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CONTROL OF SHIP CAPSIZE IN STERN QUARTERING SEAS

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Abstract: A non-linear mathematical model for the roll-yaw behaviour of a ship is used to predict capsizing of a small tanker which sank in the North Sea some years ago. This capsizing problem was initially simulated on an analogue computer by the Danish Maritime Authorities as well as being tank tested. The problem was simulated using the digital package SIMULINK, which produced comparable results indicating instability in waves of just less than 3 m in height. Validation of the results is attempted and a discussion of possible improvements to the model is given. Simulated responses of the tanker with simple hydrodynamic fin stabilisers show that capsizing could have been prevented by this means in waves up to 7 m in height. Active PID control using a simple full span elevon is used to show a factor of ten reduction in roll angle to much greater waves. This work is of use to ship designers illustrating that stability can be enhanced for a fraction of the cost of major redesign of the ship hull and can be tailored to load conditions.

Keywords: Ship capsizing, broaching, simulation, instability, ultimate half-roll, case study, SIMULINK, PID control

1. INTRODUCTION

A number of North Sea trawlers from the UK and the other EU countries have capsized in heavy seas in the last 30 years. Even quite large ships have capsized in the Pacific, and it is possible that they may have been capsized by large waves. At least one RO-RO vessel becomes a casualty each week (Vassalos et al 1997). The Maritime Authorities have exhaustively examined static stability of ships and the conditions recommended by the International Maritime Commission have been well adhered to, at least in US and European ships. Dynamic tests conducted in wind and wave basin (Pauling & Rosenburg 1959) showed however that static stability was not a guarantee of safety. Because of the high degree of coupling between the motions of ships in several axes especially when large motions occur it is still an important area of research, especially if the coupling is caused by non-linear terms. The earliest investigation of roll-yaw coupling was due to Froude (1955). With the extensive use of digital computers it became possible (Salvesen 1970) to calculate the wind and wave forces that resulted from various ship motions. Experimental and computer investigations were made (Bird & Odabasi 1975, Kure & Bang 1975, Kure 1976) to try to ascertain the cause of these sinkings that have become frequent.

One of the most serious such incidents was illustrated by the car ferry Wahine (Conolly 1972) disaster in 1968. This type of disaster is referred to as broaching. In this case the ship is travelling with a stern sea slightly to one quarter. The ship will experience

difficulty in steering with the rudders being increasingly ineffective. Large yaw angles will be experienced and the ship will roll through a large angle to leeward. The ship is said to be 'broached-to' and the breaking waves over the ship and the wind effects may be sufficient to capsizing the vessel.

In the 1990's Lin & Yim used the new subject of chaos to analyse the non-linear equations devised to represent the motion of ships in roll-sway coupled motions, although the first use of such methods was by (Kuo & Odabasi 1975).

They showed four types of capsizing:

- *Non-oscillatory capsizing* in which the restoring moment is small compared with the moments of wind and waves exerted on the ship.
- *Oscillatory sudden capsizing* in this case restoring moment should be sufficient but instability is caused by successive series of waves.
- *Oscillatory symmetric build-up capsizing*, here amplitudes of rolling motion increase rapidly after only a few cycles similar to linear resonance. The build-up is likely to be caused by a series of waves.
- *Oscillatory anti-symmetric build-up capsizing*. In some cases the rolling motion appears to be anti-symmetric with respect to the axis of symmetry about the time axis. This again appears as the result of passing through a succession of waves producing oscillations, which are so large that recovery is impossible.

Spyrou (1996) has analysed a phenomenon known as *surf-riding* where the ship is stationary relative to the wave trough. The situation is an entrapment of the vessel for prolonged periods at exactly a zero encounter frequency. Spyrou goes on to show how for the controls (rudder) fixed condition surf riding is unstable. He then showed that active control could stabilise most of the states near to the trough, determining autopilot thresholds. He also identified stable conditions for the unsteered vessel up to a Froude Number of 0.36.

Hamamoto et al (1996), and Falzarano et al (1995) using only the roll equation of motion to simulate the problem of capsizing in stern seas, which were solved using the Runge-kutta-Gill method (Hamamoto), show conclusively that a ship with a linear GZ curve cannot be made to capsize in the condition of a stern sea with no sway motion present. They also show that for a non-linear GZ curve similar rapid capsizes as confirmed later by White (1988), reaching heel angles of 60° in less than 5 seconds. They concluded that for harmonic resonance to occur it is necessary to encounter the wave at the same period as the natural rolling period, which varies for a non-linear GZ with wave height. But the natural rolling period is too long to encounter in a real sea, as the wavelength would be about 500 m. Hence the possibility of experiencing a harmonic resonance cannot occur in a beam sea but only in a quartering sea where the encounter period varies with the speed of the ship.

