NUMERICAL MODELLING OF DAMAGE SHIP STABILTIY IN WAVES

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SUMMARY

This paper outlines recent advancements, achieved to date at The Ship Stability Research Centre (SSCR), in modelling of damage ship stability in waves by means of numerical simulations. Some details of the mathematical model are presented with the emphasis put on the water sloshing representation. Fundamental validation studies demonstrate that simplified methods for estimation of fluid motion and resultant loads can successfully be applied for examination of flooded ship behaviour. However, discrepancies in predictions of basic dynamics of a damaged ship with water ingress/egress are identified. Some deficiencies in current understanding of hydrodynamics of a breached hull are highlighted.

NOMENCLATURE

I'_s	Inertia matrix of ship ("s") w.r.t. G_s
I'_w	Inertia matrix of water ("w") w.r.t. G_s
$\vec{v}'_{Gs}, \vec{w}'$	Ship rectilinear and angular velocities
M_w	Mass of floodwater in a single compartment
$\vec{r}'_w$	Position vector of the centre of buoyancy of floodwater "w" in a body-fixed reference system with origin at Gs
$\vec{v}'_w$	Velocity vector of the above point
$\vec{M}'_{Gs}$	Resultant of all external moments acting on ship (three-component vector)
$\vec{g}'$	Gravity acceleration vector
$\frac{d}{dt}$	Local time derivative
w_n	Natural frequency of water sloshing
b	Breadth of the tank
h	Fluid level

Superscript denotes that vectors are resolved in ship bound rotating system of reference.

1. INTRODUCTION

The subject of dynamic ship stability in waves with breach in the hull has achieved in recent years much needed attention, not least because of the latest tragic maritime accidents involving significant casualties, but also in view of the growing industrial interest in ships with capacity reaching 10 000 and more passengers onboard, where it is only natural that safety is of prime importance in the whole lifecycle of such vessels.

Assessment of ship performance in terms of her survivability, however, is not straightforward an undertaking, as in addition to complexity of predicting ship behaviour in waves, further intricacies arise in accounting for progressive flooding through the vessels internal layout and ensuing ship-floodwater interactions.

Such dynamic effects of fluid motion on the ship responses, and vice-versa, have been extensively studied in the past from the viewpoint of roll stabilising tanks, oil tankers, water trapped on deck, LNG carriers, and others where the amount of fluid mass in the tank is constant. The problem of a ship undergoing progressive flooding entails further degrees of non-linearity arising from fluid mass variation, which also renders the simulated process non-stationary.

The general difficulties in dealing with the problem accurately derive in great part from the sloshing phenomenon, which mode, influenced by tank geometry, dimensions and position with respect to axis of rotation, the amount of fluid, and amplitude or frequency of motion, [1], displays a character ranging from small-amplitude short waves formation, non-linear standing waves to highly non-linear hydraulic jumps or combinations of the above, [2]. Also the dynamic pressures exerted on the tank surface are of non-linear nature as they comprise both non-impulsive loads related to fluid transfer as well as impulsive localised loading, ref. [2].

Published research on the subject exhibits a variety in levels of sophistication and type of approaches towards solving these problems. Two classes of approaches can be broadly distinguished: techniques employing latest advancements in science of computational fluid dynamics (CFD) and simplified methods based on rigid-body theory.

Recent studies on coupled ship motion and water sloshing, addressing the first of the above appraoches, have been reported by Mikelis et al, [9], Francescutto et al, ref. [10], Bass et al, ref. [11] or de Daalen et al, ref. [12], where the excited due to tank/ship motion internal fluid behaviour is dealt with by solving the Navier-Stokes equation numerically and coupling it with the simultaneous time-domain solution of more or less complex equations of intact ship motions with then fluid forces taken as external input. Further, de Veer et al, ref. [13], showed some attempts to predict in a similar manner effects of water ingress, with the rate of flooding itself estimated from Bernoulli equation. In addition to water sloshing coupled to 6d.o.f. ship motion prediction model, Woodburn et al, [14], accounts for some fluid interaction between the internal water and the outside sea domain to represent water ingress/egress in somewhat more sophisticated manner.

