [5,7,4]. Broaching is related to the directional abilities of the vessel and the reduction of stability due to the loss of directional control. This demands that the manoeuvring characteristics need to be modelled accurately.

Mathematical models are used to describe particular features of physical phenomena. In spite of remarkable advances in science, the accurate representation of a real phenomenon is not feasible. Hence, in a theoretical model, the predominant factors are emphasised and assumptions are made with the intention to realistically enclose all aspect of interest. A model will be formulated capable to reproduce in a realistic manner the actual ship behaviour in severe astern seas. In designing this model emphasis will be placed on the phenomenon of broaching and the potential dangerous situations leading to capsize.

Whereas in the context of manoeuvring theory the ship responds to external excitation created by its rudder and propellers and in the seakeeping theory the vessel responds to wave excitation, broaching is actually a parasitic motion of these two responses [1]. The motion of the vessel becomes directionally unstable as a result of the combined action of both kind of excitations. Vertical motions of the ship highly affect the effective steering and course keeping while the forward acceleration induced by a following wave significantly contributes to broaching.

The ship will be considered to operate in a wave environment and respond to its own propulsive and control systems as well as excitation and restraints caused by the environment. The vessel is assumed to be rigid body with its shape, size and mass distribution unaltered in time.

It will be assumed that viscous fluid forces can be treated independently of inviscid fluid effects. Although theoretically incorrect due to strong interactions, this is a convenient approach based on physical considerations. Often employed in marine applications, this is usually bearable assumption as viscous effects are confined to relatively thin boundary layers [3]. Therefore the treatment of fluid forces will be distinguished in two parts. In the calculation of wave excitation forces where the fluid is considered inviscid and the flow field irrotational, so that potential flow theory is valid. The second part includes viscous effects comprising rudder and propeller forces and other manoeuvring forces.

The phenomenon of capsize by broaching is encountered only at very low encounter frequencies. Furthermore the position and geometrical orientation of the vessel in respect to a wave is of great importance. Broaching is likely to happen when the vessel runs down the slope of a steep wave. Because broach and capsize are transient events and are usually completed during one wave action, regular waves are suitable for the modelling of the wave environment.

It is widely acknowledged that the stability of ships depends on the rolling motion, which also provides a solid criterion for the assessment of stability performance. However, other motions could significantly affect the stability and heeling of the ship either directly and indirectly. In extreme sea environments, coupling effects are stronger and models have to be very complicated to represent the real case meaningfully. Of considerable influence for the rolling motion are the hydrodynamic coupling of sway into roll and the hydrostatic coupling of heave with planar motions. In addition, a vessel is expected to undergo heading variations relative to the wave directions depending on its longitudinal distribution of the underwater volume where pitch motion will also be deduced. Furthermore, the oscillation of surging motion will determine the vessel's phasing with the wave. To accommodate this situation a coupled six degrees of freedom model could be developed.

Nevertheless, the importance of simplified model cannot be overlooked. In spite of the immense computational power mathematical models considering only the dominantly relevant degrees of freedom are still designed. Their advantage derives from their simplicity as well as from the need to limit the initial value space to manageable proportions. Most of the physical phenomena are described by a system of non-linear equations for which closed form analytical solutions are not available and their evolution has to be obtained through numerical integration. However, since the motion of a non-linear system depends on initial conditions, it is impossible to numerically investigate the complete pattern of the system responses. In the case that the initial value space is not of enormous dimensions, the behaviour of the system can be investigated in a global sense. Geometrical methods provide a useful way to analyse the complex behaviour of such systems. The analysis of non-linear systems includes the identification of equilibrium states, periodic motions, their stability properties and their evolution as some system parameters are changed.

In order to calculate the motion response of a marine vehicle, the external excitation to the vessel has to be considered as a suitable function of time. The excitation consists of self-imposed motions such as the motion of a rudder or propellers but most significantly of environmental forces from waves and wind. For most surface vehicles, the major excitation results from the effect of surface waves. External excitation forces and moments are classified into the following categories according to their physical relevance: wave exciting forces, resistance force, propulsive forces, steering forces, wind forces.

The theoretical treatment of forces while the ship is manoeuvring in an extreme wave environment, where viscous effects and complex flow are present, is not yet present. This treatment may not be necessary if the inclusion of complexities have no significant influence on the dynamics of the vessel. Incident wave forces, associated with the pressure of the incoming wave system, are considered critical and they could be evaluated non-linearly through the integration of the volume under the instantaneous free surface thus allowing for full coupling of the considered motions. Strip theory can be used to determine the diffraction forces, where the vessel is considered in an upright position and low encounter frequency is assumed. Viscous effects can be treated independently in an empirical, approximate manner. The ship resistance can be obtained by model experiments. Rudder forces can be evaluated based on the instantaneous relative flow velocity on the rudder and wind forces can be modelled in an empirical simplified manner.

In the following a reduced degrees of freedom model will be used in order to simplify the exhibited dynamical behaviour and also in order to reduce the initial value space into manageable dimensions. The intention is to use a simplified model, which is capable of realistically reproducing the phenomenon of broaching and capsize in order to gain a full understanding of the dynamics involved. The analysis will attempt to investigate the global behaviour of the system and study its stability properties.

Ship capsize by broaching is principally described by two motions. An extreme yaw action that will set the vessel off course and extreme heeling action induced by the turning motion that will capsize the vessel. These two motions are described in their simplest representation by a roll-yaw model, which is adopted here in order to investigate the phenomenon. Although broaching is commonly acknowledged as the most dynamic mode of ship capsize, it is believed that the capsizal resistance of a ship during a broach could be adequately assessed by considering only a roll-yaw mathematical model [5]. The restoring capability of the vessel should be accurately described including the influence from heave and pitch. Indeed for vessels operating in following seas the change of the underwater volume of the ship as the wave passes, alters significantly its transverse restoring moment. Contrary, surge motion can be excluded from the model provided that a critical value for the phasing between the vessel and the wave is used.

$$I_z \dot{r} + m x_G r U - m x_G z_G \dot{p} = N_{\dot{r}} \dot{r} + N_r r U + N_{r|r|} r\,|\,r\,| + N_R + N_W$$
$$I_x \dot{p} - m z_G r U - m x_G z_G \dot{r} = K_{\dot{p}} \ddot{\varphi} - z_Y \left(Y_{\dot{r}} \dot{r} + Y_r r U + Y_{r|r|} r\,|\,r\,|\right) + K_R + K_W$$

where :

- I_x, I_z are the moment of inertia of the vessel with respect to the body system in the x and z directions respectively,
- m is the mass of the vessel,
- x_G, z_G are the longitudinal and vertical distance of the centre of gravity from the origin O on the vessel,
- U is the forward speed of vessel,
- p and r are the angular velocities of the vessel : roll and yaw,
- φ is the roll angular rotation of the vessel,
- $N_{\dot{r}}, K_{\dot{p}}$ are the yaw-yaw and roll-roll added mass respectively,
- N_r, Y_r are the yaw-yaw and sway-yaw damping coefficients respectively,
- $Y_{\dot{r}}$ is the sway-yaw added mass coefficient,
- $N_{r|r|} r\,|\,r\,|, Y_{r|r|} r\,|\,r\,|$ are the yaw-yaw and sway-yaw manoeuvring coefficients respectively,
- N_R, K_R are the waterjet coefficients for the yaw and roll respectively,
- N_W, K_W are the wave exciting forces for yaw and roll respectively.

In the above formulation it is assumed that hydrodynamic and manoeuvring coefficients will be provided externally.

It is assumed that the ship tries to proceed along a prescribed heading angle while external causes result in heading deviations and heeling action. The effect of surge motion is neglected, so the speed of advance is assumed constant. Because of the omission of sway motion, the variation of the underwater volume with the lateral motion of the ship is neglected and the coupling effect of sway with roll motion is not considered.

The accurate prediction of broaching behaviour is served through the correct modelling of the yaw wave moment and its counteracting yaw rudder moment. These are the most influencing parameters and their calculation is performed in a precise manner for the instantaneous position of the ship. The capsizal resistance of the ship can be largely defined by the roll restoring moment, which can be modelled in a price manner for the instantaneous position of the vessel.

4.3 Methodologies used to develop and validate models

4.3.1 Seakeeping models

4.3.1.1 Introduction

The simplified ship seakeeping model, as described in section 4.2.1, will be developed and validated using reference data produced by a parametric seakeeping numerical analysis. In a first step, the numerical seakeeping tool used is selected through a benchmark of the different codes available in WP2. Afterwards, further validation of the tool will be performed by comparison with data from model tests and sea trials specifically performed in WP2.

Model tests data will also deliver additional information concerning whipping and local hydrodynamic loads for WP3.

Sea trials will also deliver data concerning noise, vibration, indoor climate and their effects, together with ship motions, on humans, for exploitation in Sub-task 2.2.1.

4.3.1.2 Numerical simulations

There are many different programs with different theories and different ways of implementing those theories but the most commonly used is 2-D strip theory. This theory has proved to give very good results for vessels with lower speed but there are doubts about its use for ships with Froude numbers higher than 0.4.

In order to use strip theory for the calculations it was required to do a study before in order to know how well this theory predicts the ship motions and loads of high speed craft.

This study was made using data available from the SEAWORTH project. From this project we obtained model test data for ship motions and loads of two high speed monohulls and this was used to compare with predictions from different computer programs.

The results from the programs and model tests were compared so that a decision could be made on which to use for the numerical simulations. These results are shown in the report number S102.00.13.050.001d (Seakeeping Predictions Using SEAWORTH Data) which is given on Appendix 1.

After analysing the results it was decided to use two programs for the parametric study, these were ShipmoPC and SEAWAY where ShipmoPC would be used for the prediction of motions and loads while SEAWAY would be used for the prediction of vertical accelerations.

The numerical calculations were done for the same ship length but with 5 different B/L ratios and 5 different T/L ratios. The basis vessel for the calculations was the SuperSeaCat3 and the ratios used were ± 20% and ± 10% of original values.

The mass distribution was based on the distributions of the SSC3 and FIN ships, where FIN ship is one of the vessels used on the SEAWORTH project.

The speeds for each simulation were 5 knots and 40 knots and the wave heading varied from 0^o to 180^o with increments of 30^o.

The results were calculated for the following points:

- Motions: at Centre of Gravity.
- Bending Moment/Shear Force: at Mid-ship and Forward Perpendicular.
- Accelerations: at Centre of Gravity and Forward Perpendicular.

The results of the numerical simulations are shown in the report number S102.20.13.050.003a (Numerical Test results) which is given in Appendix 2.

4.3.1.3 Model tests

The purpose of model scale experiments is mainly to deliver complementary data for large Froude numbers (0.6), and to check short term statistics on irregular seas. Model scale experiments will be performed at scale 1:20, on a segmented (two segments) elastic model (separation at midship). The longitudinal 2 node frequency of the model will be scaled. Two bow geometries will be tested in order to vary the bow flare coefficient. Tests will be performed at 40 kn speed, on head regular and irregular waves, with measurements of pitch and heave motions, VBM at midship, accelerations at midship and FP, and pressure on the hull, in the fore segment.

Detailed model tests specifications are presented in Appendix 3.

4.3.1.4 Sea trials

The purpose of sea trials is to give complementary seakeeping data for codes and models validations, to provide data on indoor climate, noise and vibration, and to collect information on their influences on human from people on board through the use of a questionnaire. Sea trials will be performed on the SuperSeaCat 3 vessel, with measurements of ship motions, accelerations (3 axis) at the rear, at midship and in the fore part of the ship, noise, humidity and temperature, vibrations in the rear part of the ship. Estimation of sea state will also be performed during sea trials by means of a wave radar (altimeter) mounted on the ship, of wave buoys and also by analysing ship motions.

Detailed sea trials specifications are presented in Appendix 4.

4.3.2 Dynamic stability

A mathematical model needs to be validated and verified before it can be used with confidence in the prediction of ship behaviour and its usefulness is established. The primary scope of this model is to describe the phenomenon of ship capsize by broaching. Therefore the result focuses on the conditions the vessel is expected to encounter. Before attempting to study the exhibited motion, the external forces and moments given as input to the system have to be validated. The correct evaluation of the force components will provide solid evidence for the accuracy of the mathematical model. Experimental results will be used for Self-imposed forces such as waterjet forces while theoretical results will be used for environmental forces especially wave forces. These forces will be used in equation of motion for the representation of the vessel motion during capsize by broaching and determine the trends and boundary curves derived from this model for broach and capsize zones. Experimental or other numerical results will be used to check the accuracy of the approximate results.

5. COST MODELS

The parameters of the risk models developed in Sub-tasks 2.2.2 and 2.2.3, and presented in section 4 before, concern mainly ship design parameters (dimensions,…) and operational parameters (speed, routes, se states,…). The formulation of cost models functions for these parameters should be performed together with the Shipyard, and in common with other work packages (these parameters are also input parameters in other work packages).

Discussions on cost models will be conducted during a dedicated session of the project meeting on 14th November 2002 and it is expected that more information concerning the development of cost models will be available after this meeting.

6. CONCLUSIONS

The works performed in Sub-tasks 2.2.2 and 2.2.3, and presented in this document, concern the formulation of risk/cost models concerning hazards related to ship motion with hull design and operational practice parameters.

Following the list of hazardous events related to ship motions established in the previous task (Task 2.1), two basic models, related to the ship seakeeping and dynamic stability (in broaching situation) characteristics have been formulated and will be used to derive the risk models. Methodologies for their development and validation, based on the exploitation of existing data, on dedicated numerical seakeeping analyses, model tests and sea trials, have also been specified.

The next steps will consist in refining and validating these models according to the defined methodologies, to define cost models, together with the Shipyard and other work packages, and to implement the final models in a tool form, for integration in the Project Tool.

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APPENDICES

Appendix 1 : Benchmark of seakeeping simulation codes

Safety at Speed - S@S

SEAKEEPING PREDICTIONS USING SEAWORTH DATA
INTERNAL DELIVERABLE NO. ID2.2.2

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001c	2002-05-03	ASO		DTU results added
001d	2002-05-21	AE		Final report

DISCLAIMER

CONTENTS

1. SUMMARY

Part of the work in Work Package 2 for the S@S project was to conduct a study for the comparison of the seakeeping prediction software used by different partners. The analysis is based on model test data available from the SEAWORTH project and this report contains the results of the study carried out at Newcastle University and at the Technical University of Denmark.

Three different computer programs were used to calculate motions and loads, these were SEAWAY and ShipmoPC, which are both commercial software and I-ship, which is developed at the Technical University of Denmark. The vessels modeled were one from Fincantieri and other from Chantiers de l'Atlantique and the simulations were done using conditions similar to conditions in which model tests were performed. Comparisons between the predictions and measurements were carried out.

S@S is the acronym for Safety at Speed, a project supported by the European Commission under the Growth Programme of the 5TH Framework Programme. The support is given under the scheme of RTD, Contract No. G3RD-CT-2001-00331.

2. COMPUTER PROGRAMS

The programs used are SEAWAY, ShipmoPC and I-ship.

SEAWAY is a 2-D frequency domain strip theory program (5 DOF), which has been validated for both monohull and twin-hull vessels. Results for vessels from tankers to high-speed craft and for catamarans as well as semi-submersibles were successfully validated.

The program gives a choice of using Korvin-Kroukovsky and Jacobs (1957) or Tasai (1969) strip theory methods. It also allows for the choice of methods for the calculation of potential coefficients to be either close fit and conformal mapping for deep water, conformal mapping for shallow water based on the theory of Keil (1974) or Frank's (1967) pulsating source theory for deep water.

The program also allows for viscous surge and roll damping effects on the computations but these are based mainly on empirical formulas. Anti-rolling devices can also be defined for the vessel to be analysed.

ShipmoPC is also 5 DOF Strip theory frequency domain software and is based on the theory by Salvessen, Tuck and Faltisen (1970) with improvements by Schmitke (1978) in order to incorporate appendage and viscous forces.

The program uses a near field method for the calculation of Added Resistance.

The hydrodynamic coefficients can be calculated by Frank close-fit method (1967) or by boundary element method developed by Sclavounos and Lee (1985).

Slamming calculations are based on Ochi and Motter method (1973) the calculations are for bottom slamming with the necessary condition of keel emergence. (Flare slamming not analysed).

I-ship is also a linear 5 DOF strip theory in the frequency domain based on the theory by Salvesen, Tuck and Faltinsen (1970). The added mass and damping are calculated with the Frank close fit method. The implemented method is in accordance with Potash (1970).

3. VESSEL FROM CHANTIERS DE L'ATLANTIQUE

The analysis on this ship was carried out for regular waves with ship speeds of 22 and 31 knots and wave heading of 180 degrees. General information for this ship is given on the table below.

Table 3.1: General information for CAT ship.

L_{PP}	164 m
B_{MLD}	20.6 m
D	18.8 m
T	4.5 m
Δ	6485 tonnes
KG	8.5 m
LCG	70.40 m

The results from the computer simulations are presented in the following sections. The model test results used for comparisons are the ones without pitch damping.

3.1 Heave and Pitch

Heave and Pitch RAO's are presented against the encounter frequency for both speeds. Figs. 3.1 to 3.3 show the results for 22 knots while the ones for 31 knots are shown on Figs. 3.4 to 3.6.

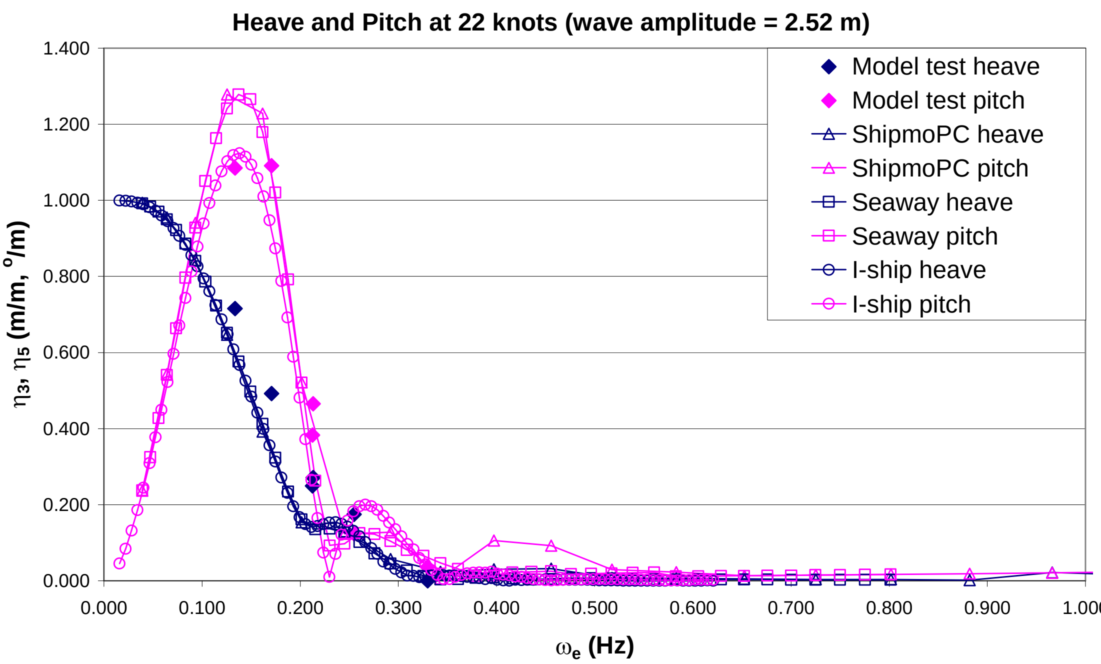

Fig. 3.1: Heave and Pitch RAO's for wave amplitude of 2.52 m and speed of 22 knots.

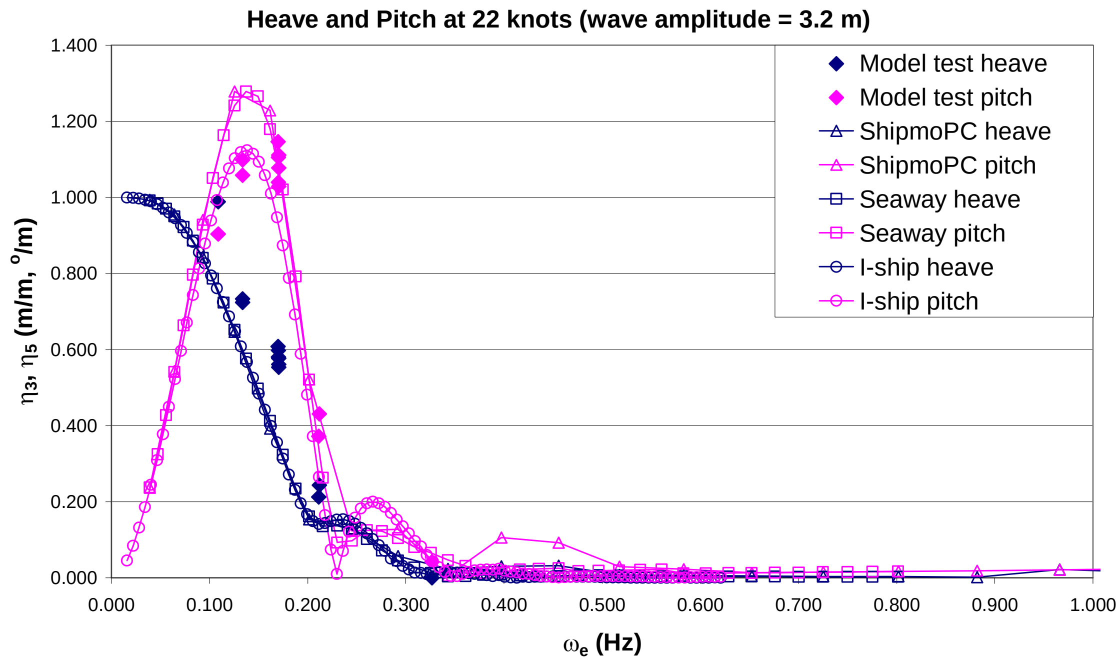

Fig. 3.2: Heave and Pitch RAO's for wave amplitude of 3.2 m and speed of 22 knots.