Umeda et al (1995) describe model experiments, which show that exact type of capsizing is dictated by the character of the GZ curve, whether it is a softening or hardening spring. Umeda et al (1997) investigated the results of different equation modelling with 3 DoF and 4 DoF representations. They concluded that the 3 DoF model in surge, sway and yaw did not predict stability bounds well. The roll wave moment representation was crucial to good stability bounds prediction.

Model experiments by Hamamoto et al (1996) showed that capsizing due to harmonic resonance was at a Froude number higher than 0.3, while that due to parametric resonance occurred at a Froude number less than 0.25.

The work described in the paper is a case-study based on the Edith Terkol sinking.

2. THE CASE STUDY

A small Danish tanker, the Edith Terkol, having GZ-curves complying with the IMCO recommendations capsized in the Baltic Sea near the Swedish Island of Gotland. The tanker was steaming in ballast in a stern quartering sea. The weather was Beaufort scale 6-7. She was rolling heavily and quite suddenly capsized. Only two people survived. The precise parameters

are given in the report by Kure and Bang and Kure et al who performed a thorough investigation using a ship wave tank and an analogue computer to simulate the situation. Scale model tests in a wave basin, which were recorded on film, are illustrated in figure 1, with a time history of its' capsizing shown in figure 2. The model tests included representation of the wind forces believed to apply at the time. Tests with the model in the loaded condition did not produce the catastrophic capsizing seen in the ballasted condition. It is clear that the ship capsized in a very short time and it is a wonder that any crew survived. The metacentric height of the model for capsizing was slightly greater than that for the real case. The physical model capsizing took place after a few waves had hit the model as in the analogue simulation.

3. MATHEMATICAL MODEL

Linear equations of motion for coupled roll-yaw motions of a ship in a seaway are derived from Salvesen et al adding a non-linear term for roll moment. These agree in form with those given by (Lloyd 1998) and Spyrou but miss out the equations in heave and surge. The evaluation of the coefficients is not quite the same, the more recent text using a more refined strip model. The form of wave input is a simplification of that used by Spyrou. Kure and Bang evaluated the validity of these equations and found that the yaw equation could be ignored, as the roll coupling was small. Kure et al also left out the sway velocity term in the roll equation. The more recent analysis of Spyrou includes the surge equation, which enables the surf riding condition to be evaluated. These equations are definitely non-linear and there is a softening spring term in the roll restoring moment and its time variation. The GZ term was obtained from experimental data by allowing for the increased wave height by setting

$$GZ = C_{44}\phi + D_{44}\phi^3 \quad (1)$$

$$\begin{aligned} \text{The oscillation due to wave passage is introduced:} \\ C_{44} = C_{440} + C_{440}\sin(\omega t + \gamma) \end{aligned} \quad (2)$$

and making

$$C_{441} = f(\text{wave amplitude}) = C_{442}H \quad (3)$$

Where D_{44} is given a negative value to make $GZ=0$ at a specified angle of heel. H is the wave amplitude and C_{442} has been determined from digital computation of the righting moments in the quasi-static case of the ship on a wave crest-trough with the Smith effect included.

The final form of the equations of motion is:

Roll

$$(A_{44} + I_{44})\ddot{\phi} + B_{44}\dot{\phi} + (C_{440} + C_{442}H\sin(\omega t + \gamma))\phi + D_{44}\phi^3 + (A_{42} - MZG)\ddot{y} = H(F_{41} + jF_{42})e^{j\omega t} \quad (4)$$

Sway

$$(A_{22} + M)\ddot{y} + (A_{24} - MZG)\ddot{\phi} = H(F_{21} + jF_{22})e^{j\omega t} \quad (5)$$

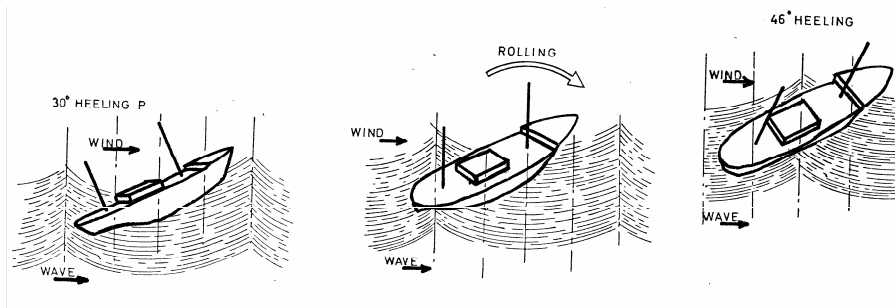


Figure 1: Ship model test observation (adapted from Kure)