There does not seem to be much of a doubt that in the fairly foreseeable future these approaches will become a naval architect's routine procedures. As is the general consensus at present, however, these methods require excessive computational as well as expert efforts, preventing their methodological application in studies on dynamic ship stability.

The second class of approach, therefore, has found considerable research interest and recognition of the balance between simplicity and sufficiently meaningful representation of physics. Here, the mass of the liquid is regarded as behaving like a pendulum attached to the ship, with its mass located at the centre of the fluid buoyancy, which in turn is found from intersection of the tank geometry and fluid free surface assumed flat. The fluid free surface is most commonly assumed to always remain parralel to the sea level, e.g. Vassalos et al, [4], [5], de Kat, [3], or more recently assumed to be moving in accordance with some basic physics motion mechnism, e.g. Papanikolaou et al, [7].

The purpose of this paper is to discuss some fundamental validation study on the implications of the above-mentioned simplifications in modelling of the fluid behaviour onboard the flooded ship and building on that demonstrate the degree of agreement achieved in predicting damaged ship dynamics. For this purpose, a very brief overview of the mathematical model for generalised ship motion is given, followed by some details of the floodwater motion mechanism under study. Next, results of numerical simulations of bench-testing of water sloshing derived experimentally by de Bosh and Vugts, ref. [1], are presented together with discussions. Finally, the outcome of predictions of the damage ship dynamic behaviour by means of such an approach is demonstrated, with the concluded nuances pointed out.

2. GENERALISED SHIP MOTION MODEL

Equations for damaged ship behaviour description are derived from fundamental motion principles: the conservation of linear and angular momentum law. The law applied for rigid bodies, whereby this definition is also extended on the internal fluid mass, is resolved in body-fixed system of reference, see Figure 1. Rigorous derivation leads to a set of 6 scalar equations for linear and angular motions. Three such equations for angular motions are presented here in vector form (1).

The right hand side of the equation, $\vec{M}'_{Gs}$, and respective force vector in the set of equations for rectilinear motions, represents all the external forces and moments acting on the vessel expressed in a body-fixed system of reference, $G_s xyz$, located at the ship centre of mass. These forces are predicted with conventional for Naval Architecture methods. The Froude-Krylov and restoring forces and moments are integrated up-to the instantaneous wave elevation, the radiation and diffraction forces and moments are derived from linear potential flow theory and expressed in time domain based on convolution and spectral techniques, respectively. The hull asymmetry due to ship flooding, is taken into account by a "database" approach, whereby the hydrodynamic coefficients are predicted beforehand, and then interpolated during the simulation. The correction for viscous effects on roll and yaw modes of motion is applied based on well-established empirical methods. The second order drift and current effects are also catered for, at present, based on parametric formulations. Naturally the gravity force and moment vectors correspond to ship and flood water weights.

$$
\begin{aligned}
&(I'_s+I'_w)\cdot\frac{d}{dt}\vec{w}'+M_w\cdot\left[\vec{r}'_w\times\left[\frac{d}{dt}\vec{v}'_{Gs}\right]\right]+\\
&+M_w\cdot\left[(\vec{w}'\times\vec{r}'_w)\times\vec{v}'_w\right]+\\
&+M_w\cdot\left[\vec{r}'_w\times\left[\frac{d}{dt}\vec{v}'_w+\vec{w}'\times(\vec{v}'_{Gs}+\vec{v}'_w)\right]\right]+\\
&+\frac{d}{dt}M_w\cdot\left[\vec{r}'_w\times(\vec{v}'_{Gs}+\vec{v}'_w)\right]+\\
&+\left(\frac{d}{dt}I'_w\right)\cdot\vec{w}'+\vec{w}'\times\left[(I'_s+I'_w)\cdot\vec{w}'\right]=\vec{M}'_{Gs}
\end{aligned}
\qquad (1)
$$