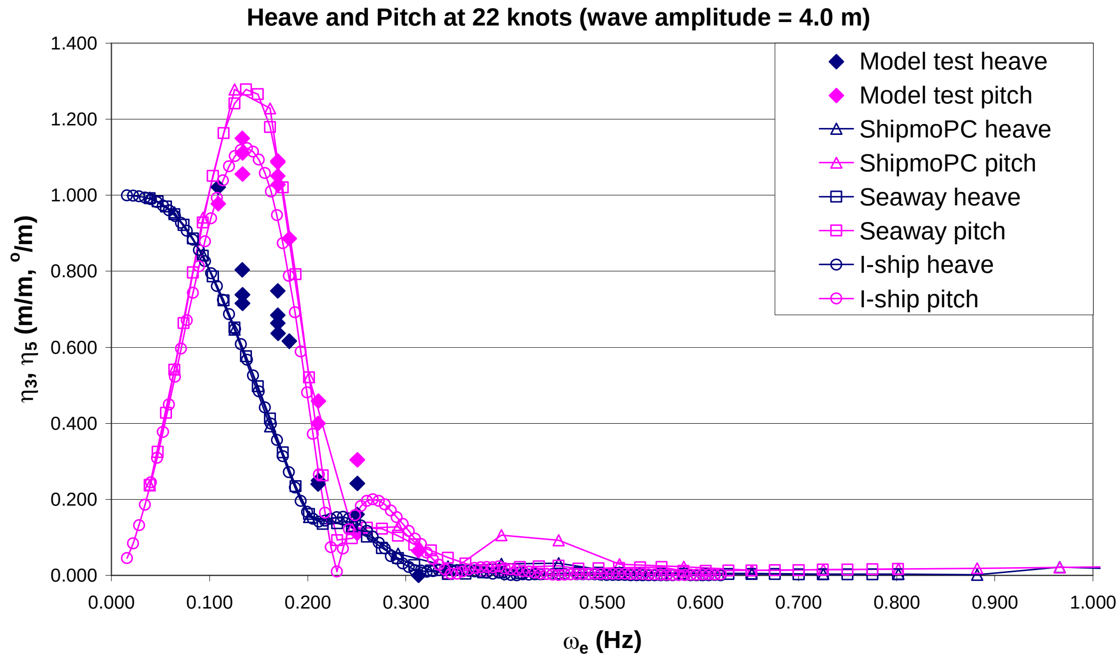

Fig. 3.3: Heave and Pitch RAO's for wave amplitude of 4 m and speed of 22 knots.

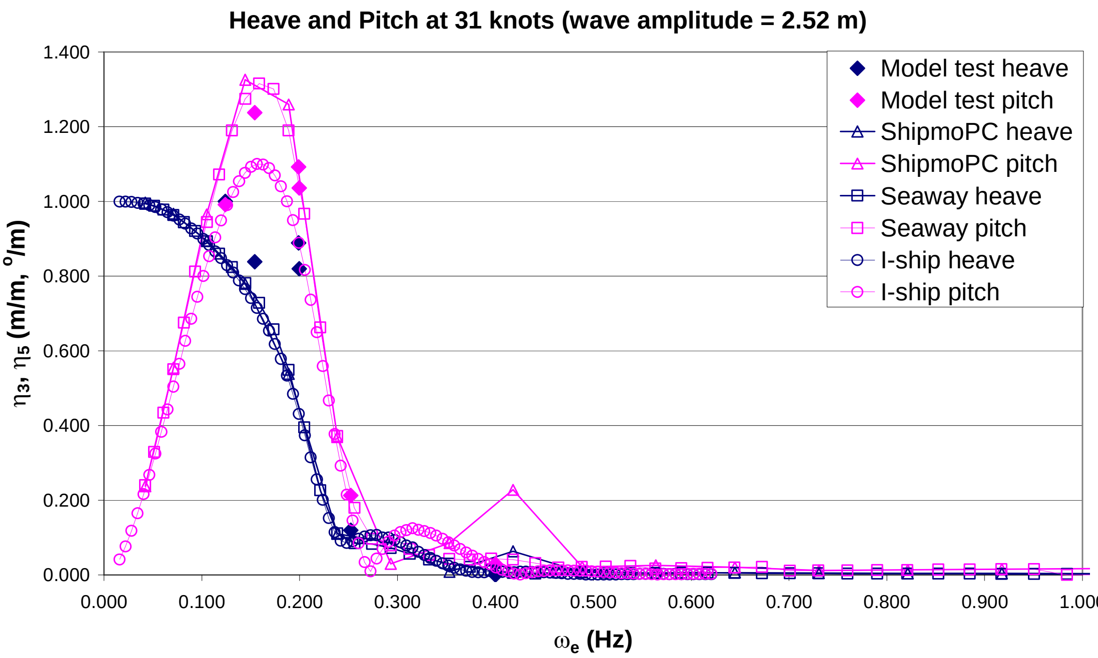

Fig. 3.4: Heave and Pitch RAO's for wave amplitude of 2.52 m and speed of 31 knots.

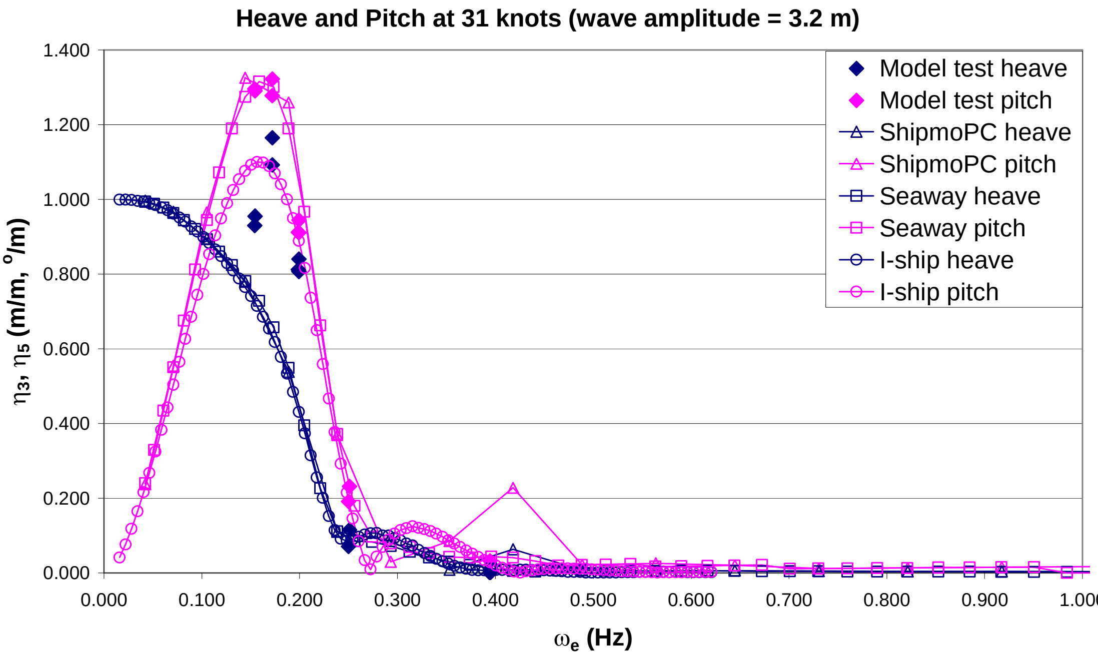

Fig. 3.5: Heave and Pitch RAO's for wave amplitude of 3.2 m and speed of 31 knots.

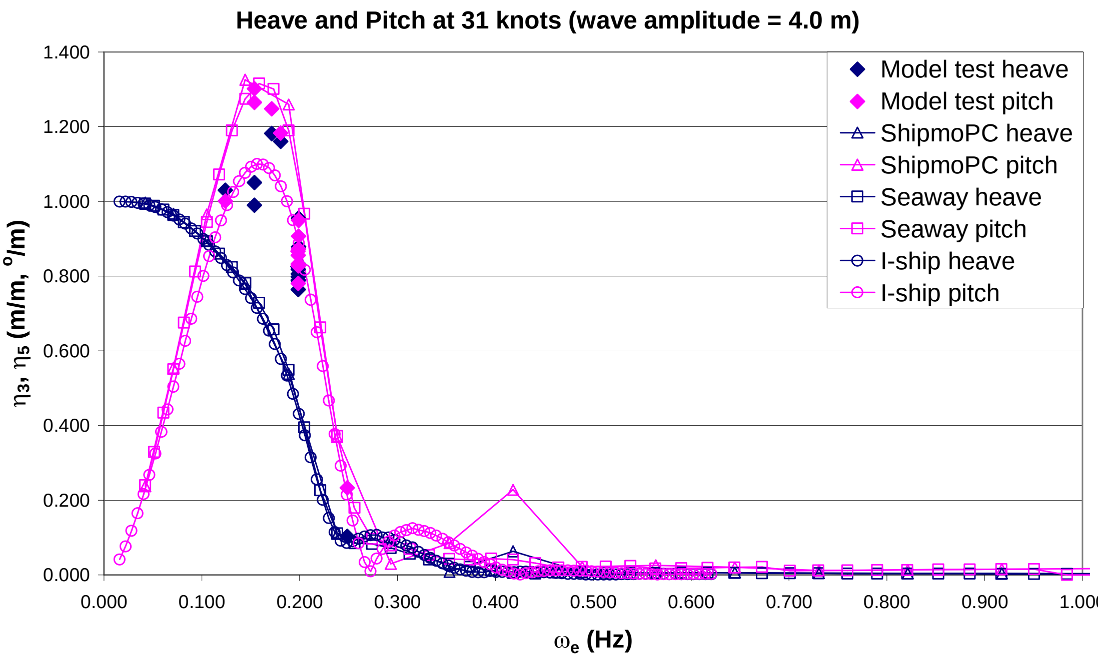

Fig. 3.6: Heave and Pitch RAO's for wave amplitude of 4 m and speed of 31 knots.

From these graphs we can see that the agreement in the heave motion is very good for all three programs. For the pitch motion there is a difference in the predicted maximum value, where I-ship is the lowest, whereas Shipmo PC and SEAWAY are identical.

The pitch predictions have a reasonable match with the model test results. For 22 knots I-ship predicts the maximum value best and for 31 knots ShipmoPS and SEAWAY predicts the maximum value best.

As for the heave results the values obtained from the programs are quite good for a lower speed and smaller wave amplitude, as speed and wave amplitude increase the difference between model test data and prediction results gets bigger where the software under predicts the results.

3.2 Bending Moment

Bending moments have been calculated at midship and at station 14, which is 114.8 m from the aft perpendicular. The results for both positions and speeds are plotted against encounter frequency and presented on Figs. 3.7 to 3.10 below.

The bending moments shown in the graphs are for the 3 different wave amplitudes that correspond to model tests.

From the graphs it can be seen that all the programs over estimate the results with ShipmoPC giving the lower values.

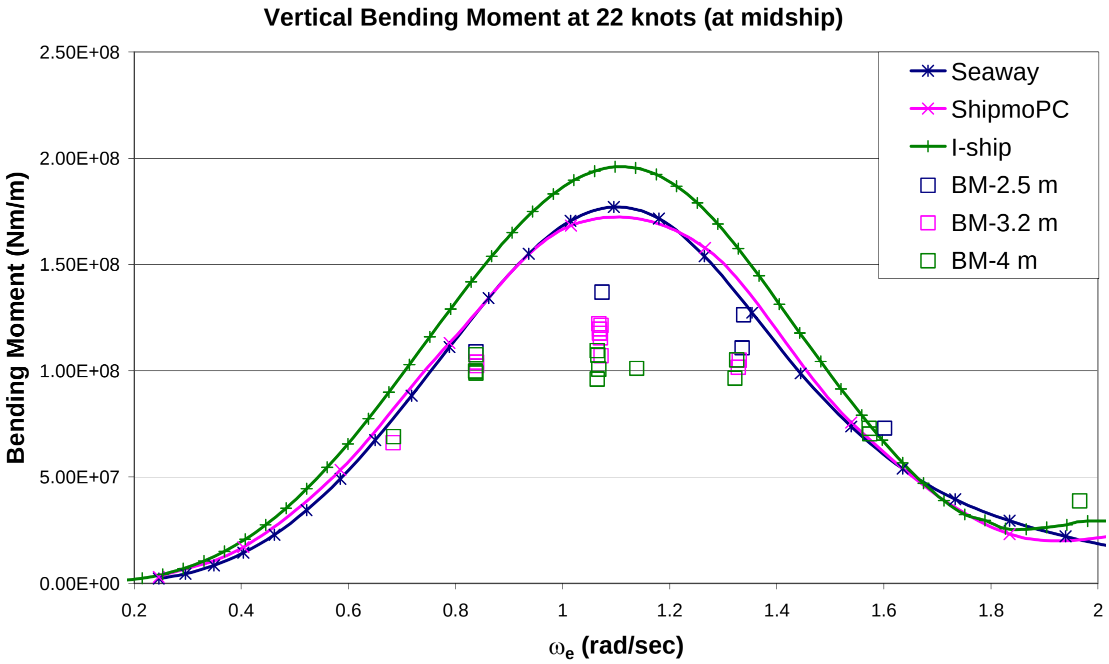

Fig. 3.7: Vertical Bending Moment RAO at midship and for 22 knots.

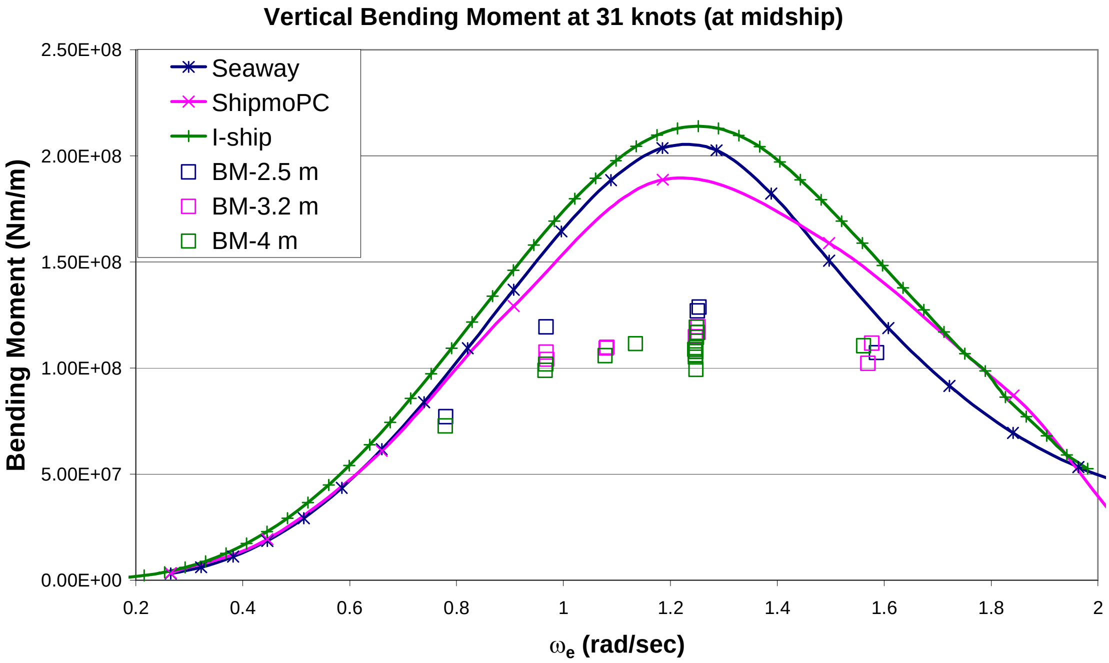

Fig. 3.8: Vertical Bending Moment RAO at midship and for 31 knots.

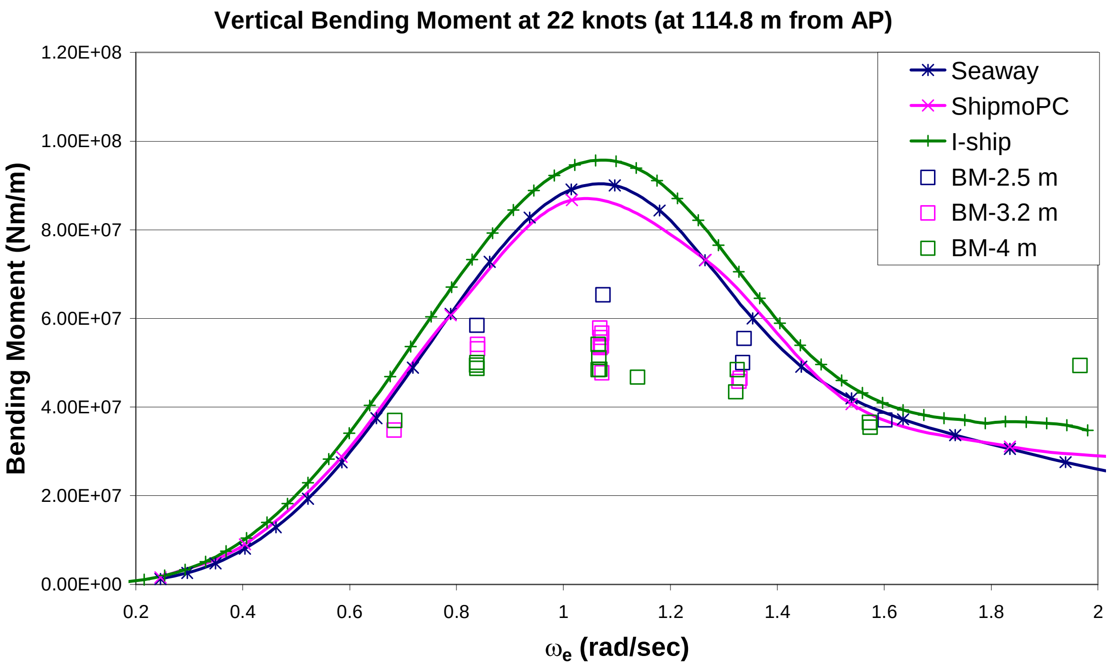

Fig. 3.9: Vertical Bending Moment RAO at station 14 and for 22 knots.

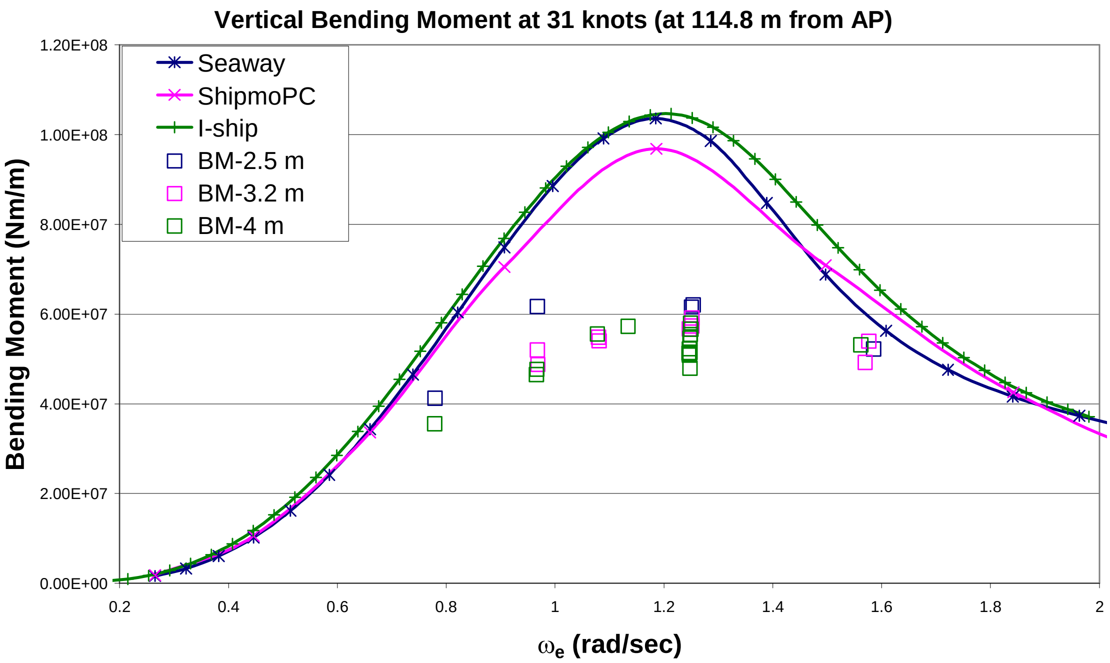

Fig. 3.10: Vertical Bending Moment RAO at station 14 and for 31 knots.

3.3 Vertical Acceleration

Vertical accelerations were measured at the centre of gravity and also at 139.5 m from the aft perpendicular, which is just after station 17. The results are presented against encounter frequency for the different wave amplitudes from model test.

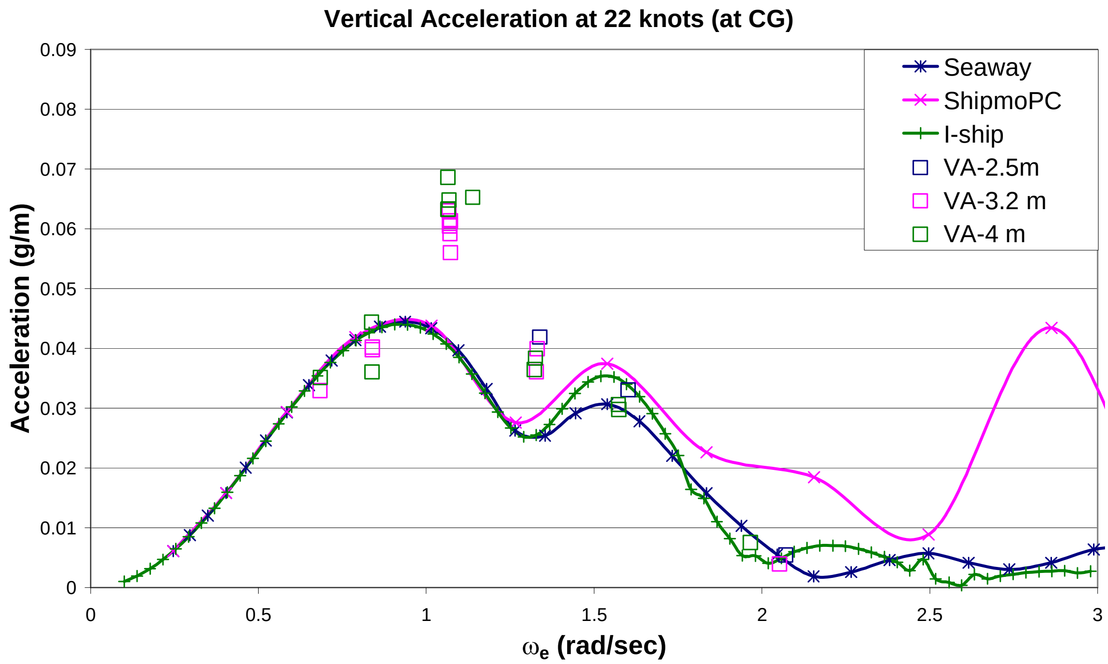

Fig. 3.11: Vertical Acceleration at centre of gravity and for 22 knots.