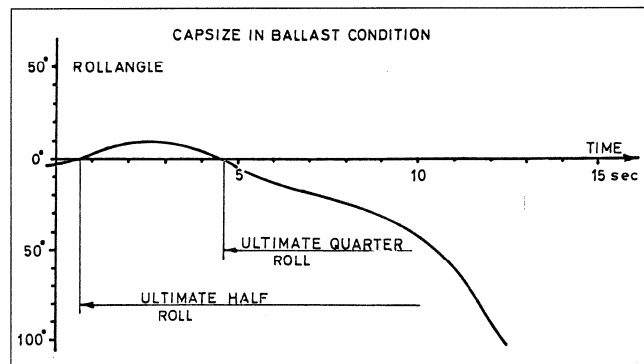


Figure 2: Time history from model test

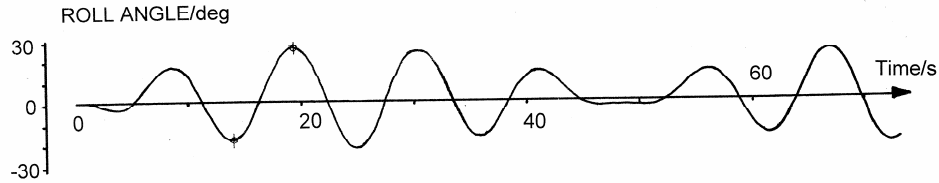


Figure 3: Time history for 4 m waves from Kure

The solution of these simultaneous, non-linear differential equations with time-dependent coefficients yields the sway and roll motion of the vessel. The Forces and Moments due to waves are converted from the complex form indicated above to a trigonometric form using the wave moment as reference

$$H(F_{41} + jF_{42})e^{j\omega t} = H(F_{41}^2 + F_{42}^2)^{1/2} \sin(\omega t) \tag{6}$$

$$H(F_{21} + jF_{22})e^{j\omega t} = H(F_{21}^2 + F_{22}^2)^{1/2} \sin(\omega t + \theta)$$

The coefficients were re-evaluated by the authors using the strip theory of Kure et al and minor variations were obtained but not sufficient to change the main results.

When the linear version of these equations is used the ship was stable in the given sea conditions.

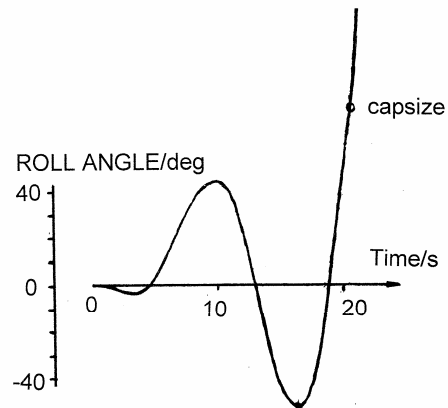
4. SIMULATION

The same equations as those in Kure et al were used in order to be able to compare their analogue computer results as well. Kure’s team used only the equations shown above missing out the heave and surge equations due to a limited number of analogue computer amplifiers

4.1 Conditions

The real ship was travelling at a speed 10 knots with quartering waves 30° off the stern and of about 100 m in length. Referred to Earth the waves had a period of

8 s. These are Doppler shifted to about 12 s close to the linearised rolling period of 10.6 s for the ship. Results for wave heights of 4 and 5 m are shown in figures 3 & 4.



Capsize is shown in figure 4 after only 20 seconds!

The model tank experiments shown in figure 2 also show rapid capsize. Although not stated explicitly it appears that the analogue model is accurate to within 20% of the scale model experiments judging from some of the linearised responses.

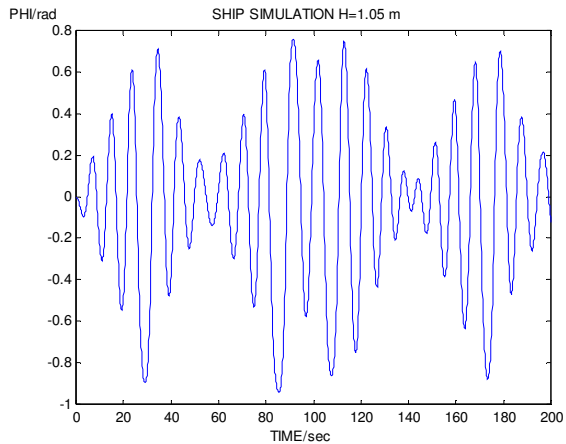


Figure 5: Time history for digital simulation for 1 m waves

The data for the ship is given in Table 1, for the capsize condition in Table 2 and for the GZ curve in Table 3.