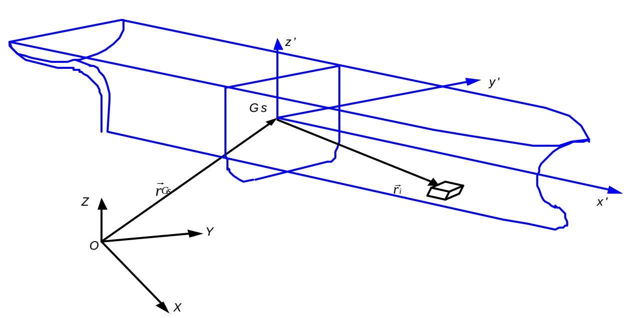

Figure 1 Coordinate system fixed to the centre of gravity of the intact vessel

The whole system, after re-arranging into matrix form as a set of twelve differential equations of the first order, are solved for position in space of the centre of gravity of the intact ship $\vec{r}_{Gs} = \int \vec{v}_{Gs} \cdot dt$ and three rotations through a 4th order Runge-Kutta-Feldberg integration scheme with variable step size.

3. INTERNAL SLOSHING MODEL

Still undetermined in equation (1), are the relevant vectors for floodwater location, velocity and acceleration, $\vec{r}'_w$, $\vec{v}'_w$ and $\frac{d}{dt}\vec{v}'_w$, respectively. These are the quantities that must be derived from a model representing the sloshing water phenomenon. In case of application of the CFD techniques, these vectors and relevant forces and moments can be derived from pressure integration due to fluid motion. Here, however, simplifications as mentioned in the foregoing, are adopted.

A model, the initial concept of which was presented by Papanikolaou et al in [7], has been developed as a free mass point moving due to the acceleration field and restrained geometrically by predetermined potential surfaces of centre of buoyancy for given amount of floodwater, FMPS (**F**ree **M**ass in **P**otential **S**urface), see Figure 2. This model derived from simple rigid body motion consideration, similar to that leading to equations (1), is presented as a set of equations (2), with graphical explanation in Figure 2:

$$\begin{cases} \frac{d}{dt}\vec{r}'_w = \vec{v}'_w - (\vec{v}'_w \cdot \vec{n}') \cdot \vec{n}' \\ \frac{d}{dt}\vec{v}'_w = \vec{a}'_f - (\vec{a}'_f \cdot \vec{n}') \cdot \vec{n}' \end{cases} \qquad (2)$$

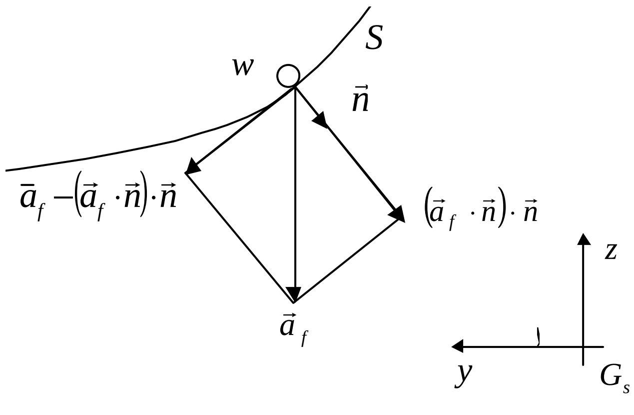

Figure 2 Fluid particle “w” (centre of buoyancy) in acceleration field $\vec{a}_f$ moving on the potential surface S All the vectors are resolved in $G_s xyz$ system of reference.