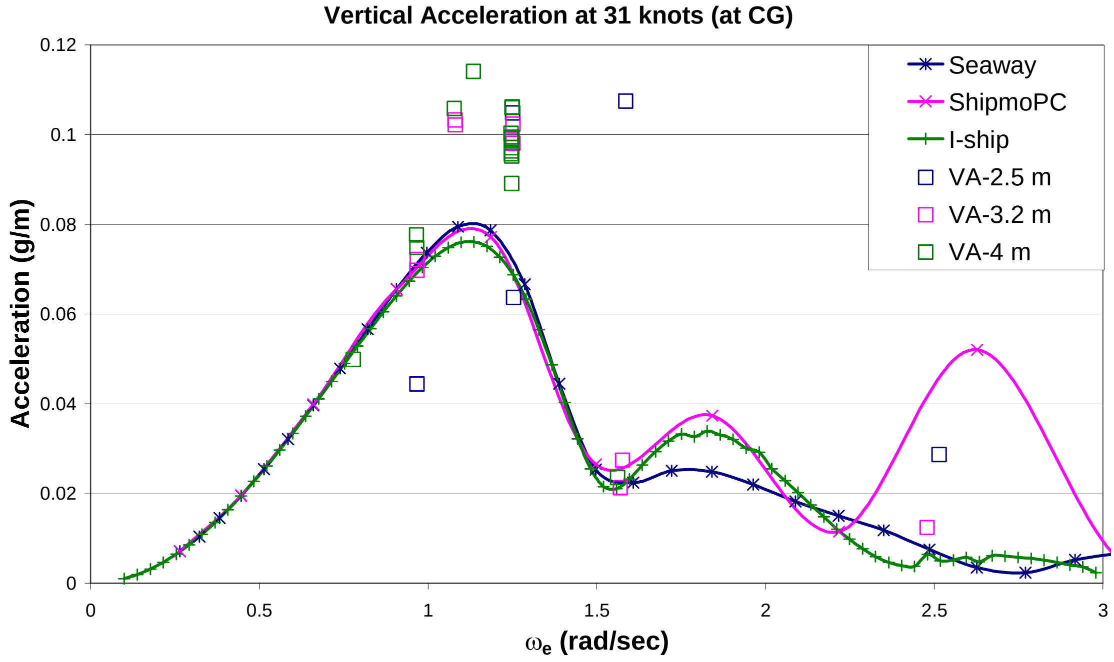

Fig. 3.12: Vertical Acceleration at centre of gravity and for 31 knots.

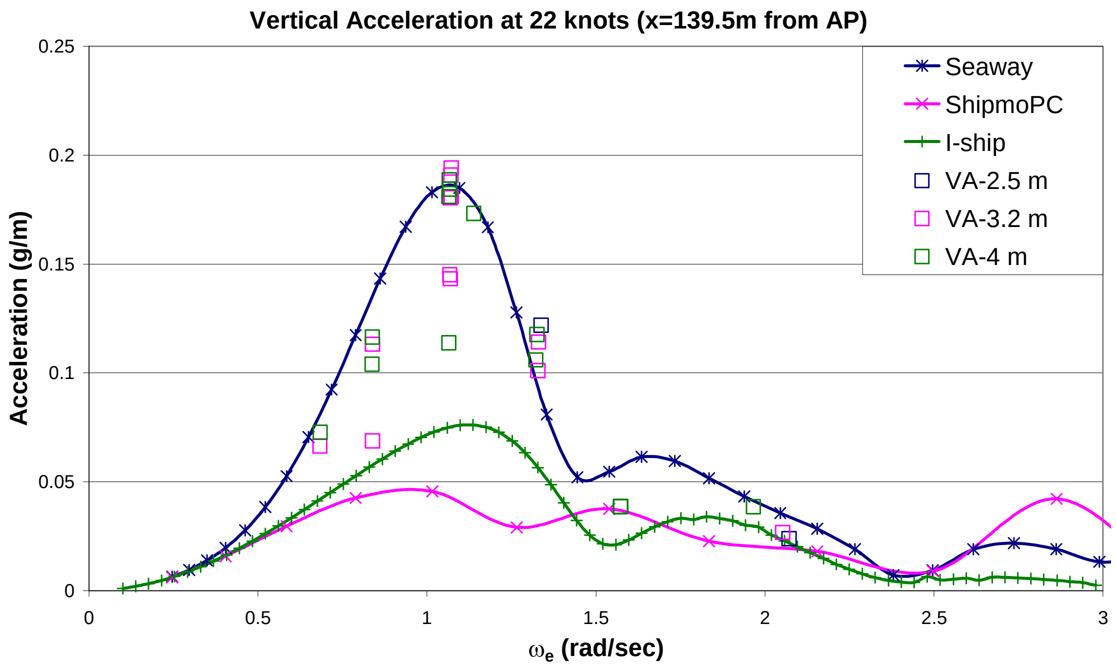

Fig. 3.13: Vertical Acceleration at 139.5 m from AP and for 22 knots.

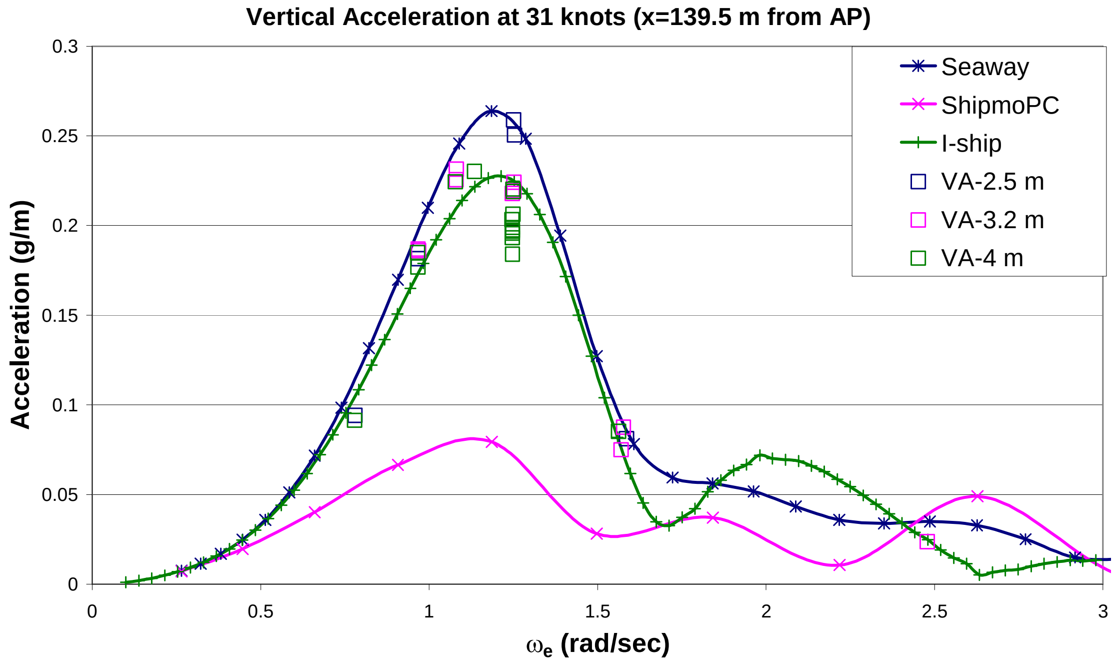

Fig. 3.14: Vertical Acceleration at 139.5 m from AP and for 31 knots.

At the centre of gravity all the programs give a similar result and although they are acceptable they don't correspond to the maximum vertical acceleration shown on the model tests. At 139.5 m from AP, SEAWAY gives the best result for 22 knots with the other programs under estimating the values while at 31 knots both SEAWAY and I-ship give a similar and good estimation but ShipmoPC under predicts the results.

4. VESSEL FROM FINCANTIERI

For the Fincantieri's ship model tests were carried out for regular as well as irregular waves with speeds of 30 and 40 knots and wave headings of 225° and 240°. The simulations were performed for the same conditions and the results are presented on the sections to follow.

The general information of the vessel is given in Table 4.1.

Table 4.1: General information for FIN ship.

L_{PP}	128.6 m
B_{MLD}	22 m
D	12.6 m
T	3.85 m
Δ	3856 tonnes
KG	8.1 m
LCG	50.27 m

The model test results were examined by spectral analysis using a Flat Top filter, which is defined as:

$$y_i = x_i\,[0.21557895 - 0.41663158\cos(\omega) + 0.277263158\cos(2\omega) - 0.083578947\cos(3\omega) + 0.006947368\cos(4\omega)]$$

For, i = 0, 1, 2, 3, …, n-1

$$\omega = \frac{2\pi i}{n}$$

n = number of elements in x

4.1 Regular Waves Results

The model test results correspond to the model with stiffness of the connection segments D1 and spring stiffness c2. These results were preferred to others as they are the ones that provide more data points for the comparison.

Sections 4.1.1 and 4.1.2 give the results for heave, pitch, shear force and bending moment RAO's.

4.1.1 Heave and Pitch

Figs. 4.1 to 4.4 show the results for heave and pitch RAO's for the different conditions that the tests were carried out.

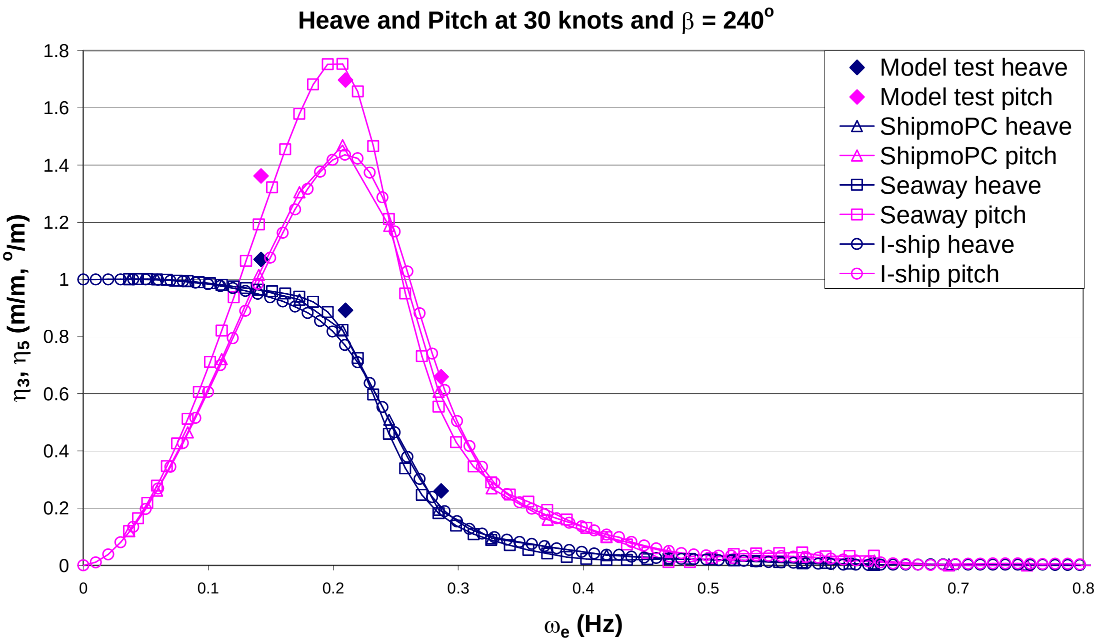

Fig. 4.1: Heave and Pitch RAO's for wave heading of 240° and speed of 30 knots.

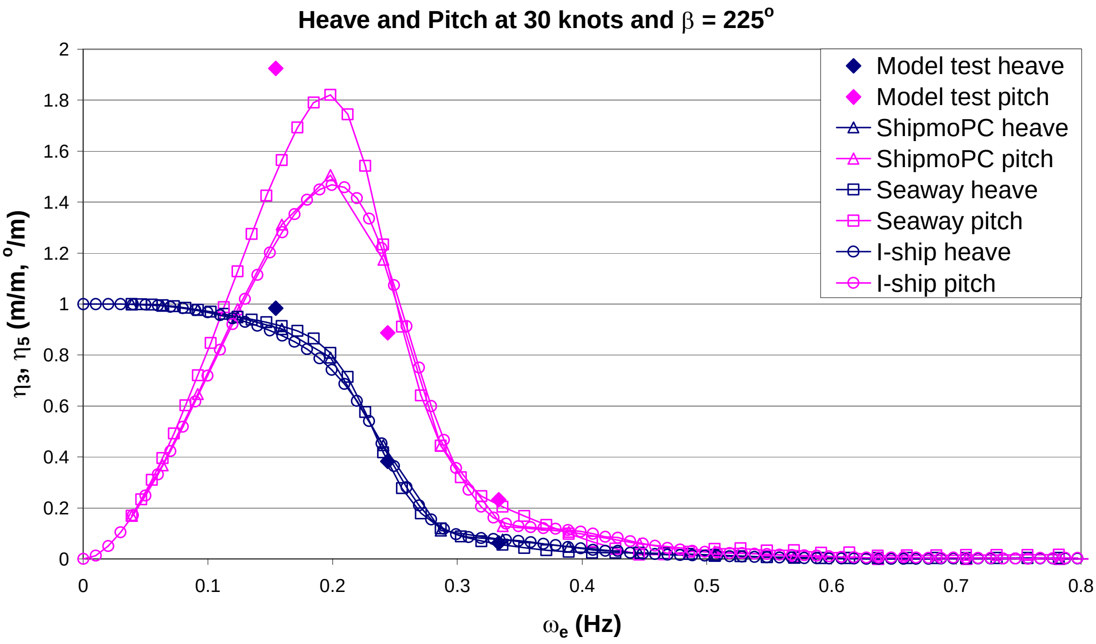

Fig. 4.2: Heave and Pitch RAO's for wave heading of 225° and speed of 30 knots.

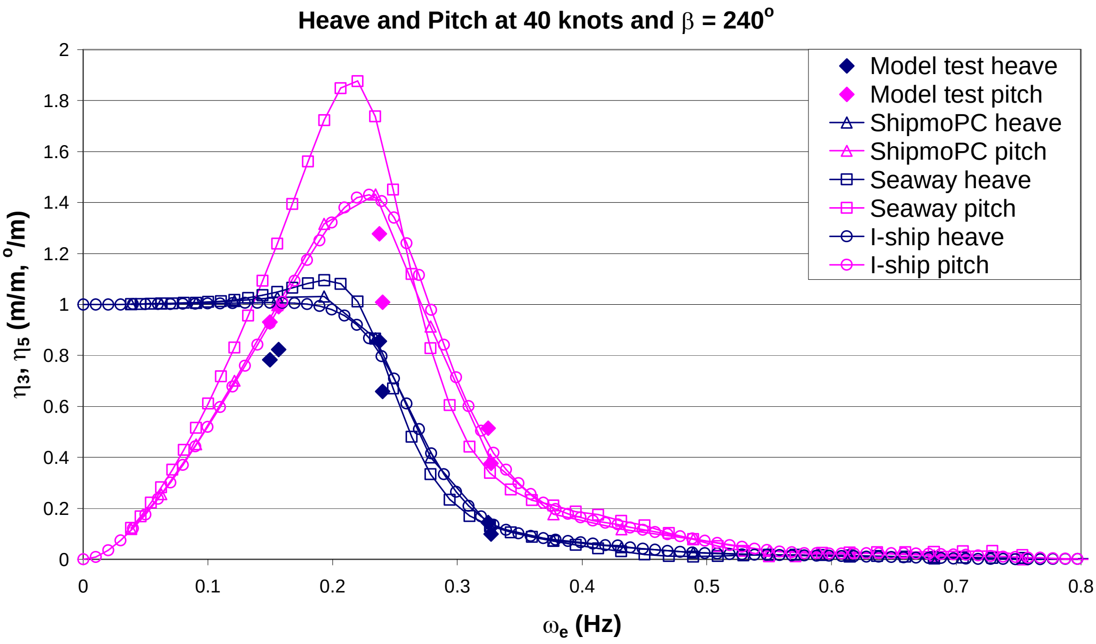

Fig. 4.3: Heave and Pitch RAO's for wave heading of 240° and speed of 40 knots.

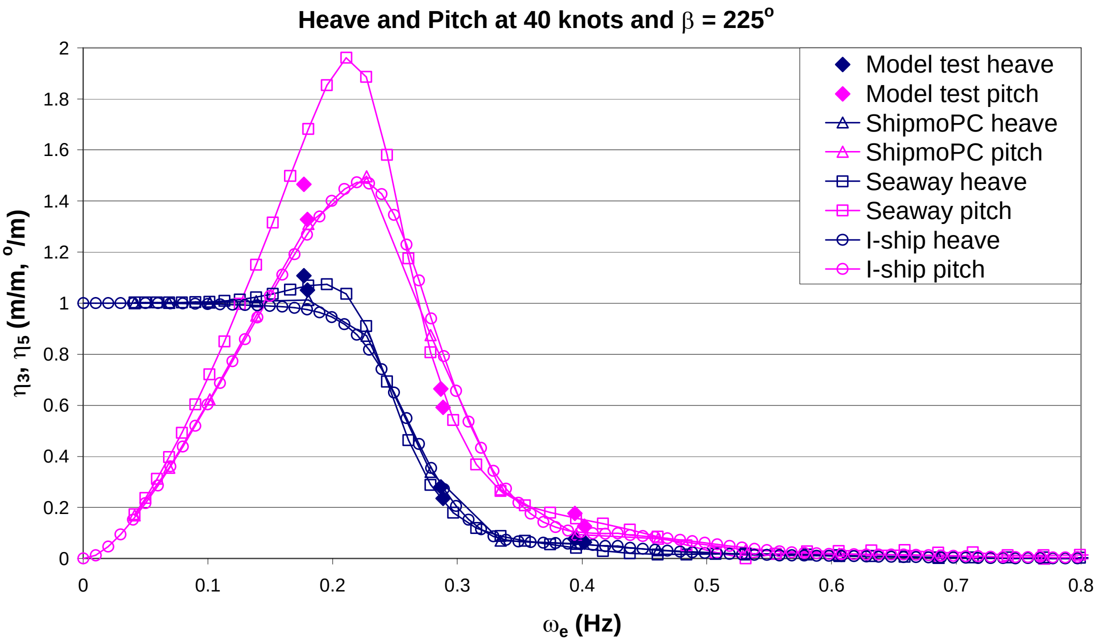

Fig. 4.4: Heave and Pitch RAO's for wave heading of 225° and speed of 40 knots.

From these results we can see that the heave predictions for 30 knots are almost identical for all three programs. For 40 knots ShipmoPC and I-ship are identical, whereas SEAWAY predicts a dynamic amplification, which is also seen in the model test results for 225° heading. For the pitch prediction I-ship and ShipmoPC are identical whereas SEAWAY predicts higher values in most of the frequency range.

The heave and pitch predictions agree well with the model test results. In the case of pitch Seaway gives a better prediction for 30 knots and ShipmoPC and I-ship give better results for 40 knots.

4.1.2 Shear Force and Bending Moment

Shear force and bending moment have been measured at midship. The results of prediction and model tests are shown in the figures below.

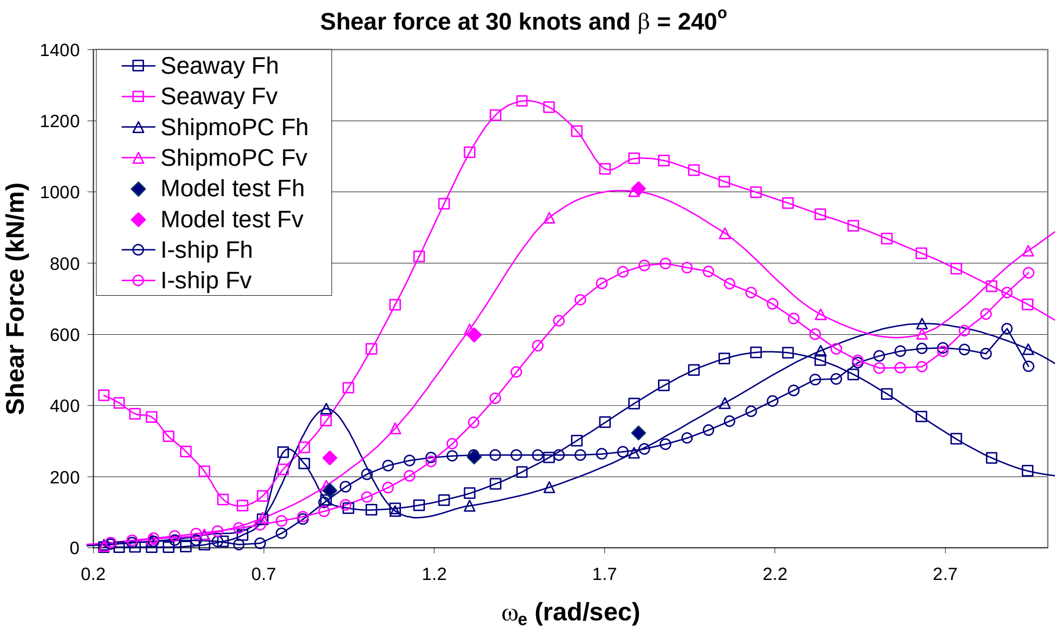

Fig. 4.5: Shear Force RAO for wave heading of 240° and for 30 knots.