Because equations (4) & (5) create an arithmetic loop inserting (5) into (4) formed a new equation.

$$(A_{22} + M)\ddot{y} = -(A_{24} - MZG)\ddot{\phi} + H(F_{21} + jF_{22})e^{j\omega t}$$

And

$$(A_{44} + I_{44})\ddot{\phi} + B_{44}\dot{\phi} + (C_{440} + C_{442}H\sin(\omega t + \gamma))\phi + D_{44}\phi^3 + \frac{(A_{42} - MZG)}{(A_{22} + M)} \left((MZG - A_{24})\ddot{\phi} + H(F_{21} + jF_{22})e^{j\omega t} \right) = H(F_{41} + jF_{42})e^{j\omega t} \quad (7)$$

This leads to the programme equations:

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= (B - B_{44}x_2 - C_{44}x_1 - D_{44}x_1^3)/A \\ \dot{x}_3 &= x_4 \\ \dot{x}_4 &= \frac{(U_2 - (A_{24} - MZG)\dot{x}_2)}{(A_{22} + M)} \end{aligned} \quad (8)$$

$$\begin{aligned} x_1 &= \phi \\ x_2 &= \dot{\phi} \\ x_3 &= y \\ x_4 &= \dot{y} \end{aligned} \quad (9)$$

and

Where

$$C_{44} = C_{440} + C_{442}H\sin(\omega t + \gamma)$$

$$A = A_{44} + M + (A_{42} - MZG) \frac{(A_{24} - MZG)}{(A_{22} + M)}$$

$$B = U_1 - U_2 \left(\frac{A_{42} - MZG}{A_{22} + M} \right) \quad (10)$$

$$U_1 = HF_{412} \sin(\omega t)$$

$$U_2 = HF_{212} \sin(\omega t + \theta)$$

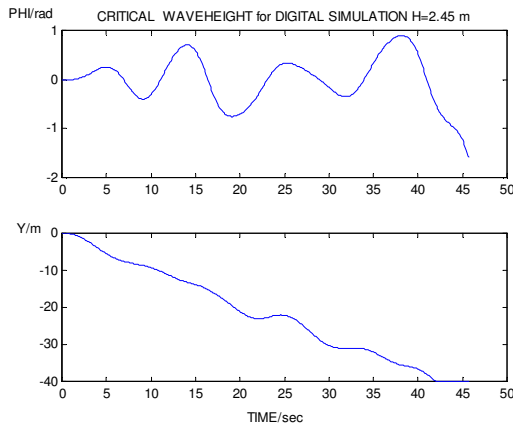


Figure 6: Unstable roll and sway motion for a wave height of 2.45 m

As can be seen from figure 5 the motion is stable for a wave height of 1 m whereas in figure 6 the ship is unstable with a wave height of 2.45 m. The integration routine was a Runge-Kutta order 4 with a step size of 0.01 seconds. The physical model is compared to the simulation in figure 7.

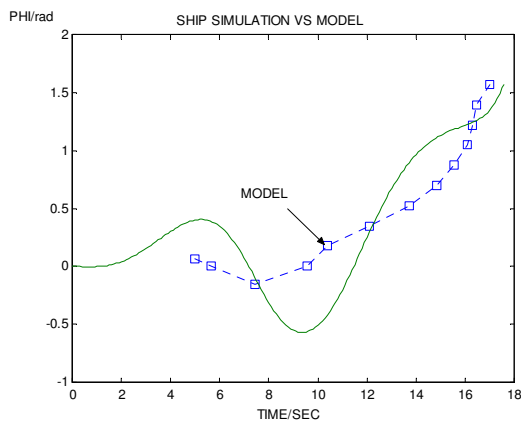


Figure 7: Digital simulation compared to model test H=3.5 m

4.3 SIMULINK Solution

The analogue computer diagram from (Kure 1976)

was directly constructed using SIMULINK. Numerical values were chosen according to the values given by Kure. Figure 8 shows the effect of small wave height changes. The overall behaviour is principally the same.

5. VERIFICATION AND VALIDATION

Verification and validation are required to make the programme useful. Without verification the programme does not calculate what you thought it did and without validation it does not match the real life situation in any meaningful way and is close to being useless.

5.1 Verification

To verify that the programme in SIMULINK delivered a known quantity all the non-linear terms were switched out and the time dependent coefficients switched out and a simple roll computation was made to yield a period of 10.6 seconds very close to the hand calculation. The sinusoidal forcing functions were then put back in to show a harmonic response of verified amplitude.