The total forcing acceleration vector is:

$$\vec{a}'_f = \vec{g}' - \vec{a}'_s - 2 \cdot \vec{w}' \times \vec{v}'_w - m^* \cdot \vec{v}'_w \qquad (3)$$

Where $\vec{a}_s$, see equation (4), is ship motion-related acceleration vector expressed in body-fixed system of reference.

$$\vec{a}'_S = \frac{d}{dt}\vec{v}'_{Gs} + \frac{d}{dt}\vec{w}' \times \vec{r}'_w + \vec{w}' \times (\vec{v}'_{Gs} + \vec{w}' \times \vec{r}'_w) \qquad (4)$$

$\vec{n}$ is the instantaneous normal vector to the potential surface of floodwater motion, determined from a damage compartment geometry database. Note that the vector is a function of $\vec{r}_w$ and volume of the fluid. Finally, m^* is an artificial coefficient introduced to represent damping of floodwater motion. This coefficient is an *ad hoc* adopted value derived for simple box-shaped compartment from comparisons with experimental data, as discussed later.

With the geometric information about the tank stored in a database, the model is complete. Equation (2) is set up for each flooded compartment within the ship and solved simultaneously with the equations for ship motion.

Having determined fluid motion, the forces and moments due to its displacement can be calculated. For demonstration purposes, the moment vector extracted from equation (1) is used and presented in the form of equation (5), where three components are distinguished to represent inertial moment, gravity moment and non-linear moment, see equations (6), (7) and (8), respectively. Note here that the fluid inertia matrix, I'_w, contains only the inertia of a single mass point located at a position $\vec{r}'_w$ in the ship-fixed system of reference at G_s. Since the mass is constant, the terms containing the time derivative of mass disappear.

$$\vec{M}'_{wat} = \vec{M}'_I + \vec{M}'_g + \vec{M}'_N \qquad (5)$$

Where:

$$\vec{M}'_I = I'_w \cdot \frac{d}{dt}\vec{w}' \qquad (6)$$

$$\vec{M}'_g = M_w \cdot \vec{r}'_w \times \vec{g}' \qquad (7)$$

$$\begin{aligned} \vec{M}'_N = {} & M_w \cdot \left[(\vec{w}' \times \vec{r}'_w) \times \vec{v}'_w \right] + \\ & + M_w \cdot \left[\vec{r}'_w \times \left[\frac{d}{dt}\vec{v}'_w + \vec{w}' \times (\vec{v}'_w) \right] \right] + \\ & + \vec{w}' \times \left[(I'_w) \cdot \vec{w}' \right] \end{aligned} \qquad (8)$$

4. NUMERICAL STUDIES

Experiments performed by de Bosch and Vugts, [1], have been the basis for studies on fluid sloshing described in this paper. In their experimental research, they performed a series of bench testing on the behaviour of the fluid in the box-shaped tank. The tank, with dimensions of 0.1m in length, 1.0m in breadth and 0.5m in depth, was filled with water, and excited at a range of rotation amplitudes and frequencies. The tank moment amplitude, K_a, as well as the angle e by which the moment lags behind the rolling was recorded. The moment was expressed as:

$$m_{wat}(t)\big|_x = K_a \cdot \sin(w \cdot t - e)$$

Since the flow behaviour in such conditions displays very complex nature, as mentioned earlier on, it was perceived of great interest to quantify to what degree the fluid loads can be predicted by simplified methods such as discussed in this paper.

After successful demonstration that a pendulum motion can be accurately simulated by model (2), see Figure 3, a basic prediction of tank natural frequencies was undertaken (note that the dimensions of the tank used for this exercise were 20m in breadth, 20m in length and 20/90m in depth). By comparison with an analytical solution (9), the test revealed that natural frequency could be predicted with reasonable accuracy only for lower filling ratios (fluid height to breadth of the tank), as shown in Figure 4. As the filling increases, the surface over which the centre of volume can travel decreases, and ultimately becomes a single point for full tank. Therefore, the natural period of such tank decreases rapidly if the filling height exceeds approximately half of the tank depth. The same tendency of under-prediction of natural period for higher filling ratios (h/b > 0.3-0.4) is noted also for greater tank depths.