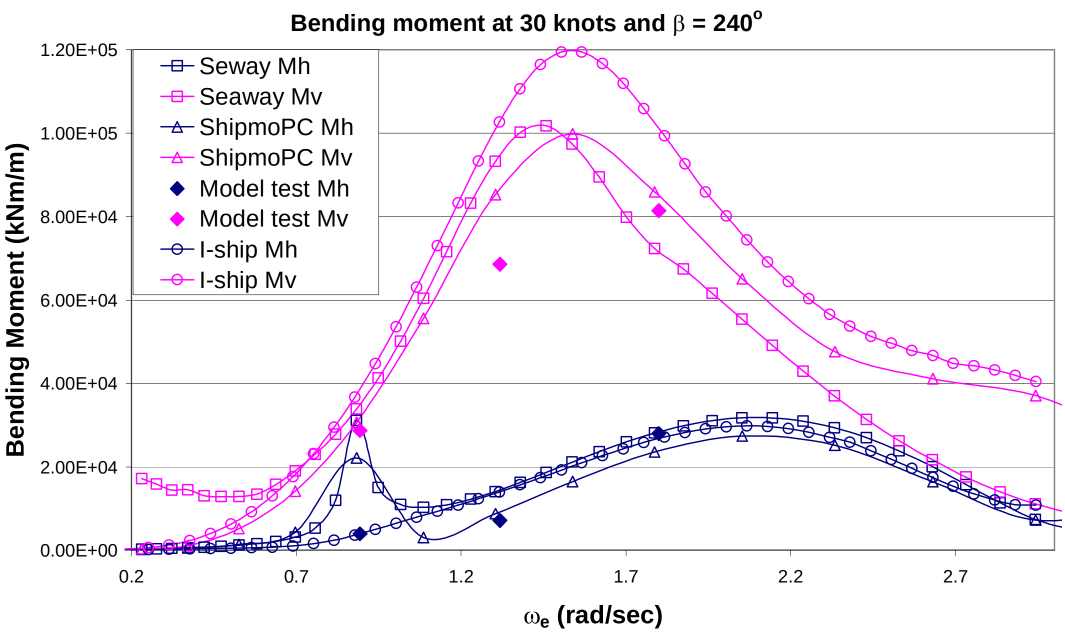

Fig. 4.6: Bending Moment RAO for wave heading of 240° and for 30 knots.

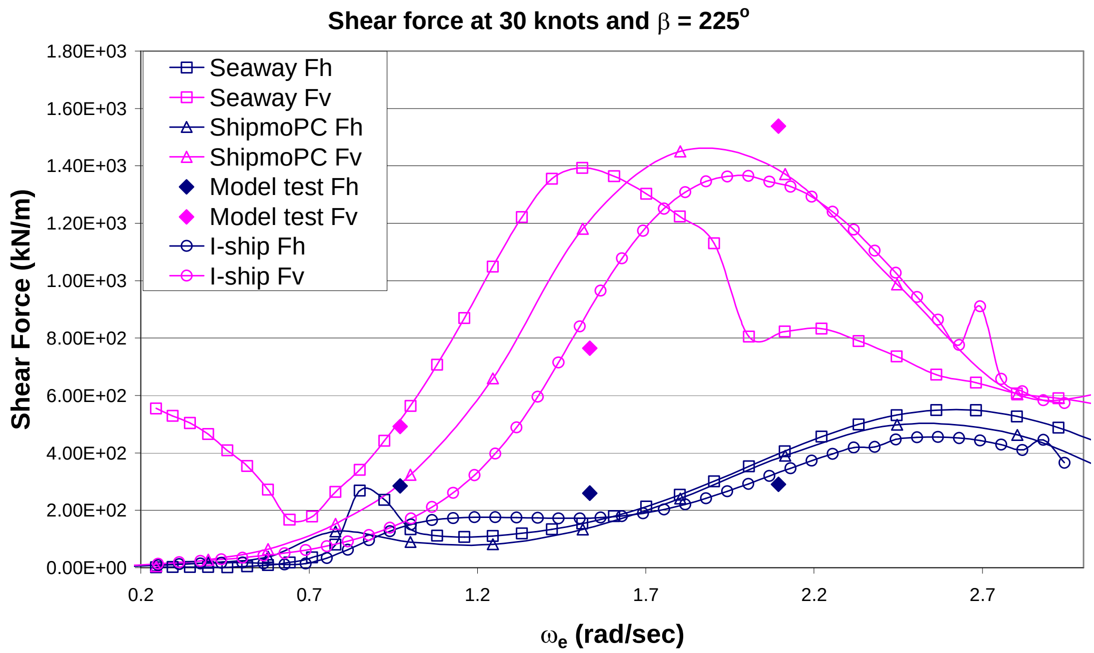

Fig. 4.7: Shear Force RAO for wave heading of 225° and for 30 knots.

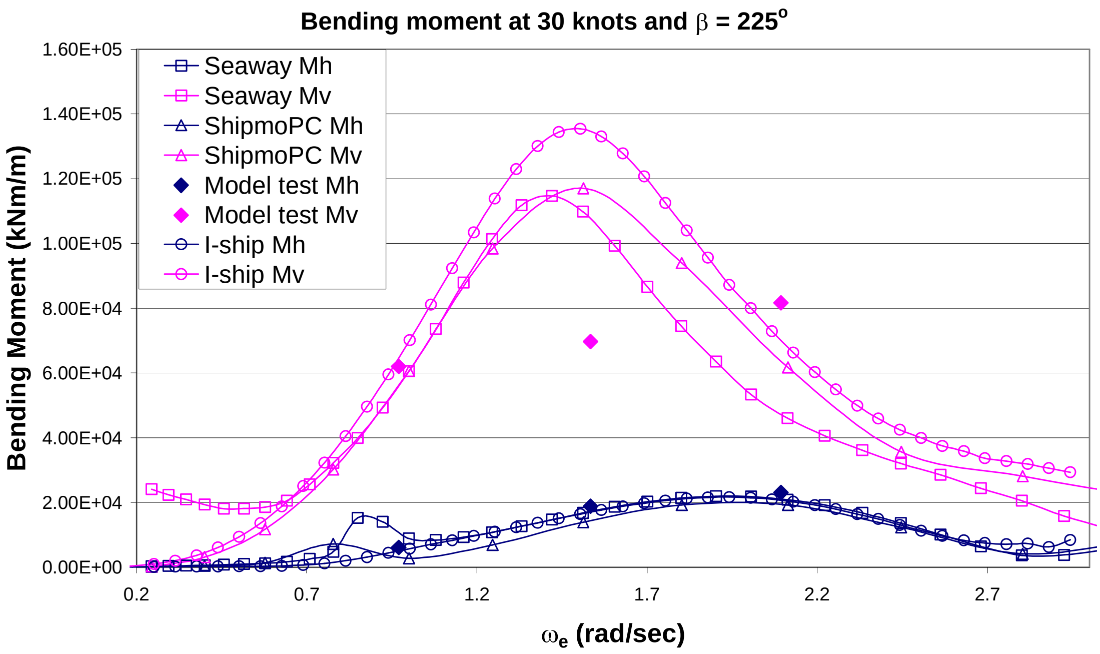

Fig. 4.8: Bending Moment RAO for wave heading of 225° and for 30 knots.

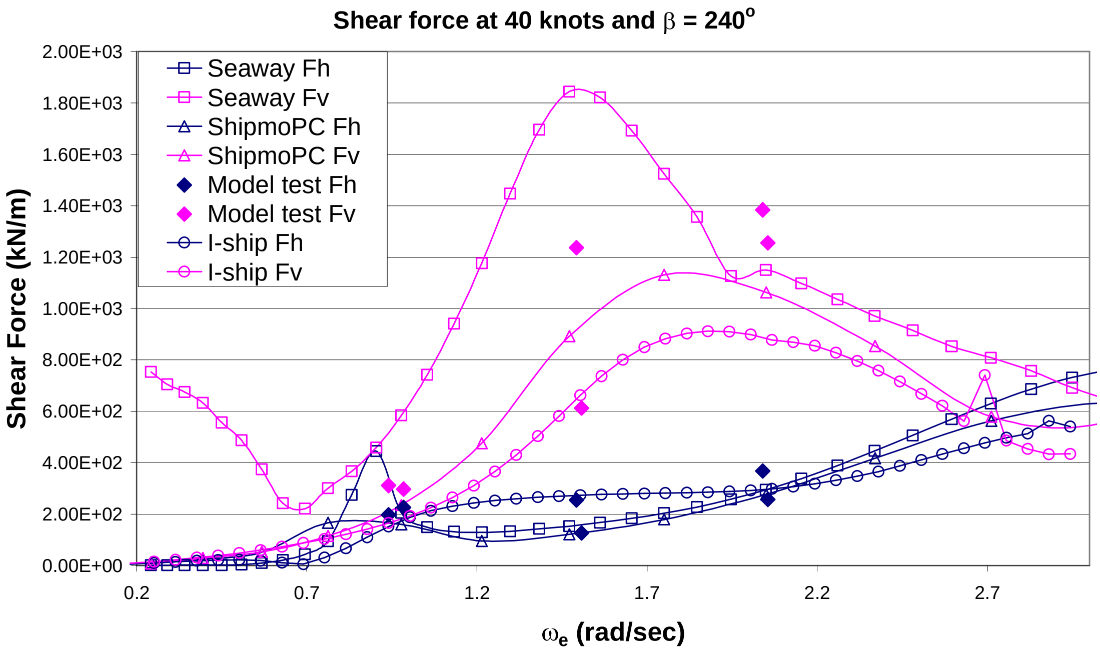

Fig. 4.9: Shear Force RAO for wave heading of 240° and for 40 knots.

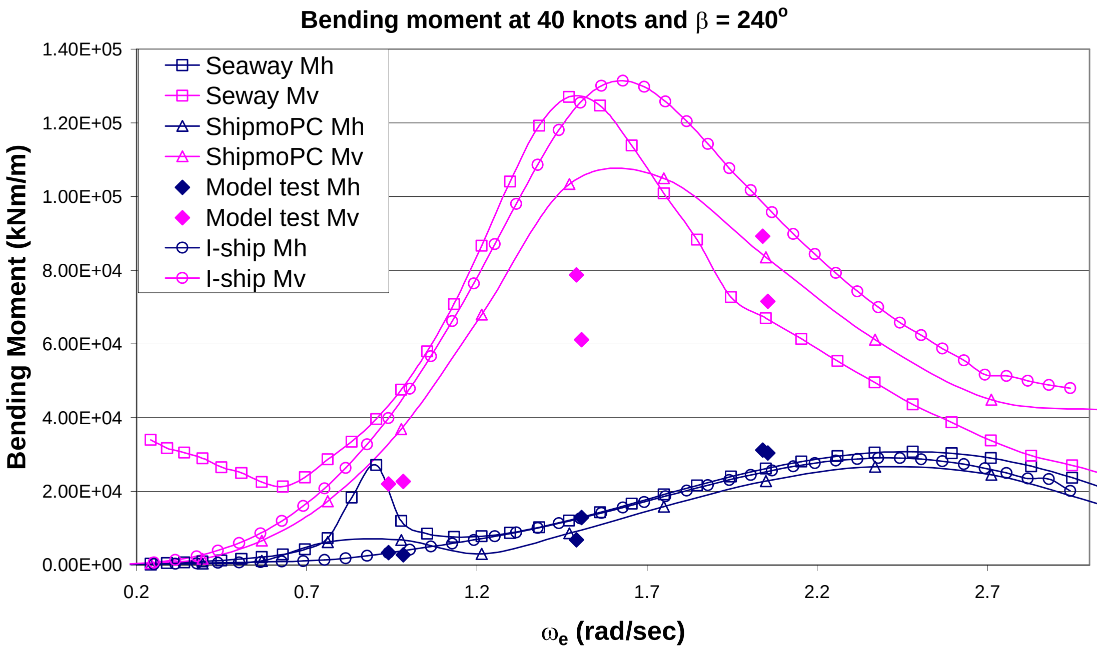

Fig. 4.10: Bending Moment RAO for wave heading of 240° and for 40 knots.

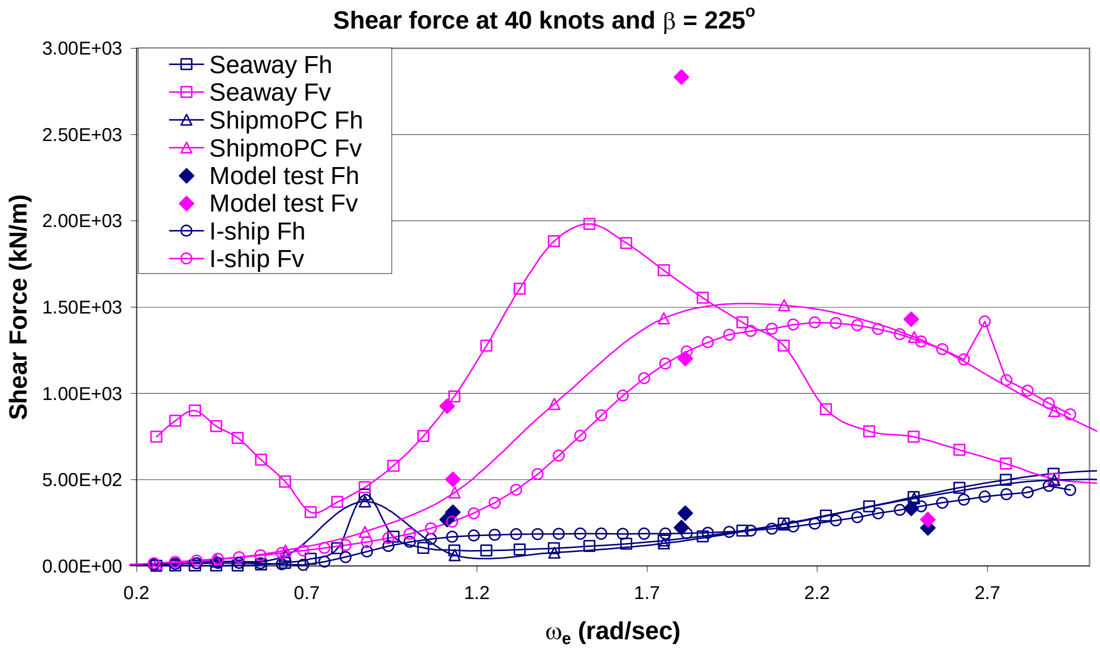

Fig. 4.11: Shear Force RAO for wave heading of 225° and for 40 knots.

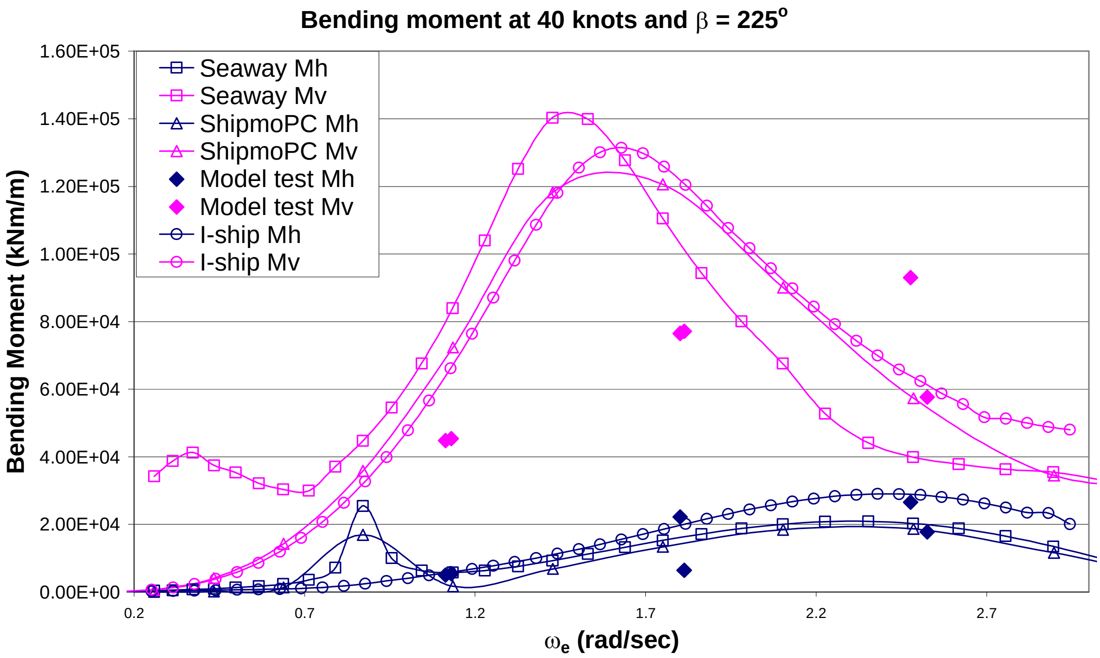

Fig. 4.12: Bending Moment RAO for wave heading of 225° and for 40 knots.

From the results presented on the graphs the ones for horizontal shear force and horizontal bending moment are the best ones. For vertical loads there seems to be a larger error between predictions and model test results as well as a larger difference between the predicted values.

It must be noted that some of the model test results might be dubious since some of the values are quite different even around the same wave frequency.

4.2 Irregular Waves Results

The programs were run for conditions similar to the ones used in model tests. The JONSWAP wave spectrum was used for significant wave height of 2 and 4 meters and a peak period of 8 seconds.

The results presented here are for H_{Sig} = 2 m, T_P = 8 sec and speed = 40 knots, which is the condition for which model test data is available.

4.2.1 Heave and Pitch

The results for heave and pitch significant amplitudes are presented separately and against the wave heading (β). These can be seen in Fig.s 4.13 and 4.14 respectively.

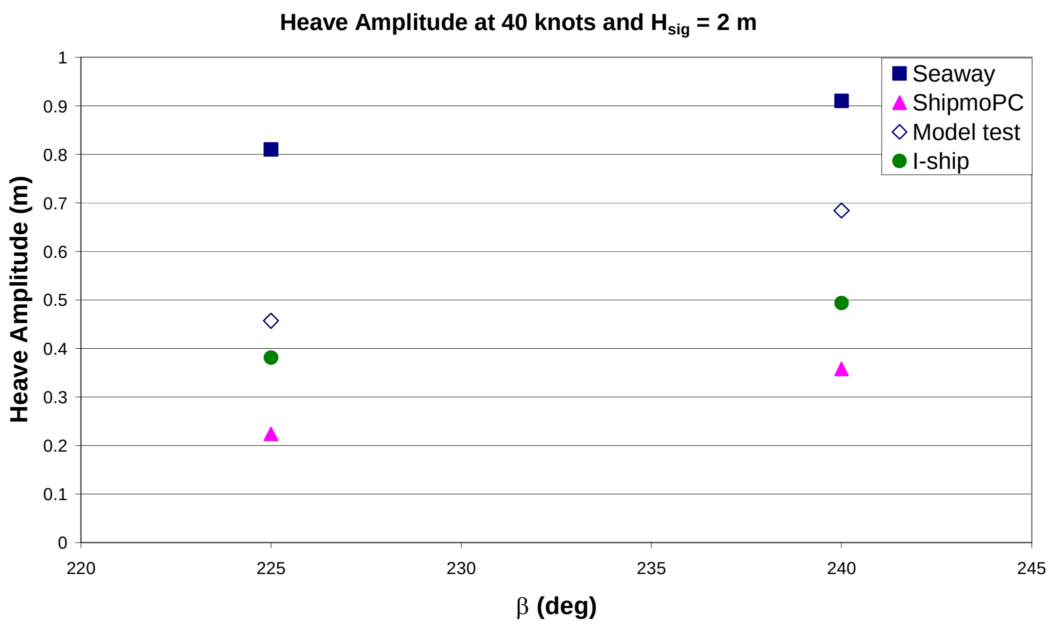

Fig. 4.13: Heave Significant Amplitude at 40 knots.

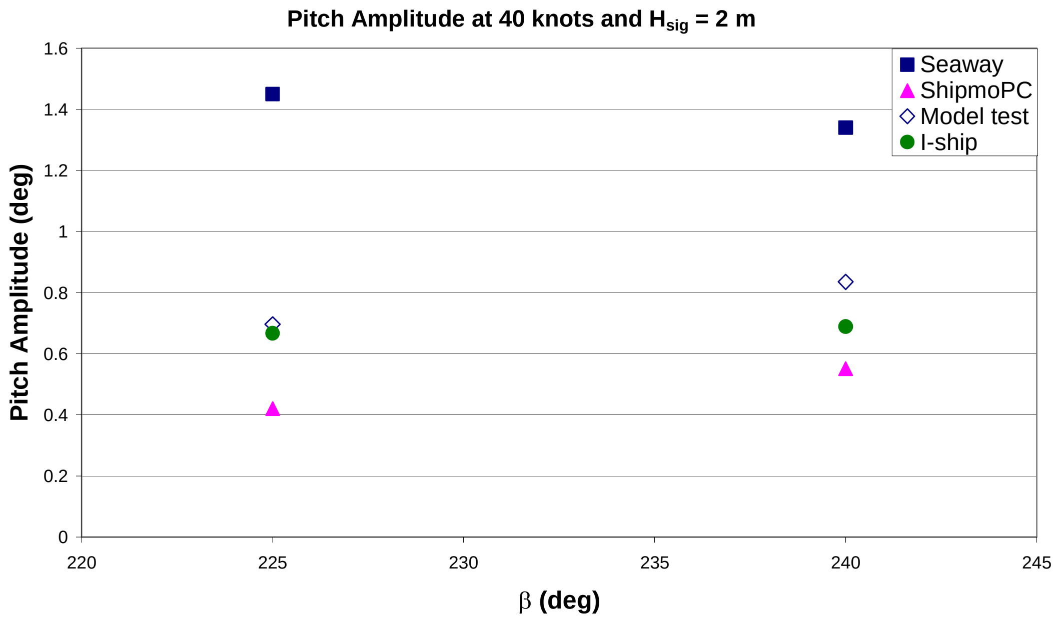

Fig. 4.14: Pitch Significant Amplitude at 40 knots.

From the results shown here we can see that I-ship predicts the amplitudes reasonably well and that the differences for the ShipmoPC and SEAWAY results are larger. For both heave and pitch Seaway over predicts and ShipmoPC and I-ship under predicts the results.

4.2.2 Shear Force and Bending Moment

Shear force and bending moment results at midship are presented below also against wave heading (β).