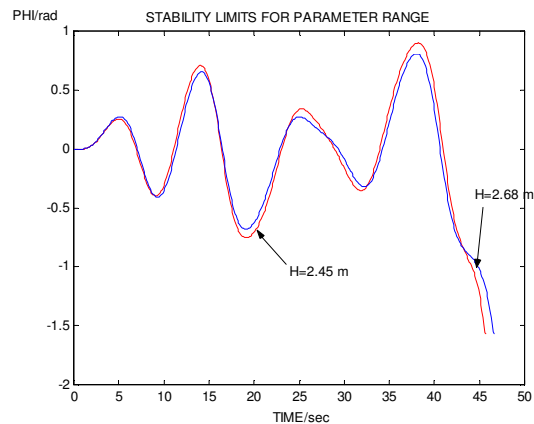


Figure 8: Effects of parameter variations

5.2 Validation

Validation can be divided into three parts:

1. the representation of the real problem by the equations used
2. the validity of any further approximations used for solution of the equations,
3. Representation of known physical events by the simulation and the accuracy of those solutions.

When these are fully understood and are assured then valid predictions can be made.

Let us now examine this simulation.

1. The equations used are forms of the general equations of motion for ships now widely

accepted (see Lloyd).

2. There are three sub-parts to the modifications used by Kure et al and similarly by us.
 - Only two of the equations of motion are used, for sway and roll. Spyrou uses roll, sway and surge equations and Hamamoto uses only the roll equation, but not for a particular case. None of these authors uses the heave equation to examine this stability problem. For symmetrical ships the longitudinal and the lateral planes are not coupled together. Ignoring the heave equation is reasonable as it is only weakly coupled by coefficient variation.
 - This is not true if the deviations are large. Hence when the roll angle is large the equations in this simulation are no longer valid. The justification for ignoring the surge equation is that it is completely uncoupled from the others (except by the modified coefficients which are a secondary effect). The sway terms left out are very small (~0.5%).
 - The representation of the non-linear GZ curve is taken from calculations based on the shape of the hull and the wave as it rises up the side of the ship. It is felt to be a good approximation.
3. The proof here is that the simulation and the model tests showed a similar behaviour, a sudden and sharp increase in roll angle after only a few waves hit the model ship and that agreed with survivors' account of the disaster that happened to the real ship. This is also true of all the other workers. However when the results of the various simulations are compared numerically they do not show such a good correlation. The peak roll before capsize is not the same. Scale model friction is very difficult to simulate. In this case a Froude simulation was used. This means that the Reynolds number was not the same and therefore the hydrodynamic boundary layer was not simulated. In a scale model exceptional smoothness has also to be achieved for scaling the surface roughness. Both lead to the possibility of a model surface friction force higher than the real ship. This would lead to reduced peak amplitudes. In figure 7 the digital results are compared to the model values.

A more detailed comparison follows in section 6.

To complete the validation we have to be sure that the integration of the equations did not depend on the integration step size or integration routine. No effects were found with the SIMULINK run. (Kring and Sclavovnos 1995) indicate numerical instability for some variable step algorithms used to solve similar equations but for ship heave motions with $\Delta t/(g/L)^{0.5} = 0.1$ whereas we were using 0.24 for a 4th order Runge-

Kutta method. For SIMULINK the same method was used.

For both the SIMULINK model three different integration methods were used with no variation in the answers.

6. DISCUSSION OF THE "EDITH TERKOL" DISASTER SIMULATION

The first observation to make is the overall similarity of the scale model, analogue computer, SIMULINK results. The gross stability pattern is revealed by all the techniques but at different wave amplitudes. Data from the sources were numerically re-evaluated and found to differ slightly from the analogue coefficients. A detailed comparison can be made from the figures where the results are plotted on the same axes for SIMULINK. These reveal very small discrepancies, particularly in the time that events take place.

6.1 Detailed Comparisons

The model basin results give a 10° roll before the ultimate half roll, whereas the analogue computer gives an angle of about 50°. The SIMULINK results are about 36° for the same event. Time to capsize is 16.5 s for the model test, 17 s in the SIMULINK simulation. The equivalent analogue computer run gave about 20 seconds.

Wave height for instability for the model test value is 3.5 m whereas the SIMULINK simulations gave a value of 2.45 m. The analogue computer solution gave approximately 4.6 m. The survivors stated that the sea conditions at the time of the disaster corresponded to a sea state with waves of 4 to 5 m in height.