$$w_S = \sqrt{\frac{g \cdot p}{b} \cdot \tanh\left(\frac{h \cdot p}{b}\right)} \qquad (9)$$

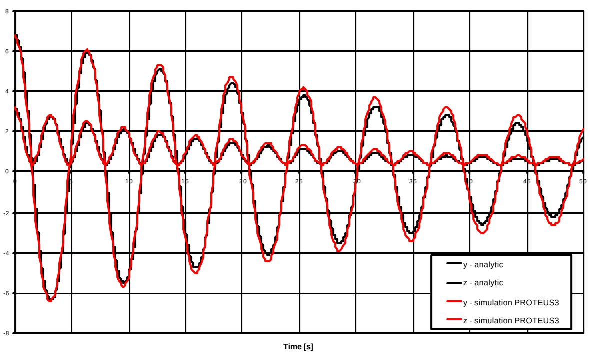

Figure 3 Simulation of the free motion of mass point in sphere-shaped tank, compared with the analytic solution

Further tests with imposed harmonic oscillations were performed to estimate the tank response in terms of forces generated by the fluid. Figure 10 and Figure 11 show amplitudes and phases, respectively, of the total moment (5) around rotation axis x. Notable in these figures is the effect of the damping coefficient m^*, the value for which thereafter has been adopted as 0.15. For lower values of this coefficient, the predicted moment shows characteristics of an under-damped spring-mass system. With the damping adjusted as mentioned, however, the predicted moment amplitude and phase compare very favourably with the measurements. Although the calculated amplitude curve shows slight difference, as it resembles typical damped spring-mass systems behaviour, it is the accurate estimation of the phase angle, which renders the modelling a very reliable tool for sloshing estimations. Since the filling ratio is relatively low, the natural period is predicted also quite accurately.

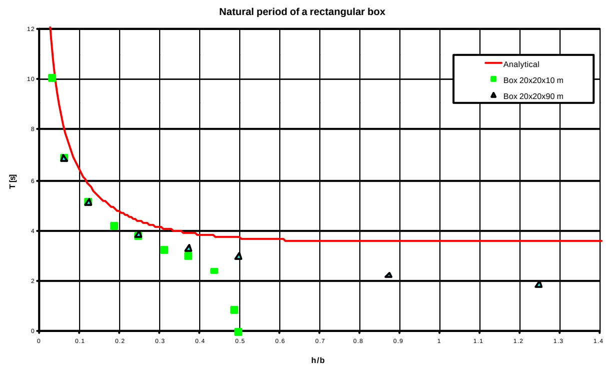

Figure 4 A comparison between the theoretical natural period for a rectangular box and the simulated natural period of fluid motion based on the FMPS model

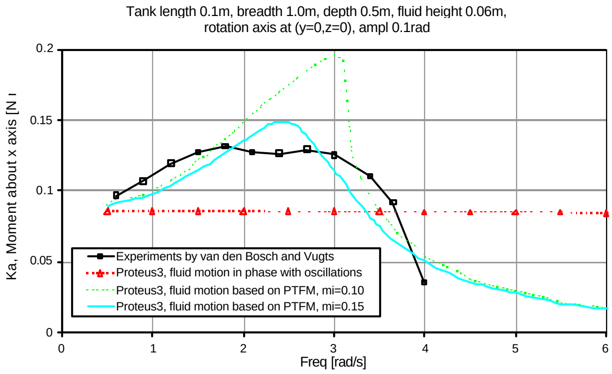

Figure 10 Comparison of fluid moment amplitudes derived by experiments and in-phase and FMPS sloshing models

Additionally, results from a simpler model are shown where the floodwater free surface is assumed to move in phase with the ship rotations. Note that in this case the velocity and acceleration vectors, seen in (8), can be derived by means of backward differentiation on the instantaneously estimated centre of buoyancy. As can be seen from Figure 10, the moment remains virtually constant irrespective of the frequency of oscillation, with the phase angle shown in Figure 11, by assumption being zero. The moment amplitudes are considerably lower than the values measured experimentally or predicted by model (2) for most of the frequency range, implying that the free surface slopes derived in the latter are consistently exceeding the rotation amplitudes.