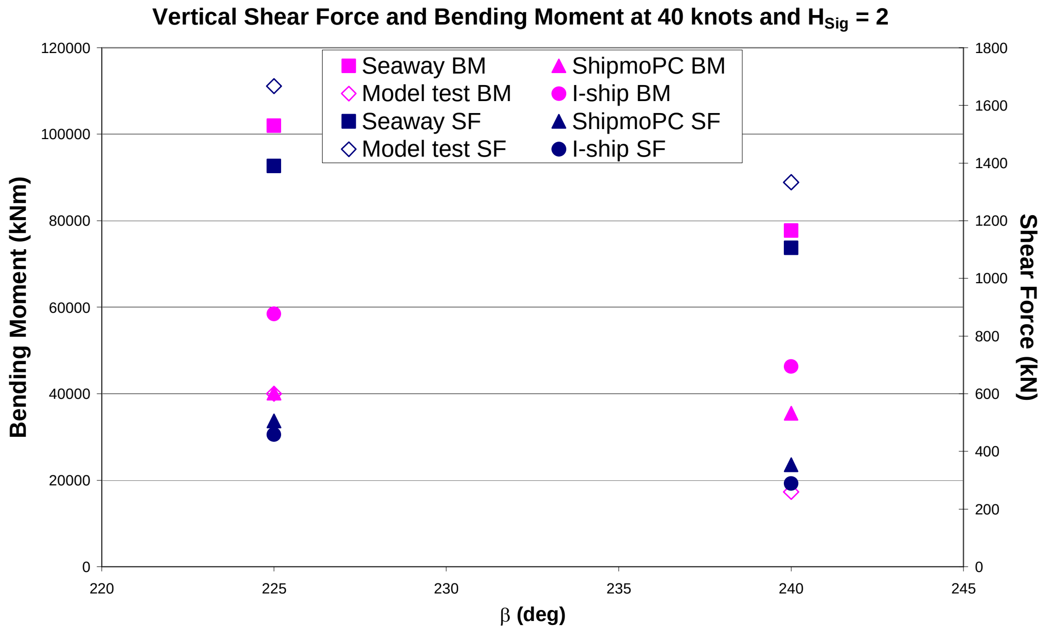

Fig. 4.15: Vertical Shear Force and Bending Moment at 40 knots.

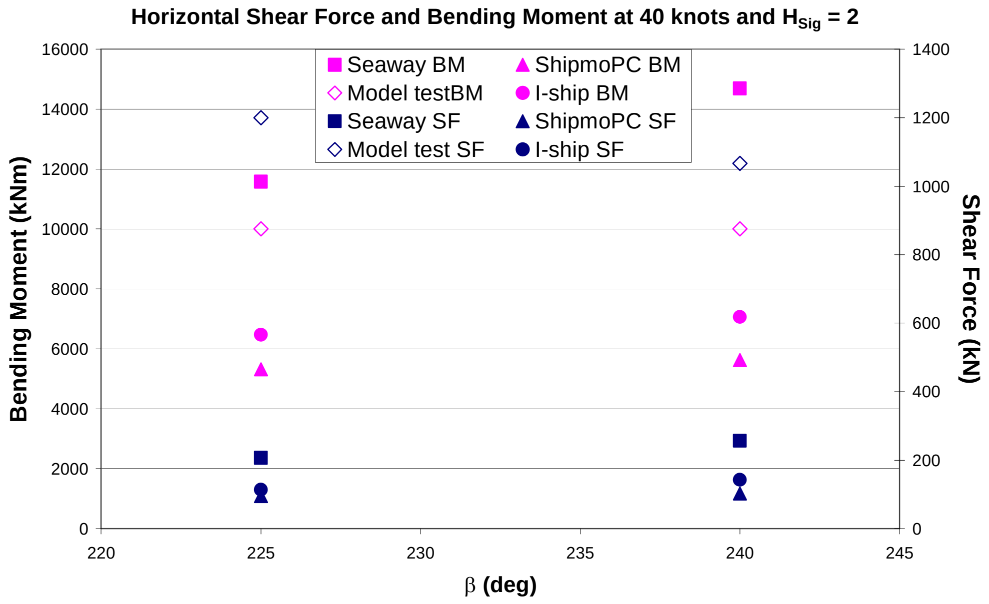

Fig. 4.16: Horizontal Shear Force and Bending Moment at 40 knots.

The results obtained are not very encouraging. The values for shear force are completely away from the model test results. For the bending moment calculations the differences are smaller but still not very reliable. The results from ShipmoPC and I-ship are similar, which is encouraging since both programs are based on the same theory.

4.2.3 Vertical Acceleration

Vertical acceleration was measured at different stations and the results shown below.

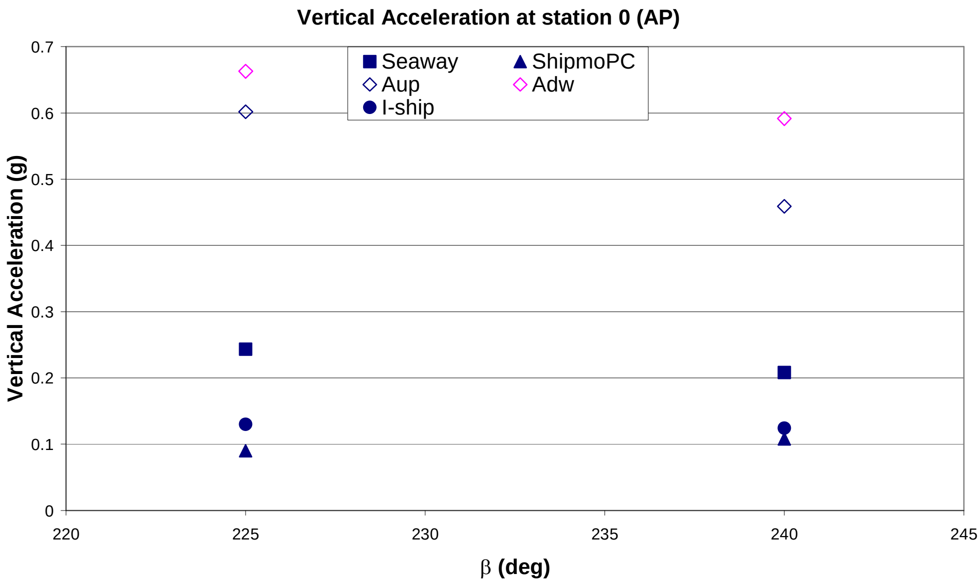

Fig. 4.17: Vertical Acceleration at AP.

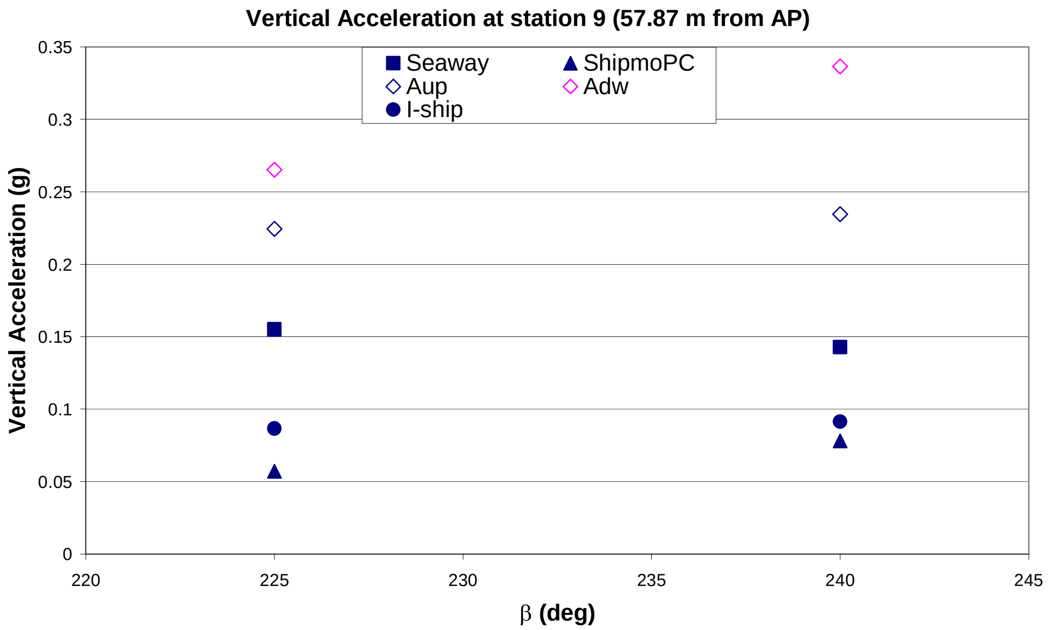

Fig. 4.18: Vertical Acceleration at station 9.

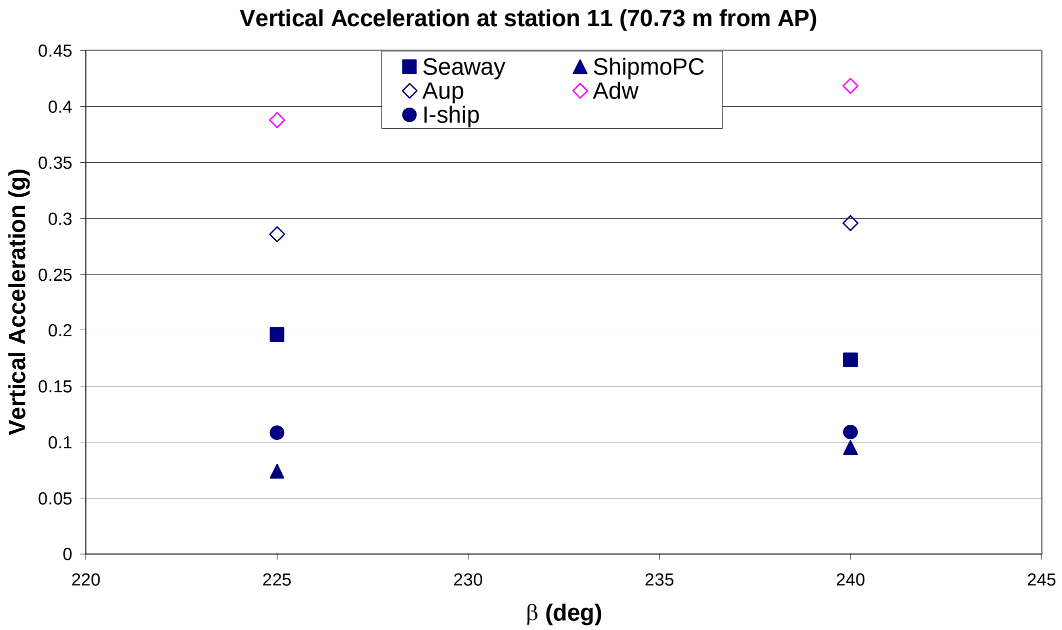

Fig. 4.19: Vertical Acceleration at station 11.

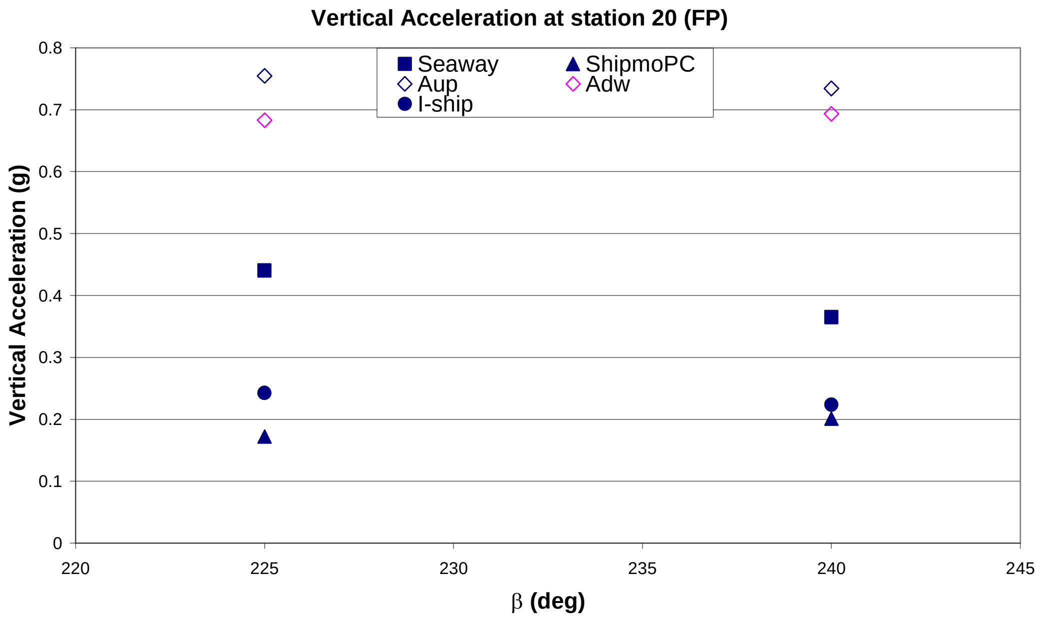

Fig. 4.20: Vertical Acceleration at station FP.

The results shown above are quite discouraging. All the predicted values are under estimated with ShipmoPC giving the worst results.

4.2.4 Slamming Pressure

For the slamming calculations keel emergence is required, flare slamming is not analysed. Because of this and also since there are only two sets of model test results for irregular waves slamming pressure can only be compared at two stations. These are for pressure panel at station 17.5 and pressure gauge at station 16.75 and the results are shown in Fig. 4.21 below.

If model test data were available for the other tests carried out (H_{Sig} = 4 m) further comparisons could be made and more reliable conclusions can be drawn as more data becomes available.

From the graph we can see that for the pressure gauge the results are very different while for the panel they are more reasonable. This is quite acceptable since for the panel we look at the mean pressure on the area while for the gauge it is just at that point.

It is quite difficult to make a definitive conclusion but the results obtained from the panel pressure can be encouraging start for further work.

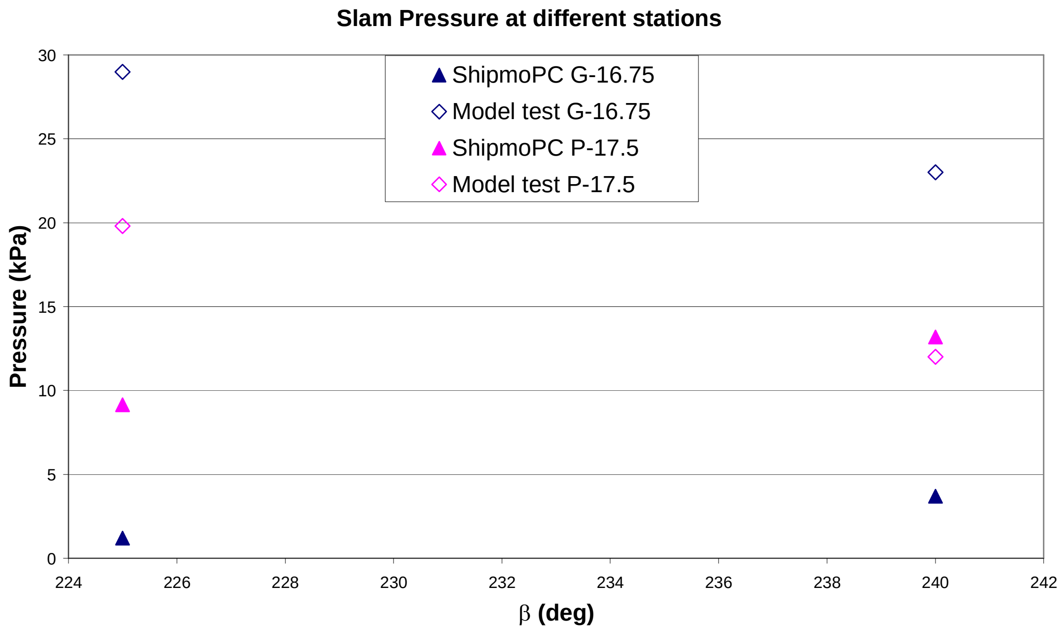

Fig. 4.21: Slamming Pressure.

5. CONCLUSIONS

The purpose of this work was to compare the results of model tests with predictions obtained from different computer programs. The programs used were ShipmoPC, Seaway and I-ship.

In the case of regular waves the motion results obtained from all three programs for both vessels are quite good. It seems that with respect to heave and pitch the programs give similar and acceptable results. The predictions from the programs correlate better with the experiments in the case of pitch than heave.

With respect to the loads prediction the results seem reasonable but some more experimental data would help to find a trend in the predictions. For the CAT ship there all the results are over predicted but with the same trend of the data while for the FIN vessel the predictions seem less reliable.

The results from the programs for vertical accelerations are similar at the centre of gravity while at a forward section there are some differences in the predictions with SEAWAY giving the best results.

When looking at the results from irregular waves the general conclusion is that predictions are not good enough. It should be mentioned that since there are only two values available for comparison it is difficult to reach any firm conclusions.

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Appendix 2: Parametric Numerical Seakeeping Analysis

Safety at Speed - S@S

NUMERICAL TEST RESULTS
WORK PACKAGE 2

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DISCLAIMER

CONTENTS

1. SUMMARY

Part of the work in Work Package 2 for the S@S project was to conduct a study for the comparison of the seakeeping prediction software used by different partners. Based on the results shown on the report number S102.00.13.050.001d it was decided to use two of the programs for the parametric study, these were ShipmoPC and SEAWAY.

The present report shows some of the results obtained from the calculations using ShipmoPC and SEAWAY.

S@S is the acronym for Safety at Speed, a project supported by the European Commission under the Growth Programme of the 5TH Framework Programme. The support is given under the scheme of RTD, Contract No. G3RD-CT-2001-00331.

2. INTRODUCTION

The numerical calculations for the parametric study were done by means of two computer programs and the choice was based on the report no. S102.00.13.050.001d (ref. 2).

ShipmoPC was used for the prediction of motions and bending moments while SEAWAY was used for the prediction of Accelerations.

The prediction results presented here will be used to correlate with the closed-form equations shown in the report no. S102.00.13.060.001 (ref. 3) produced by DTU.

3. NUMERICAL TEST RESULTS

The tests were done for the same ship length but with 5 different B/L ratios and 5 different T/L ratios. The basis vessel for the calculations was the SuperSeaCat3 and the ratios used were ± 20% and ± 10% of original values. Table 1 shows the different breadth and draft values used.

	B or T	-10% B or T	-20% B or T	+10% B or T	+20% B or T
B/L	17.1	15.39	13.68	18.81	20.52
T/L	2.627	2.3643	2.1016	2.8897	3.1524

Table 1: Values of B and T.

In total 25 simulations were carried out. Tables 2 and 3 show the different combinations for breadth and draft and the file names for each run.

B	T				
13.68	2.1016	2.3643	2.627	2.8897	3.1524
15.39	2.1016	2.3643	2.627	2.8897	3.1524
17.1	2.1016	2.3643	2.627	2.8897	3.1524
18.81	2.1016	2.3643	2.627	2.8897	3.1524
20.52	2.1016	2.3643	2.627	2.8897	3.1524

Table 2: Combinations of Breadth and Draft.

B / T	2.1016	2.3643	2.627	2.8897	3.1524
13.68	SSC13-21	SSC13-23	SSC13-26	SSC13-28	SSC13-31
15.39	SSC15-21	SSC15-23	SSC15-26	SSC15-28	SSC15-31
17.1	SSC17-21	SSC17-23	SSC17-26	SSC17-28	SSC17-31
18.81	SSC18-21	SSC18-23	SSC18-26	SSC18-28	SSC18-31
20.52	SSC20-21	SSC20-23	SSC20-26	SSC20-28	SSC20-31

Table 3: File names.

The speeds for each simulation were 5 knots and 40 knots and the wave heading varied from 0^{o} to 180^{o} with increments of 30^{o}.

The mass distribution was based on the distributions of the SSC3 and FIN ships, where FIN ship is one of the vessels used on the SEAWORTH project (ref. 1). The non-dimensional mass distribution (mass for each section/total displacement) is shown in Fig. 1.

The results were calculated for the following points:

- Motions: at Centre of Gravity.
- Bending Moment/Shear Force: at Mid-ship and Forward Perpendicular.
- Accelerations: at Centre of Gravity and Forward Perpendicular.

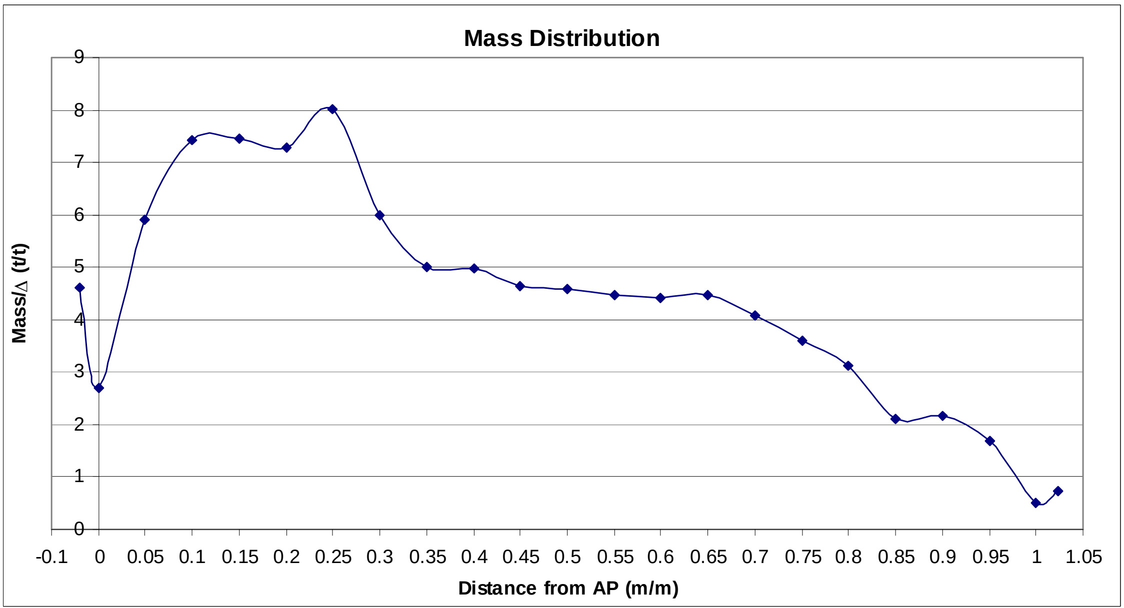

Fig. 1: Non-dimensional mass distribution.

Calculations for following parameters were also done:

- Block Coefficient : $C_B = \dfrac{\nabla}{LB_{WL}T}$
- Midship Section Coefficient : $C_M = \dfrac{A_M}{B_M T}$
- Waterplane Area Coefficient : $C_W = \dfrac{A_W}{LB_{WL}}$
- Prismatic Coefficient : $C_P = \dfrac{\nabla}{A_M L}$
- Displacement : Δ
- Waterplane Area : A_W
- Midship Area : A_M

- Transverse Radius of Inertia : $K_{XX} = 0.4B$

The values of K_{YY} and CoG are the same for all the simulations.