6.1.1 Effect of wave frequency on ship roll stability

As indicated earlier the digital model ship was stable up to a wave height of $H=2.45$ m and the wave frequency was 0.7854 rad/s. Several further runs were made in which the wave frequency was systematically altered. Since in most cases the system was stable the duration was extended to 200 s and the time increment was increased to 1 s. Reducing the forcing frequency to 0.6 rad/s at the same wave height gives a larger response but with a longer beat period. Reducing the frequency to 0.55 rad/s gives smaller amplitudes. At an intermediate frequency of 0.57 rad/s we get a result similar to linear resonance but with a beat period of 170 s. Small increases of wave height, at this frequency, results in capsize.

These results agree qualitatively with those of Spyrou but do not agree numerically since he modelled a smaller ship. He obtained unstable motion at a Froude number of 0.36, $\lambda/L=2$ and $H/\lambda=0.05$ whereas we achieved unstable motion at a Froude number of 0.215, $\lambda/L=1.71$ and $H/\lambda=0.0245$. Spyrou showed

that the inclusion of propeller effects changed the margins quite considerably. This may explain why the computer results differ markedly from the model tests. A further explanation may be due to the amount of damping present. As will be shown later even small changes in damping radically changes the wave height to cause capsizes. The damping in the computer models is very small and does not include any hydrodynamic drag effects from a well-worn ship. Kuo's fourth description of unstable modes would fit best with the data for this example of ship behaviour.

7. STABILISATION

Stabilisation of ships is common practice in large vessels but not so common in small fishing vessels. All the modern panoply of control mechanisation has been used to control roll in vessels such as hydrofoils and catamarans (Yang et al. 2002). The main feature of this system if examined is the small, almost negligible amount of damping present in the system. If this could be improved then the catastrophic behaviour could be modified. To add damping a simple hydrofoil stabiliser pair was added to the ship simulation model (figure 9). If we apply strip theory to this foil (see Lloyd) then:

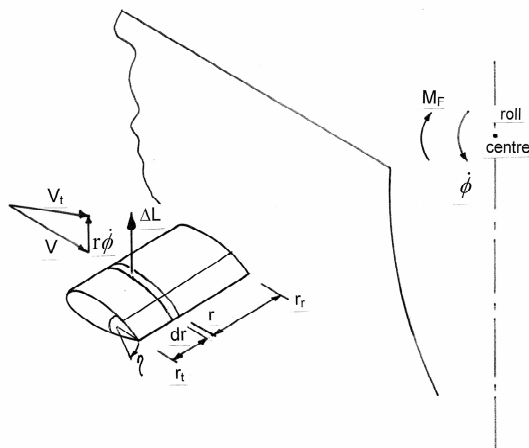


Figure 9: Hydrofoil schematic

$$\Delta L = \frac{1}{2} \rho V^2 C_L c dr$$

$$\Delta L = \frac{1}{2} \rho V^2 a \left(\frac{r \dot{\phi}}{V} \right) c dr \tag{11}$$

The total moment on the foil is:

$$M_f = \rho V_t^2 c a \left(\frac{\dot{\phi}}{V} \right) \int r^2 dr$$

$$M_f = \rho V_t^2 c \left(\frac{\dot{\phi}}{V} \right) \left[\frac{r_t^3}{3} - r_r^3 \right] \frac{a}{3} \tag{12}$$

This modifies the value of B_{44} (figure 10). The SIMULINK model is shown to be fairly simple with the model stopped when capsizes is reached. A chord of 0.5 m was chosen with $r_t = 5.825$ m with only a span of 1 m this changed the value of the damping coefficient from 5 to 416 kN m s. This now stabilises the ship model at wave heights of 7 m (figure 12). This passive damping still leaves the ship rolling quite badly, and doesn't prevent capsizes at much larger values of wave height. If we now introduce some active damping into the system as shown in figure 11, with a full span elevon coupled to a roll angle detector and driven via a PID controller (figure 13) then the results are spectacularly better. As shown in figures 14 & 15, the reduction is about 90% in angle of roll and for larger wave heights of 10m, which would overwhelm the passive system, the active control still reduces the angle of roll to about 2°! The addition of the foils and elevons will produce some asymmetric drag, which will couple the Yaw and roll together but the contribution is small. PID control is essentially an industry standard and appears to do the job quite adequately. A fully pivoting foil is usually used in larger ships but here would be less rigid and would pose a danger when fishing. So a full span elevon was chosen.