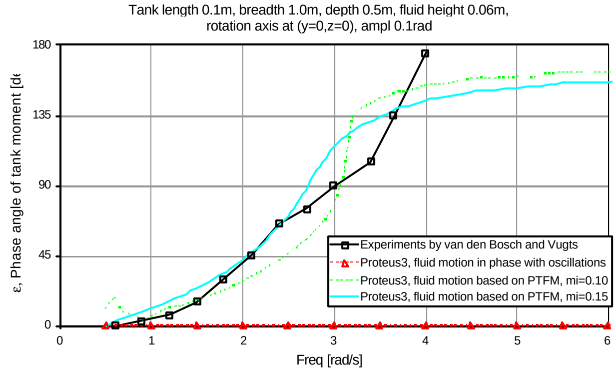

Figure 11 Comparison of fluid moment phase angles derived by experiments and in-phase and FMPS sloshing models

Simulations with different filling ratios confirm consistent predictions of the amplitudes of the tank moments, as is shown in Figure 12.

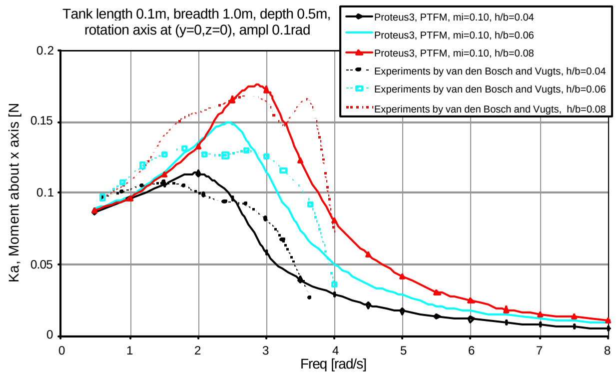

Figure 12 Comparison of fluid moment amplitudes derived by experiments and FMPS sloshing models. Effect of filling ratio.

Finally, partially surprising it was discovered that the most predominant component of the water sloshing moment in this case is due to gravity, as is shown in Figure 13. However, some further testing showed that the non-linear terms are of considerable importance for greater filling ratios (h/b~0.25), which is demonstrated in Figure 14.

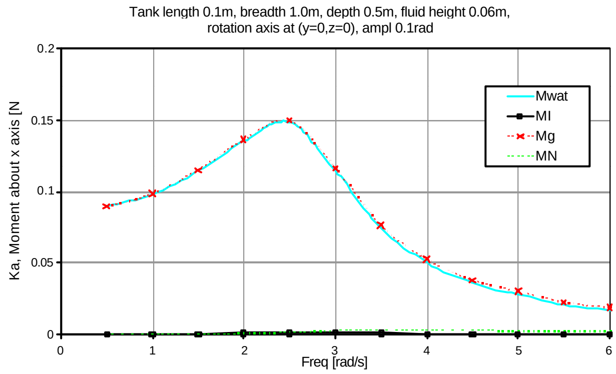

Figure 13 Fluid moment amplitudes derived by FMPS sloshing model. Comparison between different moment components. For low filling ratio, the gravity moment is the predominant component

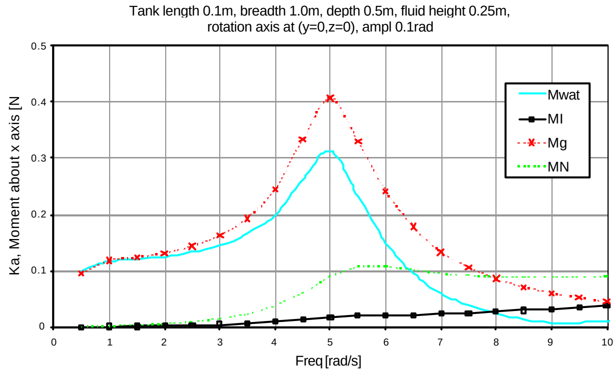

Figure 14 Fluid moment amplitudes derived by FMPS sloshing model. Comparison between different moment components to elucidate importance of non-linear terms for higher fill ratios (h/b~0.25)