The results shown below are for the test number SSC13-23 (ship with B = 13.68 m and T = 2.3643 m)

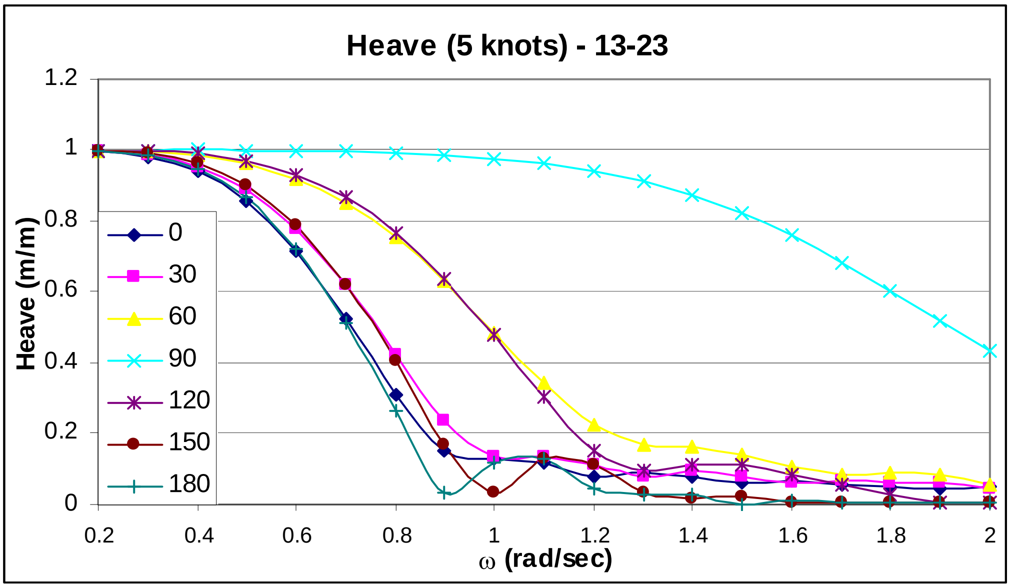

Fig. 2: Heave at 5 knots.

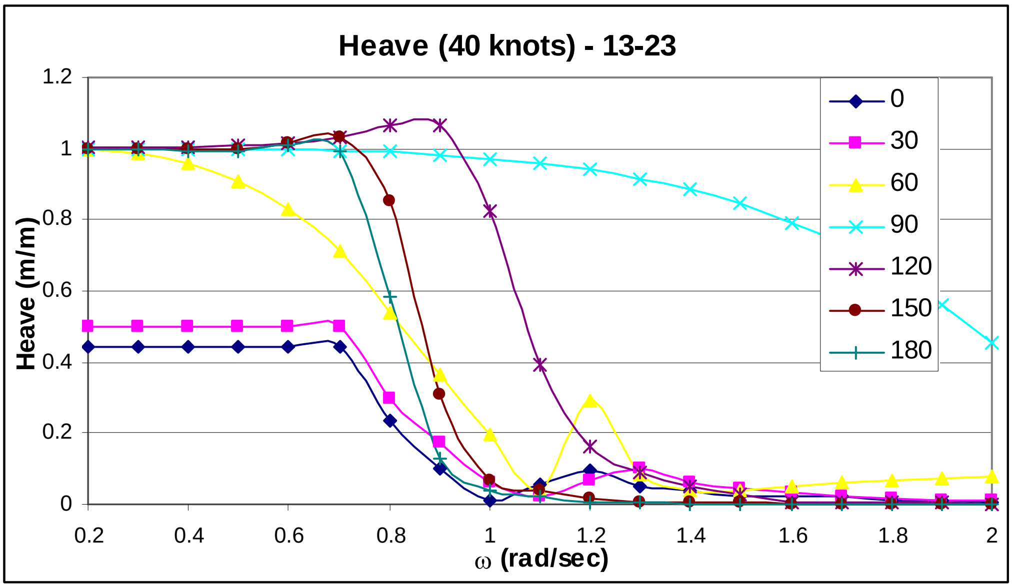

Fig. 3: Heave at 40 knots.

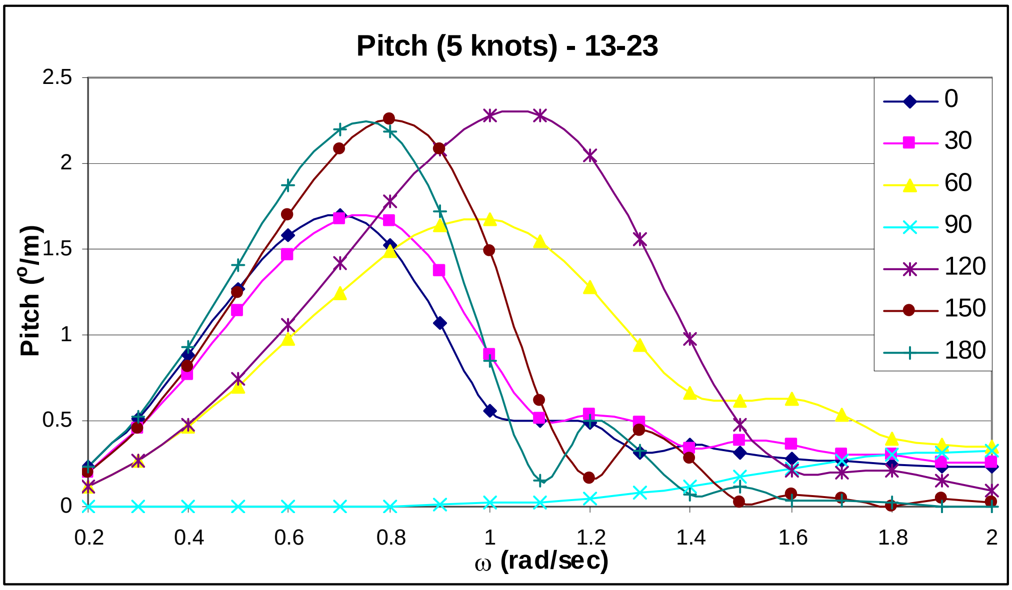

Fig. 4: Pitch at 5 knots.

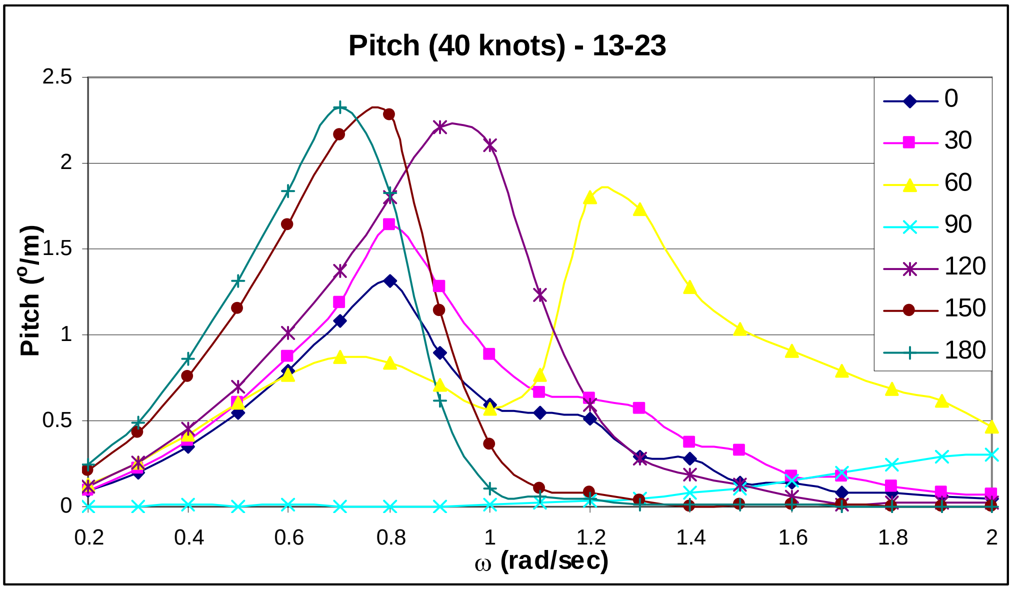

Fig. 5: Pitch at 40 knots.

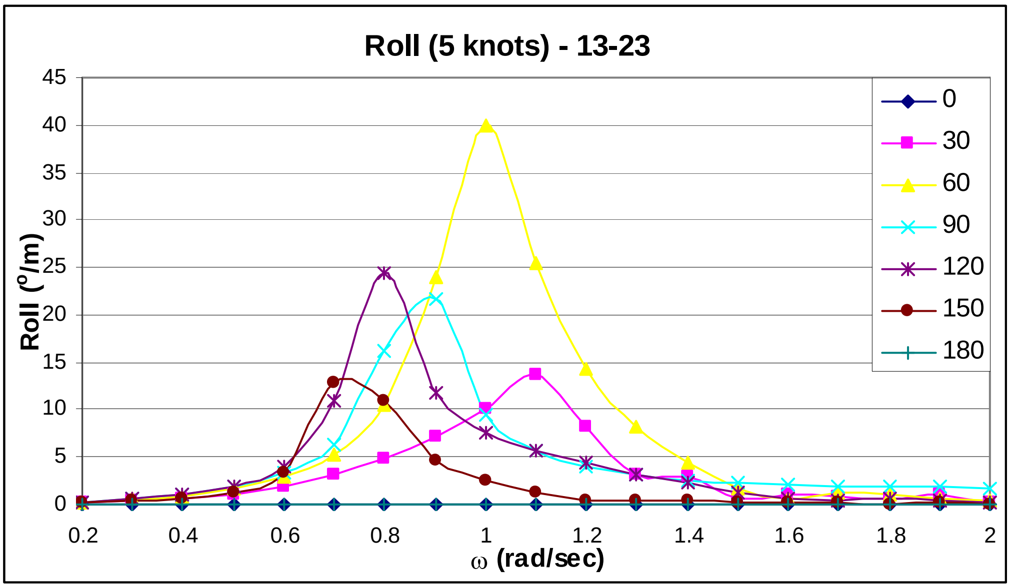

Fig. 6: Roll at 5 knots.

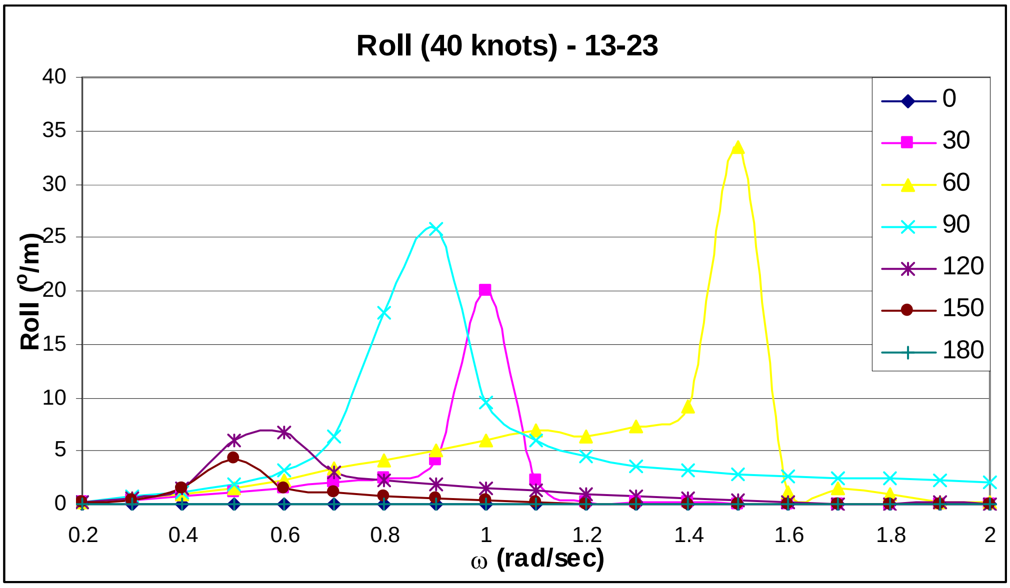

Fig. 7: Roll at 40 knots.

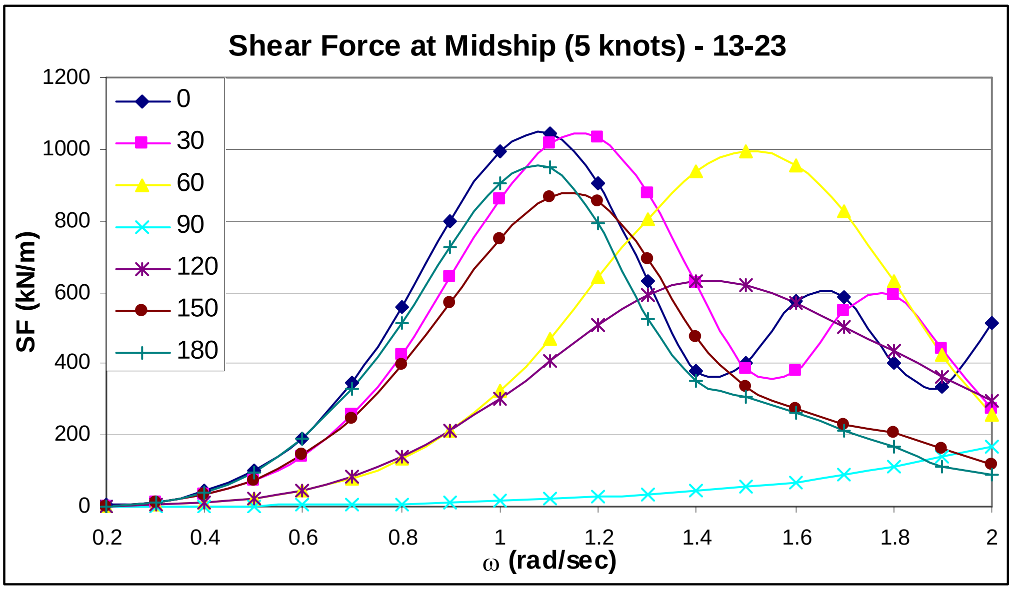

Fig. 8: Shear Force at midship at 5 knots.

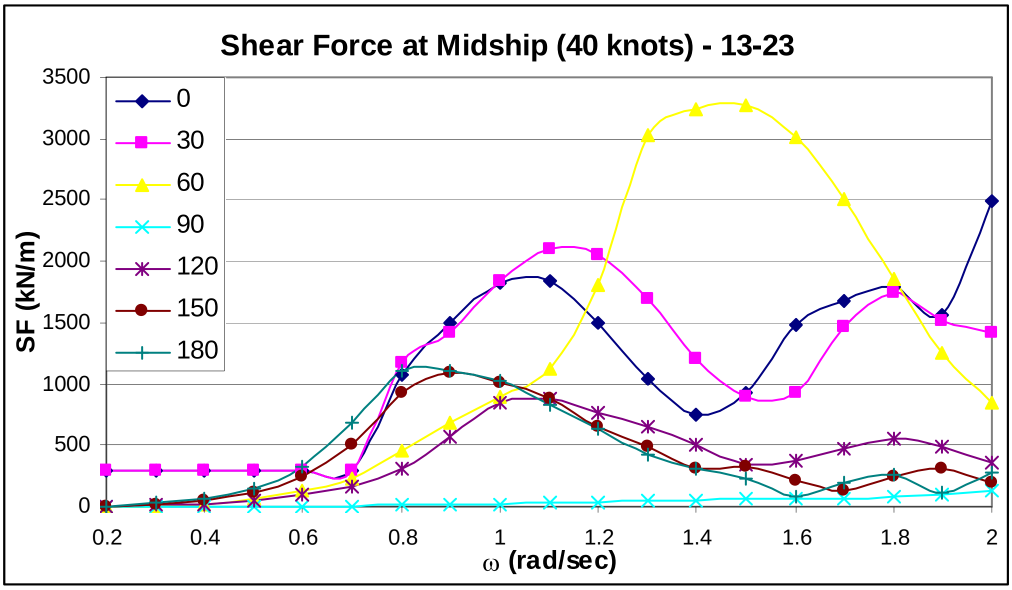

Fig. 9: Shear Force at midship at 40 knots.

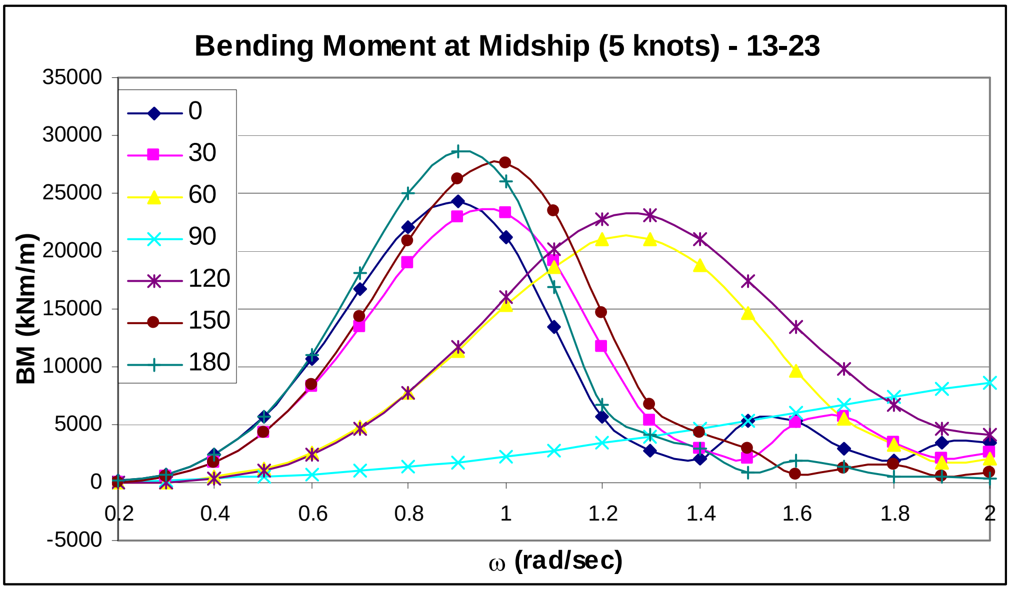

Fig. 10: Bending Moment at midship at 5 knots.

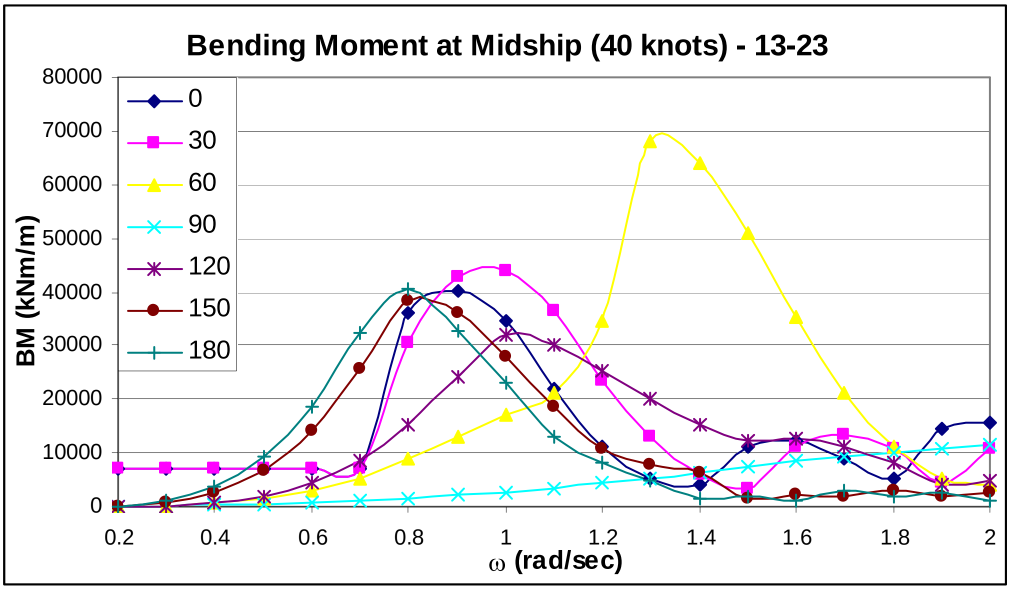

Fig. 11: Bending Moment at midship at 40 knots.

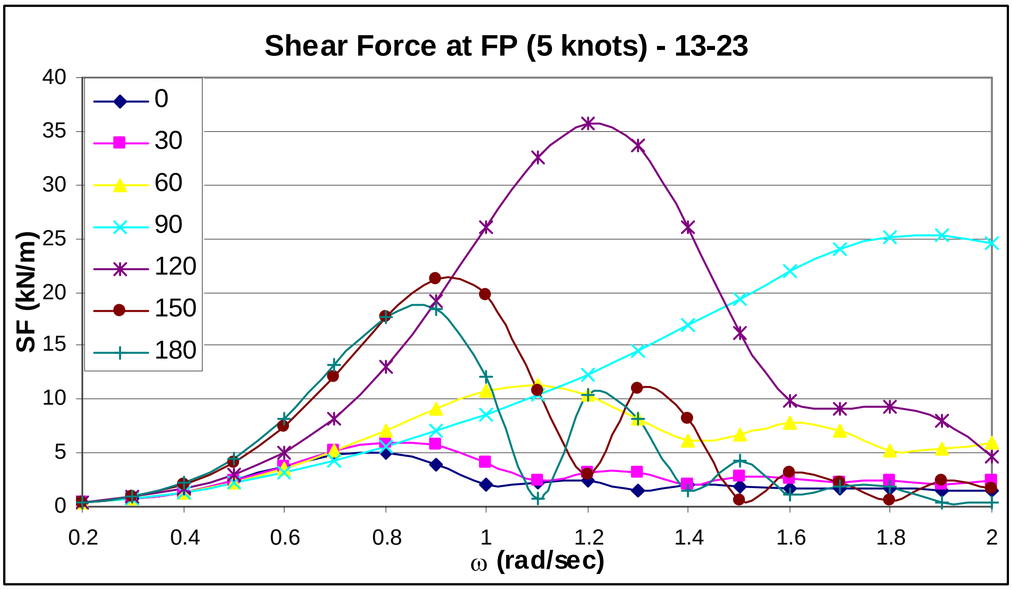

Fig. 12: Shear Force at forward perpendicular at 5 knots.