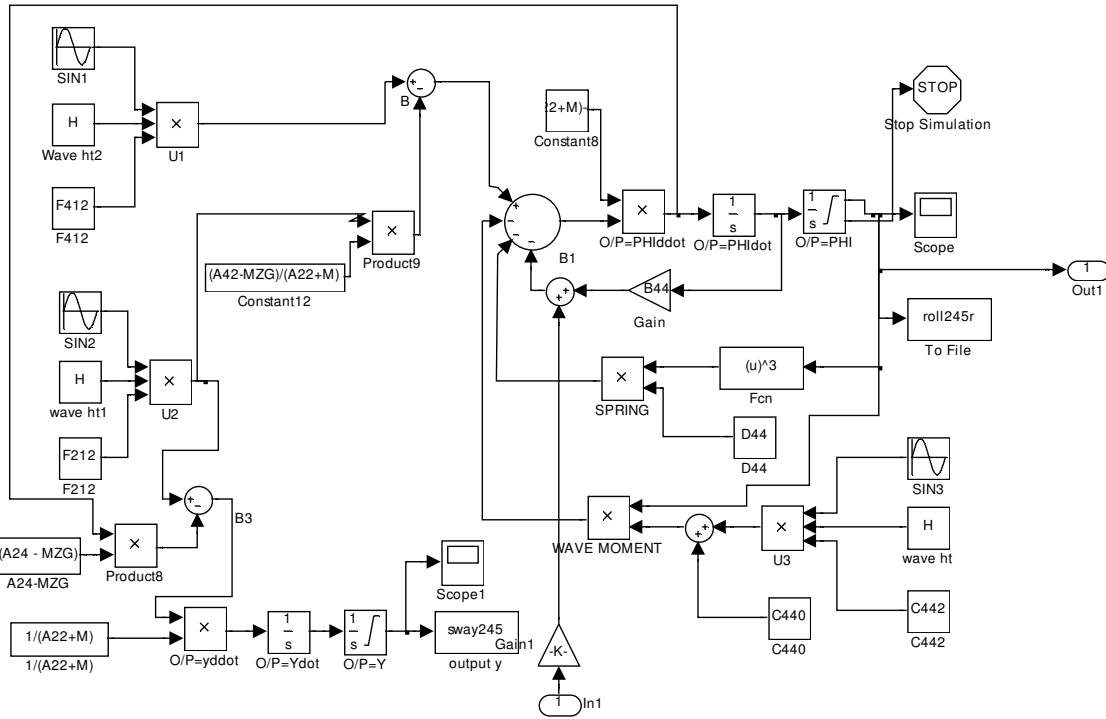


Figure 10: Simulink ship model

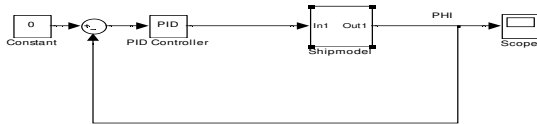


Figure 11: Ship control model

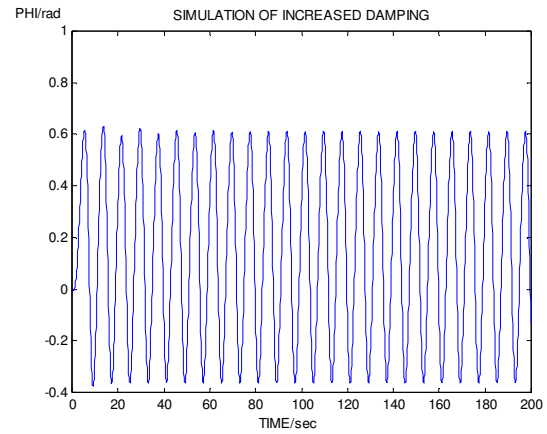


Figure 12: Effect of increased damping from the hydrofoil H=7 m

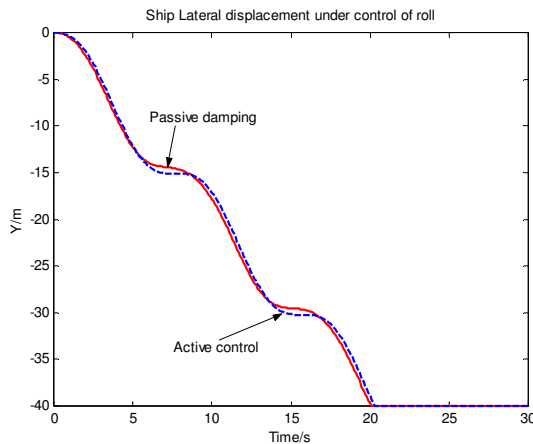


Figure 13: Lateral displacement for wave height 5m

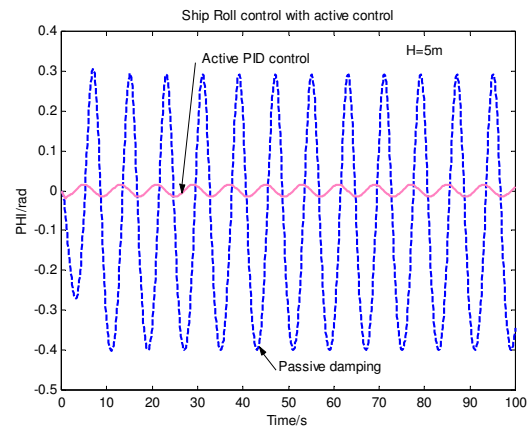


Figure 14: Ship roll control with wave height 5m

8. CONCLUSIONS

- a) A ship capsize has been modelled using strip theory for the hydrodynamic forces and moments. Static stability is represented as a softening spring.
- b) Simulation comparison has been undertaken using the physical model and analogue computer solution of Kure and Bang with that of the digital package.
- c) All the simulations and model tests agree with sudden half roll capsize in 20 seconds or less.
- d) The digital simulation agrees with the model tests.
- e) Predicted wave heights for instability are 3.5 m for the model test and 2.45 m for the digital simulator.
- f) Results from these simulations agree in character with those of other simulations in the literature.
- g) A simple hydrofoil passive damping system would have stabilised roll motions up to at least 7 m in wave height
- h) The active control system using PID controlled elevons produces even greater reduction in roll angle and is still effective at wave heights of at least 10m. In this case the ship would have been swamped by waves of this size before this condition was reached. This work is of use to ship designers illustrating that stability can be enhanced for a fraction of the cost of major redesign of the ship hull and can be tailored to load conditions. In effect the ship can be designed as in aircraft to have variable roll stability. The effects of fore and aft motions on the coupling ahs not been examined and would need to be analysed before installation. Different control algorithms may prove to be even more effective.