The very fundamental case studies discussed above demonstrate that the techniques presented in this paper for predictions of fluid sloshing and its effects, can be confidently applied for examining the dynamic stability of flooded ships. This derives from the fact that the main load components due to fluid transfer can be modelled from basic of dynamic laws, and that the highly non-linear effects present during water sloshing are of minor importance, perhaps relevant for more focused studies on e.g. impulsive loads on localised elements of tank structures.

Length between perpendiculars	170.00	m
Subdivision Length	178.75	m
Breadth	27.80	m
Depth to subdivision deck	9.00	m
Depth to E-Deck	14.85	m
Service Draught	6.25	m
Displacement	17301.7	t
KMT	15.522	m
KG	12.892	m

Deriving from this conclusion, a study into basic dynamic behaviour of a damaged ship has been undertaken. A representative of typical modern passenger

Ro-Ro ship is used in this study, with its general particulars given in the table above and Figure 15. The frequency roll response curve derived numerically as well as by means of physical testing for the intact ship is presented in Figure 15. The agreement achieved is satisfactory. The comparison of the derived responses in damaged condition, however, has proved less favourable. As can be seen in Figure 17, the numerically derived roll response curve does not show any noticeable change in the natural frequency of the damaged ship, which phenomenon is clearly seen in the experimental data. Also the damping present in the damage ship system does not seem to be reproduced sufficiently high.

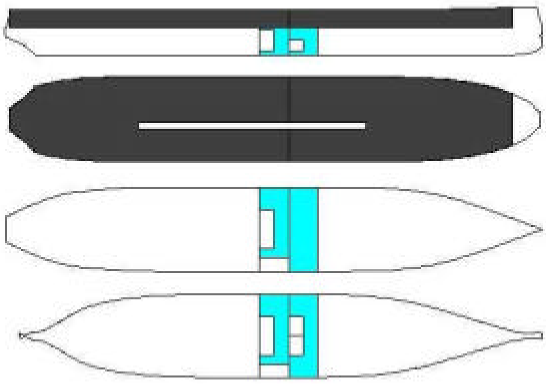

Figure 15 Internal arrangement of damaged compartments on PRR1 vessel.

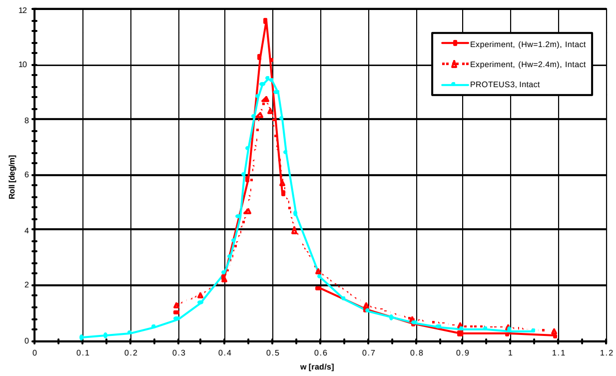

Figure 16 Roll frequency response curve for Ro-Ro vessel PRR1 in intact condition