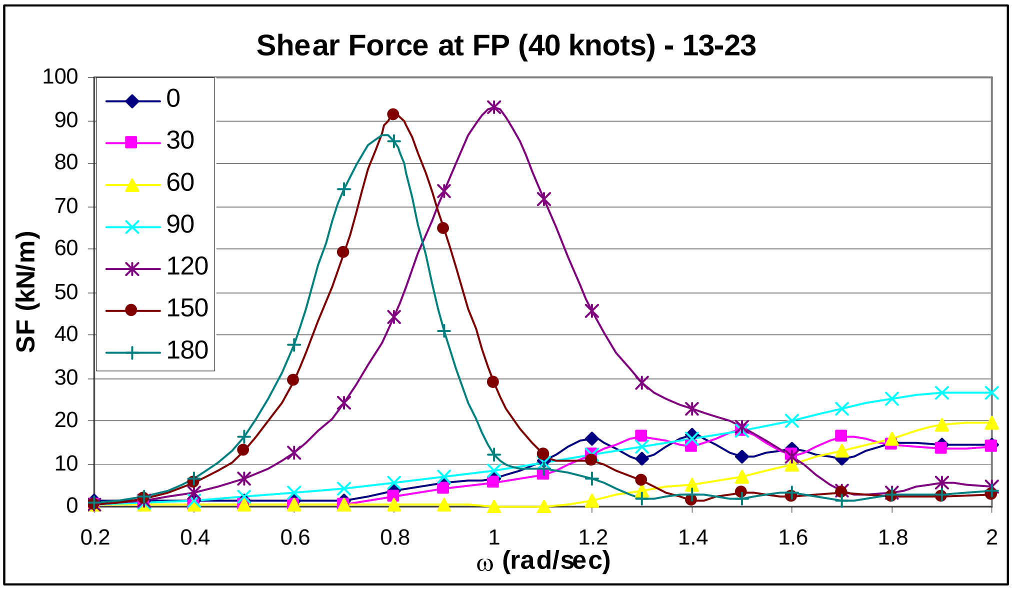

Fig. 13: Shear Force at forward perpendicular at 40 knots.

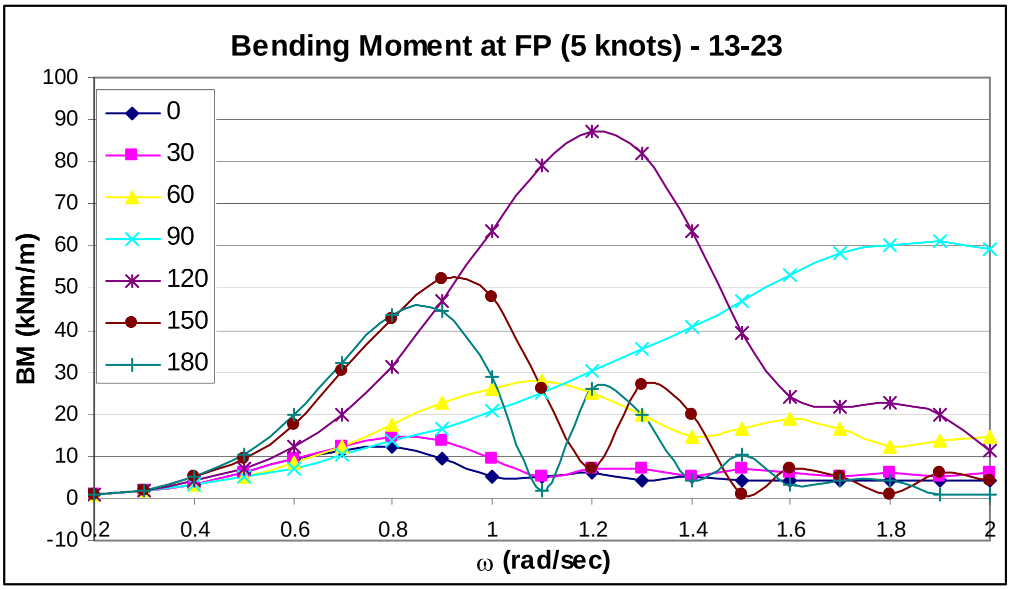

Fig. 14: Bending Moment at forward perpendicular at 5 knots.

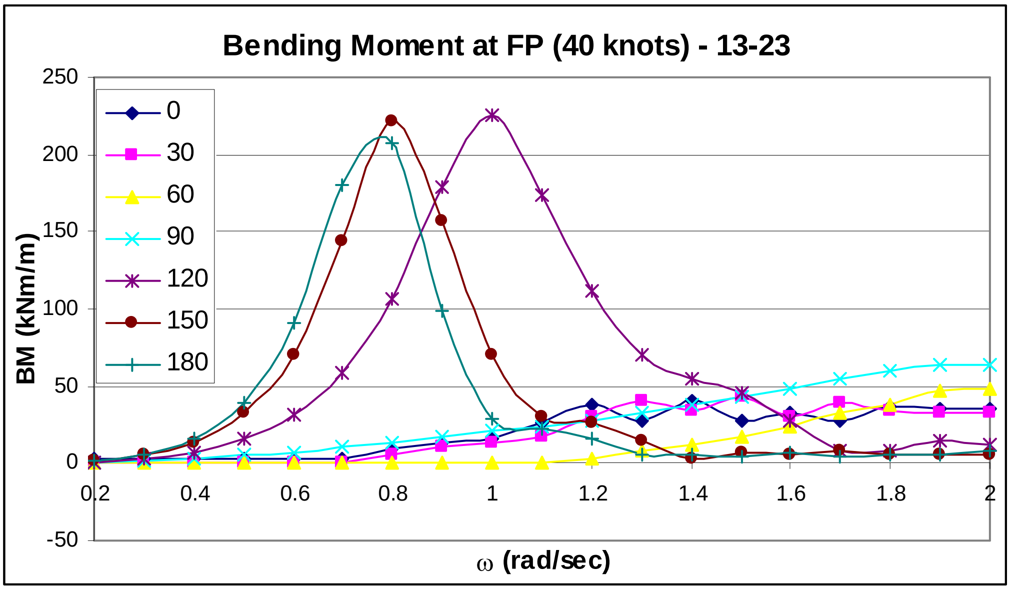

Fig. 15: Bending Moment at forward perpendicular at 40 knots.

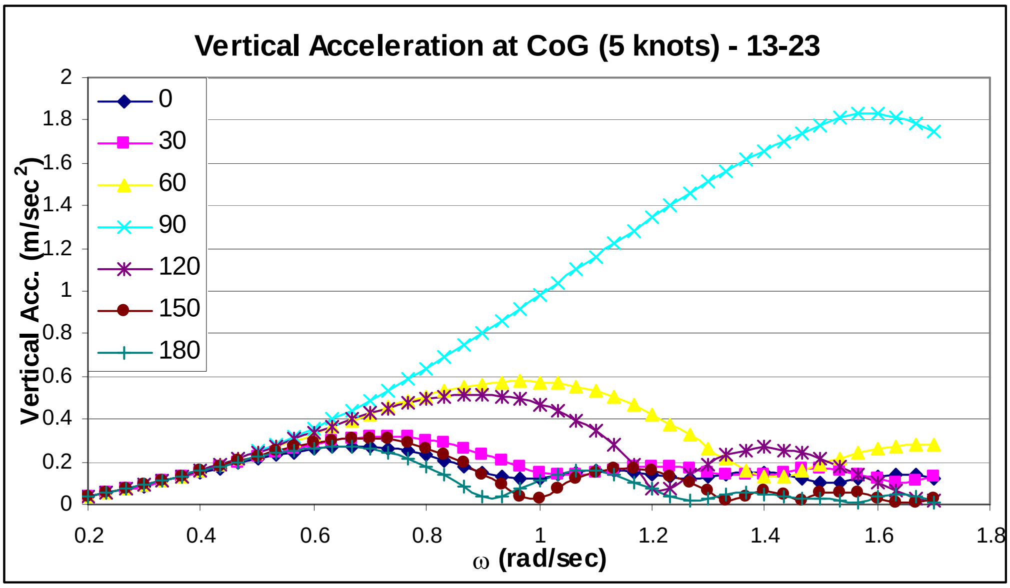

Fig. 16: Vertical Acceleration at CoG at 5 knots.

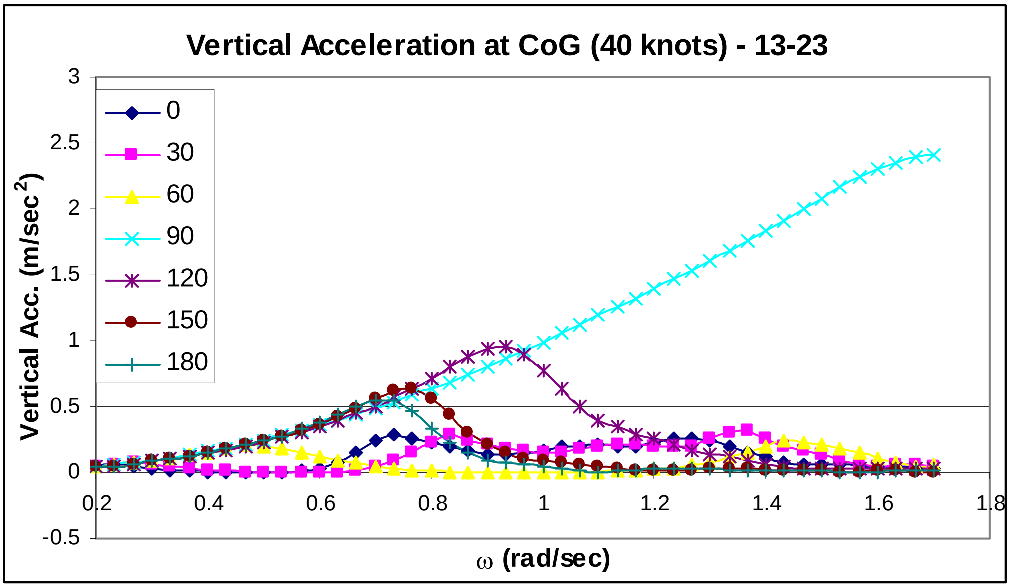

Fig. 17: Vertical Acceleration at CoG at 40 knots.

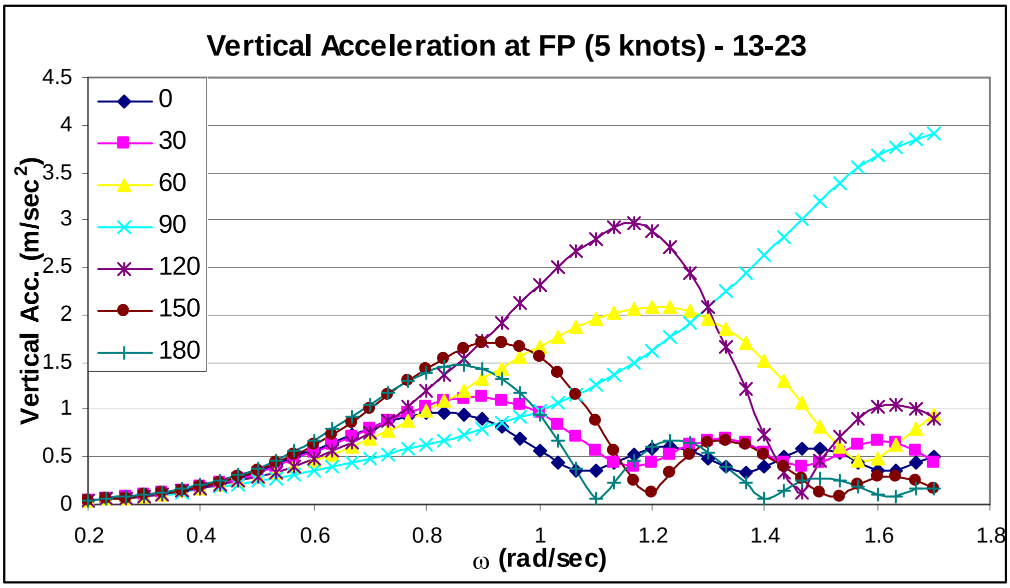

Fig. 18: Vertical Acceleration at forward perpendicular at 5 knots.

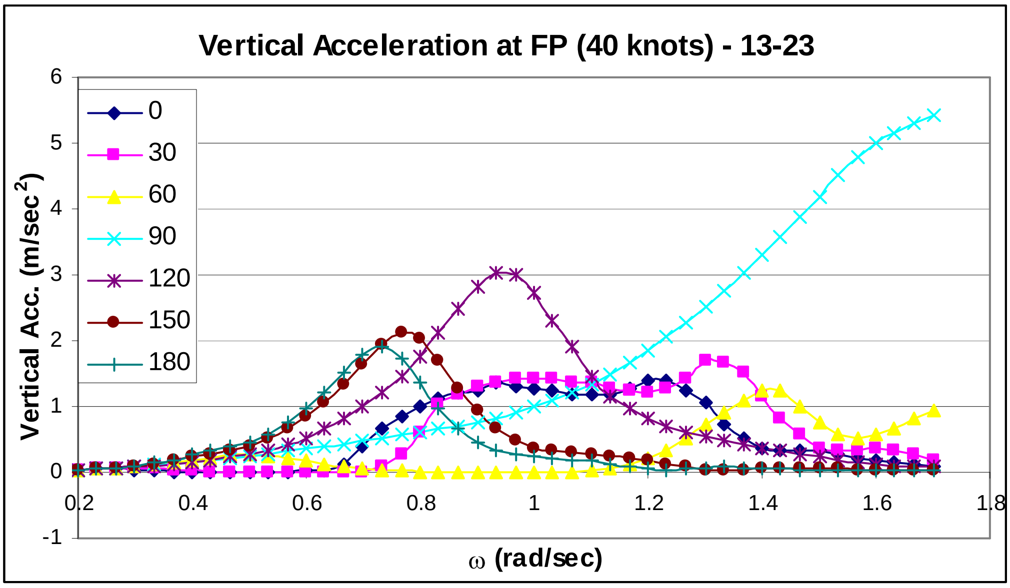

Fig. 19: Vertical Acceleration at forward perpendicular at 40 knots.

4. CONCLUSIONS

In general the results from the simulations are quite good. The best results are for the lower speed but for the high speed they are still acceptable.

Although only the results for one of the cases are shown here all the results for the 25 runs behave in a similar way. The amplitudes and the peak positions changes but the curve follows the same trend.

From the graphs it can be seen that the results for high speed (40 knots) with following waves is not well predicted. This is most particular for wave headings of 0 and 30 degrees and up to a frequency of 0.7 rad/sec.

It is also necessary to note that the roll prediction is for the bare hull, so the values obtained for these calculations are assumed to be much higher than the real values of the ship. This could be improved by introducing the ship's details of stabilizers and appendages.

5. REFERENCES

Corrignan, P., "Seakeeping Test Results on the Stabilised Fast Monohull", Report No. TEC-C036-01, SIREHNA, April 2000.

Esteves, A., Olsen, A. S., "Seakeeping Predictions Using SEAWORTH Data", Report No. S102.00.13.050.001d, UNEW and DTU, S@S, May 2002.

Jensen, J. J., "Seakeeping Predictions Using Closed-Form Equations", Report No. 102.00.13.060.001, DTU, S@S, June02.

Appendix 3 : Model Tests Specifications

S@S

MODEL TESTS SPECIFICATIONS
WORK PACKAGE 2

Document ID Code : S102.20.11.064.002
Date: 2002-09-30
Contract No. G3RD-CT-2001-00331

CONTENTS

1. INTRODUCTION

This report describes the specifications of the seakeeping experimental tests that will be performed by SIREHNA on the SUPERSEACAT3 (SSC3) vessel within Task 2.2 of the S@S project.

2. SEAKEEPING TESTS

2.1 TOWING TANK

The seakeeping tests will be conducted in the towing tank No 3 of the DGA in Paris. The main dimensions of this tank are :

- Length : 220 m,
- Width : 13 m,
- Depth : 4 m.

One end is equipped with an oscillating flap wave maker. The opposite end is equipped with a wave absorber. The tank is equipped with A carriage which allows to tow models with velocities up to 10 m/s.

2.2 MODEL

2.2.1 Model scale and main particulars

Tests will be performed on a model at scale 1:20 of the SUPERSEACAT3 (SSC3) vessel that will be manufactured from wood material. The ship displacement will be fixed to its Full Load. Departure value with deadweight of 312 tons. The full scale and the corresponding model ship characteristics are :

Description	Units	Abbr.	Ship value	Model scale (1:20) value (fresh water)
Length overall	m	Loa	100.3	5.015
Length between perpendiculars	m	LPP	88.0	4.400
Breadth moulded	m	B	17.1	0.855
Breadth waterline	m	BWL	14.2	0.710
Depth upper deck	m	D	10.7	0.535
Design draft	m	T	2.6	0.132
Displacement	Ton, kg	Δ	1239.4	151
Installed power	MW	PINST		
Maximum speed	kts	VMAX	40	8.94
Service speed	kts	VSERV	38	8.50
Height centre of gravity above keel	m	KG	6.5	0.326
Longitudinal position CoG forward of Station 0	m	LcG	33.5	1.676
Pitch radius of gyration	m	kYY	24.2	1.208
Pitch inertia	Ton.m²,kg.m²	IYY	722846	220

In order to consider a variation of the bow flare coefficient, tests will be performed with the nominal shape (C_{BF}=0.18) and with a modified shape (C_{BF}=0.23). The two shapes are schemed on figure 1. The bow will be modified for X≥73.5 m ahead of AP, and only above the waterline in order not to require a model loading adjustment when passing from one shape to the other.

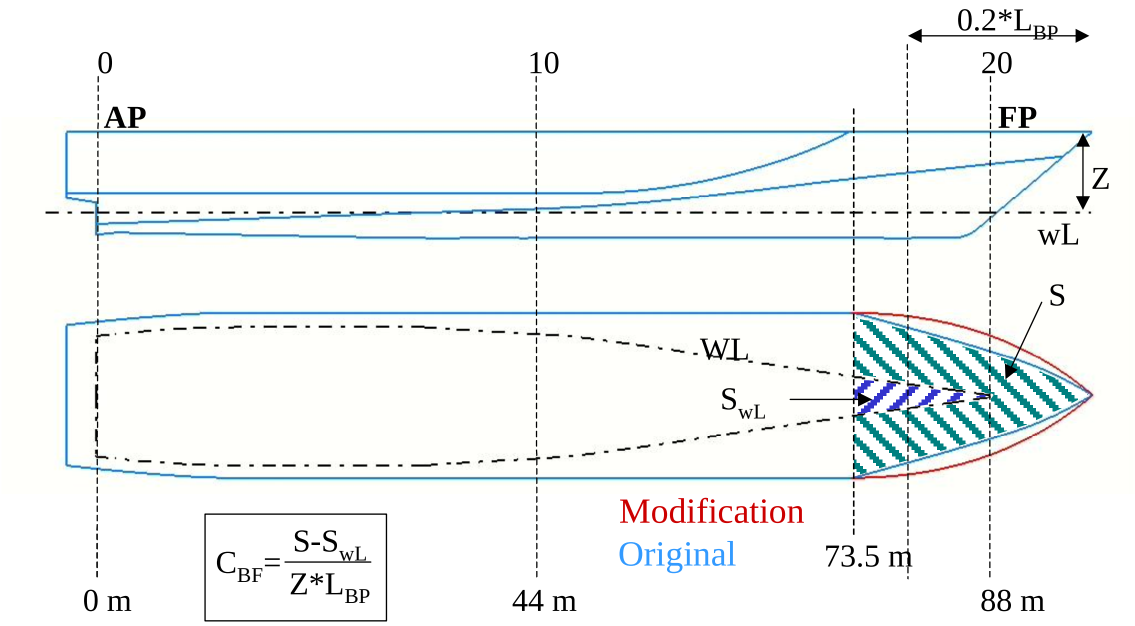

Figure 1 – Original and modified bow shapes

2.2.2 Modelling of ship elasticity

The SSC3 model will be composed of two rigid segments linked by an elastic beam. The separation between segments will be located at 0.5 Lpp from the aft perpendicular. They will be made watertight by means of rubber strips (thickness 0.2 mm) overlapping the separation.

The mass, the longitudinal position of the centre of gravity and the longitudinal radius of gyration for each segment and for the whole ship will be adjusted to values derived from the SSC3 mass distribution. The heights of the centres of gravity for each segment will be equal to the one for the whole ship.

The reference frame will be : X – axis horizontal from stern to bow ; Z – axis vertical and upward ; Y – axis to complete the direct frame. (The pitch angle is positive when the bow is down ; the relative wave elevation increases when the free surface goes up on the bow).

The pitch radius of inertia of both segments will be checked in air by means of oscillation. The loading characteristics to obtain on both segments are as follows :

Segment	Stations	Description	Units	Abbr.	Ship value	Model (1:20) value
1	stern- 10	Longitudinal position CoG forward of St0	m	LcG1	18.9	0.947
		Height centre of gravity above keel	m	HcG1	6.5	0.326
		Mass	Ton,kg	M1	829.2	101
		Pitch radius of gyration w.r.t segment CoG	m	kYY1	12.4	0.622
		Pitch inertia	Ton.m²,kg.m²	IYY1	128322	39.1
2	10-bow	Longitudinal position CoG forward of St0	m	LcG2	63.0	3.148
		Height centre of gravity above keel	m	HcG2	6.5	0.326
		Mass	Ton,kg	M2	410.2	50
		Pitch radius of gyration w.r.t segment CoG	m	kYY2	12.4	0.620
		Pitch inertia	Ton.m²,kg.m²	IYY2	63154	19.2

The control of trim and draft of the complete model will be realised in still water.

The dimensions of the elastic beam will de selected in order to obtain a model longitudinal bending natural frequency in accordance with the full scale one. The SSC3 estimated bending frequency is about 2 Hz, which corresponds to 8.9 Hz at model scale. The model longitudinal bending frequency will be checked in water.

2.2.3 Model towing

The model will be towed with free pitch and heave motions.

The interface between the model and the towing carriage will be composed of two vertical columns, located in the model vertical plane of symmetry, guided in frames linked to the carriage (vertical translations only for the aft column). The columns lower ends are linked to the model.

Both columns will be fixed on the model, in the aft segment, right above the fixations of the beam to the segment.

2.3 INSTRUMENTATION

The following measurements will be performed :

- Impulsive pressures,
- Bending moments,
- Ship motions and accelerations,
- Encountering waves.

In addition tests will be video recorded.