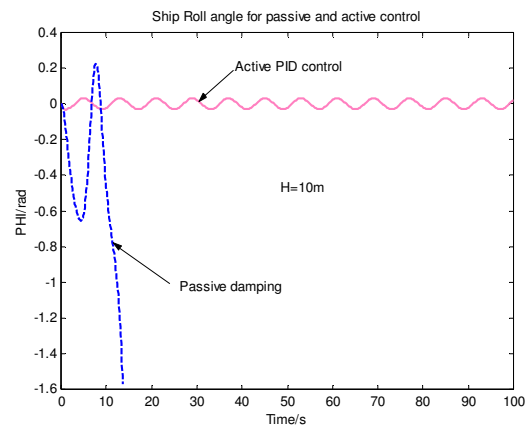


Figure 15: Ship roll for wave height 10m with control

NOMENCLATURE

Roman

- a Lift curve slope
- A_{22} Hydrodynamic mass in sway
- A_{24} Hydrodynamic coupling coefficient roll into sway
- A_{42} Hydrodynamic coupling coefficient sway into roll
- A_{44} Hydrodynamic mass moment of inertia in roll
- B_{44} Damping coefficient
- c Hydrofoil chord
- C_L Lift coefficient
- C_{44} Restoring moment stiffness
- C_{440} Non-wave stiffness
- C_{441} Wave stiffness
- C_{442} Normalised wave stiffness
- D_{44} Cubic term stiffness coefficient
- F_{21} Wave force, sway, real part
- F_{22} Wave force, sway, imaginary part
- F_{41} Wave moment, roll, real part

F_{42} Wave moment, roll, imaginary part
 G Centre of gravity of ship
 GM Metacentric height
 H Wave height
 I_{44} Mass moment of inertia of the ship
 ΔL Lift force on element of hydrofoil
 M Mass of the ship
 M_f Hydrodynamic moment on hydrofoil element
 r Radius from the centre of roll
 r_r Root radius
 r_t Tip radius
 y Sway motion
 ZG Vertical ordinate of G
Greek
 γ Phase angle
 ϕ Roll angle
 θ Phase angle
 ρ Water density

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Table 1: Ship data

| | |
|--------------|---------|
| Length | 58.6 m |
| Breadth | 9.65 m |
| Depth | 4.15 m |
| Tonnage | 498 BRT |
| Engine Power | 800 hp |

Table 2: Capsize data

| | |
|--------------------|--------------------|
| Displacement | 645 m ³ |
| Draft at | 1.75 m |
| Trim aft | 1.52 m |
| Metacentric Height | 0.64 m |

Table 3: GZ- curve particulars

| | GZ _{max} | GM _{max} deg | GM m |
|-----------------------------------|-------------------|--------------------------|---------|
| IMCO | 0.20 | 25 | 0.15 |
| Ballast (capsize condition) | 0.032 | 27 | 0.64 |
| Loaded | 0.29 | 46 | 0.65 |