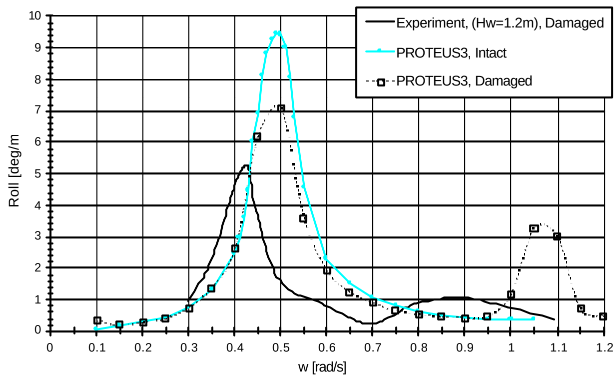

Figure 17 Roll frequency response curve for Ro-Ro vessel PRR1 in damaged condition

Bearing in mind the evidence presented in the foregoing on the ability to represent fluid sloshing in a closed tank with sufficient accuracy, to model intact ship behaviour accurately, and assuming that <u>variation in the ship hydrostatic properties due to damage is negligible</u> for the relevant roll range of up to 10deg, as shown in Figure 18 (GM_I=2.6m, GM_d=2.4m), the following have been suggested as the most likely sources of the discrepancy in modelling of the damaged ship dynamics by the presented method:

a) The natural frequency of the flooded compartment below the car deck and therefore the phase angle between the ship roll and fluid loads are not represented accurately, (approximately 70% of the space, h/b~0.25, is flooded).
b) The constant water ingress/egress (represented herewith by Bernoulli equation) affect the internal fluid behaviour.
c) The constant water ingress/egress affect the ship hydrodynamic properties.

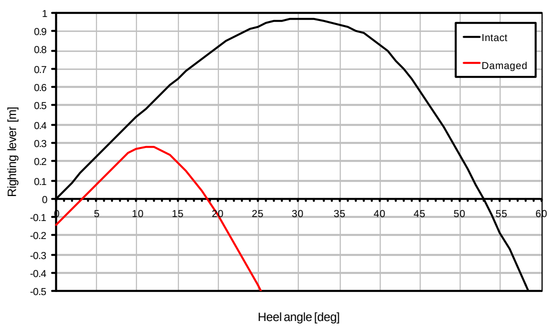

Figure 18 GZ curve for intact and damaged conditions, PRR1 vessel

As a first steps to investigate the above point (c) an *ad-hoc* adjustment has been made, whereby the total roll inertia of the ship has been increased by ~24% (2.2 times the added roll moment of inertia) and the predictions of viscous roll damping with the well known formulae by Himeno, [17], has been increased fivefold. The results of predictions of frequency roll response curves in damaged conditions after such modifications are shown in Figure 19.

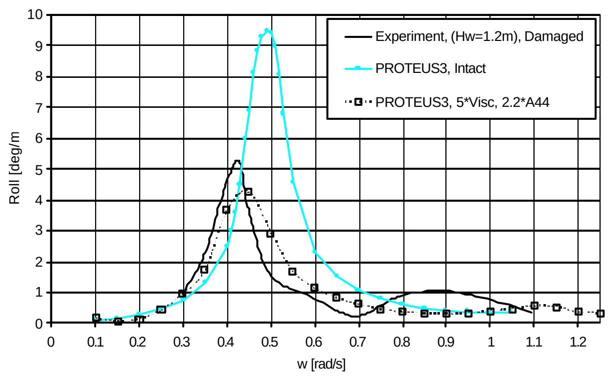

Figure 19 Roll frequency response curve for Ro-Ro vessel PRR1 in damaged condition, adjusted coefficients

The study into the problems highlighted above is ongoing and it is hoped that the sources of discrepancies will soon be identified and possibly resolved.

5. CONCLUSIONS

A mathematical model for the prediction of damaged ship dynamics has been presented. The emphasis has been put on outlining the model for water sloshing. Validation studies undertaken have demonstrated reliability of simplifed modelling of the fluid sloshing phenomenon. Nothwithstanding these advancements, however, some discrepancies in predicting basic dynamics of a damaged ship with water ingress/egress have been identified. Some reasoning behind this has been put forward. However, no firm conclusions can be made at present.

6. ACKNOWLEDGEMENTS

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