2.3.1 Impulsive pressures

Impulsive slamming pressures will be measured at seven locations in the bow (on port side). The pressure transducers membranes will be flush mounted in the model hull. Piezo resistive pressure transducer will be used. Their characteristics are :

- Range : 1.7 bars
- Sensitivity ~ 5.8 V / bar
- Resonance frequency (with amplifiers) : 400 kHz
- Acceleration sensitivity : 0.34 mbar / g
- Membrane diameter : 8.1 mm

The locations of highest probability of slamming occurrence have been determined by means of seakeeping calculations performed with head seas. The procedure consists in calculating the relative velocity normal to the hull at control points distributed along the hull and by comparing it with the threshold velocity determined by the Ochi criterion. The most critical zones are then defined as the control points for which the threshold velocity is exceeded for the lowest wave amplitudes. The results of this analysis is presented on figure 2.

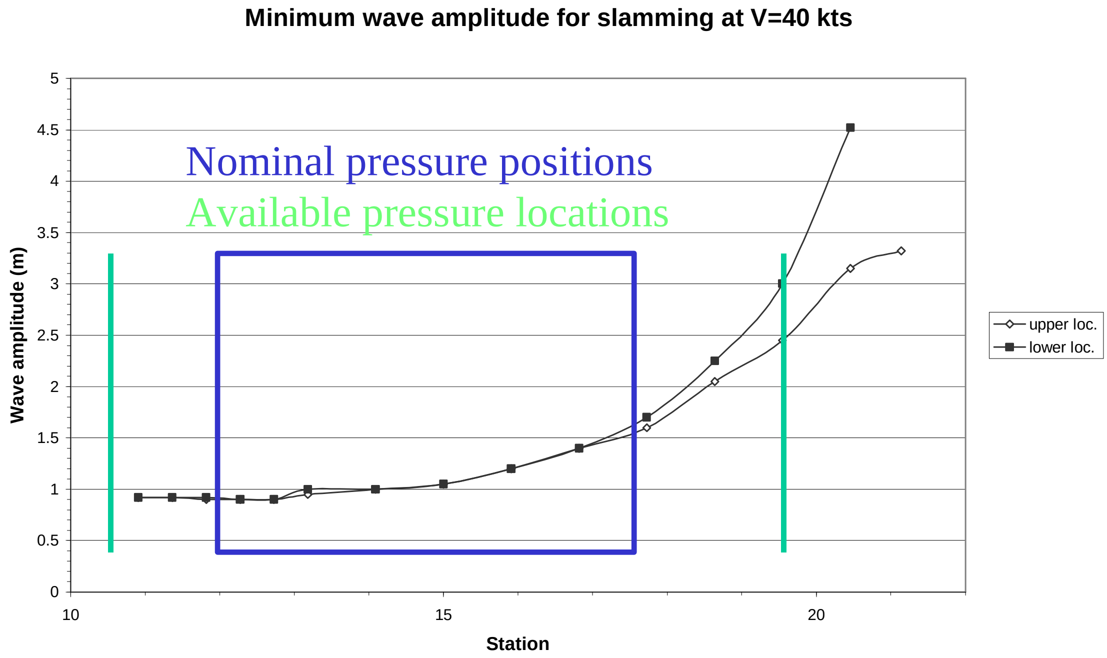

Figure 2 – Determination of slamming areas

According to these results, eleven pressure locations will be available on the fore segment. Their co-ordinates are presented in the table below.

			full scale			model scale		
location	transducers	station	X/C0 m	Z/LWL m	Z/0H m	X/C0 m	Z/LWL m	Z/0H m
I1		19.55	86	3.69	6.32	4.3	0.1845	0.316
I2		18.6	82	3.5	6.13	4.1	0.175	0.3065
I3		17.7	78	3.13	5.76	3.9	0.1565	0.288
I4	P2	16.8	74	2.89	5.52	3.7	0.1445	0.276
I5	P3	15.5	68.2	2.28	4.91	3.41	0.114	0.2455
I6	P4	14.55	64	1.9	4.53	3.2	0.095	0.2265
I7	P5	13.5	59.4	1.5	4.13	2.97	0.075	0.2065
I8	P6	12.5	55	1.02	3.65	2.75	0.051	0.1825
I9	P1	12.5	55	-1.31	1.3	2.75	-0.0655	0.065
I10	P7	11.5	50.6	0.66	3.29	2.53	0.033	0.1645
I11		10.5	46.2	0.26	2.89	2.31	0.013	0.1445

Ten positions are located below the chine. Following WP3 request, one additional is located half way between the keel and the waterline, below position I8.

The seven sensors will be distributed over the eleven available locations. The nominal distribution is represented in green in the chart above, and in blue on figure 3. The position of the sensors will be adapted if necessary during the tests according to the measurements in order to cover as best as possible the area of highest pressures.

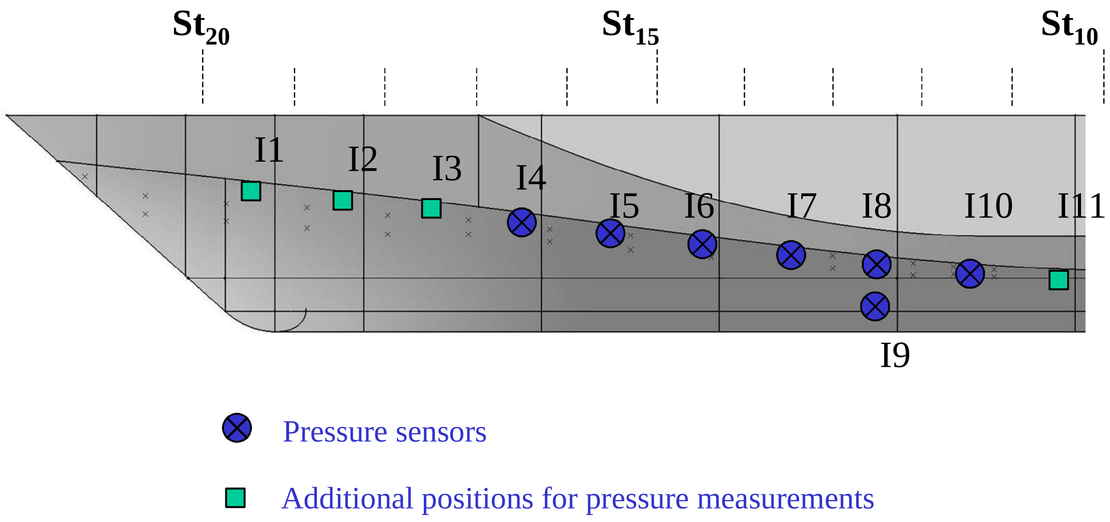

Figure 3 – Positions of pressure sensors

2.3.2 Bending moments

The bending moments will be measured at station 10 , between the for and aft segments by means of two strain gauges glued on the upper and lower sides of the elastic beam.

2.3.3 Ship motions and accelerations

Pitch and heave will be measured.

The vertical acceleration will also be measured at two locations : at the centre of gravity of the model ship, and on the deck at the fore perpendicular.

2.3.4 Wave measurement

The encountering waves will be measured at the bow of the model by a controlled pin wave gauge.

2.3.5 Speed measurement

The towing carriage speed will be measured during tests.

2.3.6 Visualisations

The tests will be video recorded with a field of view focused on the bow part.

2.4 DATA ACQUISITION

Pressure signals will be recorded with a 1 kHz sampling rate. Other signals will be sampled at 150 Hz. Both will be synchronised in time.

2.5 TESTS CONDITIONS

Tests will performed on head waves with ship speed of 40 knots. Both regular and irregular waves (defined by Jonswap spectra) will be tested on both bow shapes. In a first step, the motion, acceleration and bending moment RAOs will be determined by splash tests. The regular wave frequencies will then be selected around the RAOs peaks.

The nominal test program is presented below. It will be possibly adapted according to the results obtained.

V = 40 kn	Splash tests (RAOs)	Regular Wave amp. (m)	Head Wave Period (s)	Waves Number of periods	Irregular * Tpic (s)	Head Waves HS (m)
Original bow BFC = 0.18	1 splash	3	7 to 11	5	9	4.5
		2	7 to 11	5	9	2
Modified bow BFC = 0.23	1 splash	3 or 2	7 to 11	5	9	2 or 4.5 m

* JONSWAP spectrum

Appendix 4 : Sea Trials Specifications

S@S

SEA TRIALS SPECIFICATIONS
WORK PACKAGE 2

Document ID Code : S102.20.11.064.001
Date: 2002-09-30
Contract No. G3RD-CT-2001-00331

CLASSIFICATION AND APPROVAL

Classification: Confidential

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001	2002-09-30	DM		First issue of the complete document

DISCLAIMER

CONTENTS

1. INTRODUCTION

This document presents the specifications for the experimental programme for full scale measurements on a High Speed Craft (HSC) within the Work Package 2 of Safety@Speed project.

These experiments aim to perform full scale measurements on a large high speed mono-hull to provide reference data on ship motions, accelerations, vibrations, noise and indoor climate, with sea-state and vessel speed within Task 2.2 " Formulation of models". These data will be used for the formulation and the verification of models developed also in Task 2.2.

Sea trials will be performed on the Sea Container (SEA) SuperSeaCat 3 in exploitation between Liverpool and Dublin.

Measurements will include ship motions, accelerations at various locations over the ship, wave heights and frequencies, vessel speed and running trim and draft, vibrations, noise and indoor climate (temperature, humidity).

S@S partners involved in sea trials are : SIREHNA (co-ordinator of trials), SEA, NTUA and VTT.

2. HIGH SPEED CRAFT PRESENTATION

SEA will provide a ship for sea trials. The selected one is a large high speed mono-hull named SuperSeaCat3.

The main characteristics of the SuperSeaCat3 are given in appendix 4.1. The figure 1 here after shows the ship :

Figure 1 : The SuperSeaCat 3

The operating speed of the SuperSeaCat 3 is 38 knots, and the maximum speed is 40 knots.

The ship is currently operating a shuttle service between Liverpool and Dublin with some additional trips to Isle of Man (see figure 2 hereafter) :

Figure 2 : The SuperSeaCat 3 routes

The SuperSeaCat 3 is available until the end of October 2002 for sea trials (Last trip on Monday 4 Th November).

3. MEASUREMENTS

3.1 Ship motions

The motions of the ship will be measured at two locations :

- SIREHNA will measure the motions at midship by means of an inertial/GPS motion measurement unit, named POS/MV produced by APPLANIX (Canada) and owned by SIREHNA.
  Main motions (Heave, Pitch, Roll and Yaw) will be provided. The heave motion will be calculated for one reference point of the ship (generally, the CG of the vessel).
- VTT will measure the motions of the ship at the bow with the MRU-6 unit by Seatex, Norway.
  Measured signals will be : roll, pitch, yaw, heave, sway and surge. Some of these signals will be used when determining the comfort criteria.

3.2 Accelerations

Accelerations will be measured at various locations over the ship :

- Close to the bow, by VTT: Accelerations are measured in three directions using three identical sensors. Sensors are made by Kistler, type K 8310A10, range +- 10 g, 0...180 Hz. They are attached to aluminium cube placed inside

a protective casing. The measured data will be processed using program Famos by Imc. The acceleration signals are frequency weighted according to ISO 2631. Rsm-, VDV-values etc. of relevant signals will be given for each measuring period or duration of constant environmental condition.
- At midship, by SIREHNA with the IMU (Inertial Motion Unit)of the POS/MV.
- Close to stern, by NTUA : The integrated system, Type 01dB, described in the next section for the noise measurements will also be used for measuring and evaluating the vibration parameters (Time history, Spectrum analysis, Statistical analysis etc).
  The measurements and the evaluation of the Vibration Parameters (I,v, γ) will be done with a complete spectrum analysis in octave bands (2 to 16000 Hz) or in 1/3 octave bands (1 to 200000 Hz).

The data analysis of front and midship acceleration measurements will focus on slamming and whipping/springing phenomena, while rear acceleration measurements will be processed to assess main engine and propulsion vibration phenomena.

3.3 Sea state

The sea state will be measured and estimated in three different ways :

- The operator gets wave statistics for each day which have been checked and found to be reliable. Theses data will be provided by SEA for each day of tests.
- The wave elevation will be measured at the bow of the ship by means of a wave sensor (radar technology) fixed on the deck and looking just in front of the bow. Measurements will be carried out by SIREHNA. Wave elevation data will be recorded in real time with ship motions data on the same acquisition system. Wave elevation measurements will be processed and analysed on board the ship to assess the significant wave heights H1/3 during the trials.
- Ship motion measurements will be processed by the Sea State Estimator developed by SIREHNA to provide the wave spectrum and the main wave heading.

Onboard estimations of significant wave height and of the wave direction with respect to the ship will be also noted down on the test sheet.

3.4 Noise Vibration and indoor climate

Noise vibration and indoor climate measurements will be performed at the stern by NTUA.

3.4.1 Noise measurements

3.4.1.1 Measurement standards

The measurements will be performed according to ISO 2923, 1996 'Measurement of Noise on Board Vessels'
The IMO Res.A468(XII)-1981 "Code of Noise Levels on Board Ships" will also be considered.

3.4.1.2 Measuring equipment

3.4.1.2.1 Sound Level Meters

- Integrating precision sound level meter Bruel & Kjaer 2221, 2231 with the octave-1/3 octave band filter 1625.

- Precision sound level meter Bruel & Kjaer 2203 with octave band filter Bruel & Kjaer 1613

3.4.1.2.2 Integrated Sound and Vibration Measuring System

- Complete system Type 01dB

The integrated system is used for measuring and evaluating the sound and vibration parameters (Time history, Spectrum analysis, Statistical analysis etc). It includes portable PC, Analog to Digital converters, microphones, accelerometers, calibrators, air screens, etc.

- Complete system Type 01dB, Measuring and Evaluating Building Acoustics Parameters (R_w, L_{nw} etc).

3.4.1.2.3 Measuring Parameters:

The noise will be measured in dB(A), in octave or in 1/3 octave bands. The overall level, as well as the ISO Noise Rating Number will be directly measured and evaluated with special computer programs. The central frequencies of the frequency bands will be normally from 31.5 to 8000 Hz, but, if needed, it can be extended to the range from 25 to 20000 Hz.

3.4.2 VIBRATION MEASURING INSTRUMENTS

3.4.2.1 Measuring Procedure

The recommendations described in "Code for shipboard vibration measurements", SNAME, 1975, will be followed when possible.

It is noted that the conditions of the measurements at hand do not correspond to the environmental conditions for the standard ship vibration measurements. The later inquires calm weather conditions.

3.4.2.2 Vibration Accelerometer sensor

Type : Bruel & Kjaer 4370 piezoelectric accelerometer

3.4.2.3 Vibration Amplifier/ Conditioner

Type : Bruel & Kjaer 2635 portable charge amplifier

3.4.2.4 Vibration calibrator

Type : Bruel & Kjaer 4291 portable vibration calibrator

3.4.2.5 Acquisition - Recording System

Portable PC Notebook, equipped with Analog to Digital converter and acquisition software.

3.4.2.6 Time duration and acquisition range

The time duration of each record will be 1 min.
The acquisition range will be 10 kHz.

3.4.2.7 Processing and delivered data

The rms values of the vibration velocity will be evaluated.
The spectral analysis of each record will be obtained and presented.

3.4.3 INDOOR CLIMATE

Indoor climate will be performed by means of Temperature and Humidity measuring sensors.

3.5 Operational parameters

During sea trials, the following operational parameters will be written down on a monitoring sheet of the tests :

- Running trim
- Running draft

These parameters will be checked at each departure and arrival.

All other important events which could influence the behaviour of the vessel will be noted down.

Furthermore, the route and the speed (relative to the ground) of the ship will be measured and recorded by the POS/MV system.

3.6 Data acquisition and synchronisation

Motions measurements will be performed at a rate of 20 Hz during 20 minutes with constant wave heading angle and ship speed. When one of these two parameters will change a new measurement will be started.

For noise and vibration measurements the sampling rate will be considerably higher (about 20 kHz) and the sampling time on the order of 1 min.

The acquisition rate for the accelerations at the bow will be close to 1 kHz.

All the data will be synchronised by means of the GPS time. The ship motion measurement system of SIREHNA receives the GPS time and all its measurements are dated in this time frame.

SIREHNA will provide an analog signal to the two other acquisition systems installed on board. This signal will be based on the 1PPS (One Pulse Per Second) signal delivered by the POSMV, which provides pulses with falling edges synchronised with the GPS seconds. To date pulses with respect to GPS time, one TTL pulse (with a higher level) is superimposed to the periodic TTL signal. This pulse is recorded with POSMV data as an event dated with respect to the GPS time. This signal can be, also, used to trig other data acquisition systems when higher frequency and shorter duration are required (noise and vibration).

4. TEST PROGRAM

Sea trials will be performed during operational link between Liverpool and Dublin with some additional trips to Isle of Man (see figure 2).

The figure 3 and table 2 in appendix 4.2 present the route of SSC3. The duration of the crossing to Dublin is about 4 Hrs, with a depart from Liverpool at 08h00 am and an arrival to Dublin at 12h00; and a depart from Dublin at 13h15 with an arrival to Liverpool at 17h00.

Different heading will be tested. The SuperSeaCat 3 will be on normal passenger service and therefore there will only be limited scope to alter course and speed on the crossing. All manoeuvres will be carried out at the discretion of the master.

All the equipment will be installed on board during stand-by periods when the ship will stay at dock in LIVERPOOL for a couple of hours.

The test campaign will be split in three periods of 2 or 3 days in September and October 2002 in order to obtain different sea states.

5. TIME SCHEDULE

The tentative time schedule is as follows :

Date	Achievements	Observations
June 2002	Specifications preparation	Working document
July 2002	Refinement of sea trials specs.	
End of July 2002	Final sea trials specifications	Deliverable item
August 2002	Sea trials preparation	
August 2002	Questionnaires NTUA & VTT	To be checked
September 2002	First slot of tests	3 days
September 2002	Data processing	
End of September 2002	Second slot of tests	3 days
October 2002	Last slot of tests	2 days
October 2002	Data processing	
End of October2002	Results of sea trials	Deliverable item

Table 1

Appendix 4.1

Super SeaCat 3 Main characteristics

DATE 5/2/96	HF REF 17
MARKING	SUPERVISOR
MASTER FILE	YARD

HF351 Fincantieri

A	R	N
Approved	Rejected	Noted

Initials [illegible] Date 13/2/96
See comments in fax no: 7351-115

MAIN DIMENSIONS

Length over all	LOA	100.30	m
Length between perpendicular	LBP	88.00	m
Breadth moulded	B	17.10	m
Depth passenger deck	D	10.70	m
Depth Main car deck	D	6.00/4.60	m
Depth Upper car deck	D	7.10	m

FOR APPROVAL
DONE.

NUM.	DATA	RIF.	DESCRIZIONE	ESEG. DA	CONTR. DA	C.O.

MODIFICHE

FINCANTIERI
Cantieri Navali Italiani S.p.A.
DIVISIONE COSTRUZIONI MILITARI

MDV 1200 PEGASUS A.A. FAST FERRY

RILEVATO SCAFO

HULLFORM OUTPUT

LUOGO E DATA: GENOA, 19 GEN. 96	FOGLI ALLEGATI N. 67	ESEGUITO DA: [signature]	COMMESSA	0 1 5 9 9 9
NOTE: VALID ALSO FOR HULL nr. 016000	SCALA /	CONTROLLATO DA: [signature]	REPERTORIO	8 3 5 0 0 0
	C.O. [signature]	APPROVATO DA: [signature]	N. DIS	G Z 8 3 5 0 0 1 1 M
			MODIFICA	1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16

Appendix 4.2

Super SeaCat 3 Route

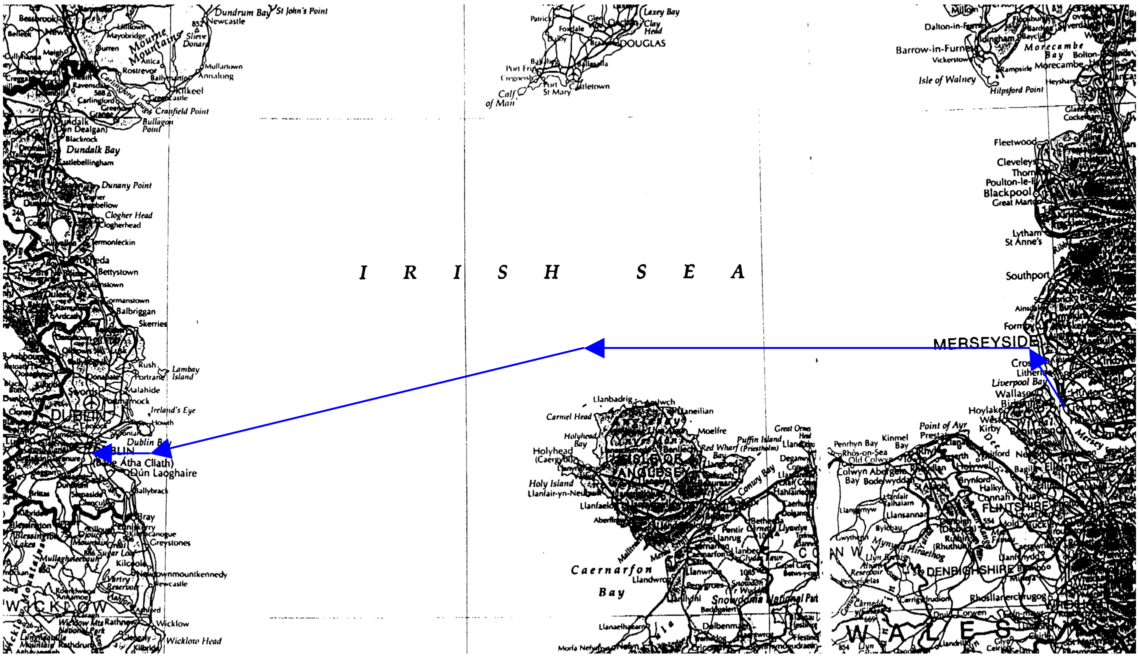

Figure 3

Rough Plot Of SuperSeaCat 3 route From Liverpool to Dublin

Location	Latitude	Longitude	Course	Distance
Liverpool Bar	N 53 31	E 03.21		
			T/C 270°	DMLS 8.3
Douglas Oil Field	N 53 31.2	E 03 34.5		
			T/C 274°	DMLS 34.2
Anglesey Sep. Scheme Point 1	N 53 33.2	E 04 32		
			T/C 262°	DMLS 6.7
Anglesey Sep. Scheme Point 2	N 53 32.2	E 04 42.7		
			T/C 257°	DMLS 48.5
Point C Dublin	N 53 21	E 06 01.3		
			TOTAL DMLS	103.7

Table 2 Way Points between LIVERPOOL and DUBLIN