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Limits to Control of Ship Capsize Using Cycloidal Propellers Compared to Active Fins

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Abstract. A non-linear mathematical model, for the roll-yaw behaviour of a ship, is used to predict the prevention of capsize of a small ship. The problem was simulated using the digital package SIMULINK. Simulated responses of the ship with simple hydrodynamic fin stabilisers show that capsize could have been prevented by this means in waves up to 7 m in height. Active control using a simple full span control flap is used to show a reduction in roll angle to much greater waves. These results are compared to the use of cycloidal propeller propulsion to provide lateral thrust and hence roll control, which showed that this method can also reduce the roll. In order for successful roll reduction with a cycloidal propeller the response time constant is determined. In this example the roll reduction is greater in the fin example but this ability falls off rapidly with forward speed. Roll control using the cycloidal propeller has the advantage of retaining its' effectiveness at low or zero speeds. To be fully effective for this ship, the reaction time to generate the side thrust from the roll actuation signal must be less than 13 seconds for the cycloidal propeller to prevent capsize. This would imply the use of the cycloidal propeller for roll control is better suited to larger ships.

Keywords: ship capsizes, broaching, simulation, instability, roll control, Voith Schneider propellers, SIMULINK, PID control

1 Introduction

Quite large ships have capsized in the Pacific, and it is possible that this may have been caused by very large waves. Vassalos et al.[36] asserted that at least one RO-RO vessel becomes a casualty each week. Dynamic tests conducted in wind and wave basins [12] showed however that good static stability was not a sufficient guarantee of safety. A high degree of coupling between the motions of ships in several axes exists caused by non-linear terms especially when large motions occur. With the extensive use of digital computers it became possible [15, 21, 25, 26, 30] to calculate the wind and wave forces that resulted from such complex ship motions.

One 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 capsize the vessel.

In the 1990's Lin & Yim[18] introduced the new subject of chaos to the analysis of the non-linear equations representing the motion of ships in roll-sway coupled motions. These methods have been further explored since that time for example [41].

They identified four classes of capsize:

(1) Non-oscillatory capsizing in which the restoring moment is small compared with the moments of wind and waves exerted on the ship.

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(2) Oscillatory sudden capsizing in this case restoring moment should be sufficient but instability is caused by successive series of waves.
(3) Oscillatory symmetric build-up capsizing, where the amplitude of rolling motion increases rapidly after only a few cycles.
(4) 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.

Spyrou[27] analysed a phenomenon known as surf-riding where the ship is stationary relative to the wave trough. The situation entraps 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 showing that active control could stabilise most of the states near to the trough. He also identified stable conditions for the unsteered vessel up to a Froude Number of 0.36.

Hamamoto et al[8] and Falzarano et al.[7] using only the roll equation of motion to simulate the problem of capsize in stern seas, which were solved using the Runge-kutta-Gill method showing conclusively that a ship with a linear GZ curve cannot be made to capsize in the condition of a stern sea if no sway motion is present. They also showed that for a non-linear GZ curve, rapid capsize occurs, reaching heel angles of 60^o in less than 5 seconds. But the natural rolling period is too long to encounter this 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.[35] investigated the results of different equation modelling with 3 DoF and 4 DoF representations. They concluded that the roll wave moment representation was crucial to good stability bounds prediction. Model experiments by Hamamoto et al.[9] showed that capsize 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.

Stabilising ships in roll falls mainly into three main different methods[19] with Perez[23] giving an exposition of the development of current methods and a comprehensive analysis tutorial:
(1) Stabilisation using active or passive tanks e.g. [6, 10, 22].
(2) Stabilisation using rudder control. Alarcin and Gulez[4] describe the use of a Neural Network controller for roll stabilisation of a fishing vessel using the rudder to control roll and yaw with a better performance than a LQR regulator. A linear model was used for the design of the controllers. Torta and Raspa[31] made an extended analysis of the achievable performance that can be obtained using the rudder as roll damping actuator. Baitis and Schmidt review the ship roll stabilization used in the US navy emphasizing the benefits of rudder roll stabilisation[5].
(3) Active and passive fin stabilisation. Ulusoy[34] compares the fin stabilisation with other methods using an advanced analysis simulation for a range of wave angles, directions and magnitudes finding reductions of roll angle reductions up to 60% while Surendran and Kiran[29] report analysis using Fluent to compute the hydrodynamics of the fin hull wave interactions. White[38] described the simulation of roll control of a small tanker in a stern quartering seas with non-linear terms in a simplified simulation but representing typical parametric resonance. The results of using active controlled fins illustrate substantial roll reduction and prevention of capsize for waves of up to 10m. These results were comparable to Ulusoy. Van Laarhoven[16] analyses the case of nonlinear parametric resonance and reports the use of fins and U tanks to reduce roll in such nonlinear resonance.

The methods that will work depend on the frequency response needed for a given situation. In the case of the stern quartering wave used here the active tanks are too slow to effect control. It is also unlikely that rudder actuation on its own will not produce enough torque to counteract the wave forces. The remaining method which will provide enough torque and sufficiently quick response is the active fin, since it is ineffective in stern seas. Having made the choice of method of actuation but then the control regime needs to be investigated. Basic control by PID (Proportional, Integral and Derivative terms), more modern methods such as adaptive[17], variable structure[40], Neural network[11] and Fuzzy control[28, 39] or sliding mode control[14] have been used. All of these controllers show excellent performance in controlling roll. The purpose of this paper is to evaluate the performance of two actuation methods of roll stabilisation and see what benefits may be obtained. For this reason a basic PID controller was used as an exemplar.

2 The case study

The work described in the paper is a case-study based on the same example as White[38] with the ship data given in Tab. 1, 2 & 3 but compares two different actuation methods. In this case the ship is to be propelled by a Voith Schneider (VS) cycloidal propeller[1, 2, 13] that allows steering to be achieved with the same device. The response of roll control using the VS propeller is compared to the active fin control. All the conditions were kept the same to enable a true comparison to be made. The model does not allow for waves breaking over the decks or the effects of the waves on the superstructure, but does represent the hydrodynamics of the effects of the waves on the hull quite well. The ship was modelled sailing in a stern quartering sea with wave heights of 5 and 10 m reported here. The model used here is not described by a complete set of 6 DoF equations because these are not necessary to prove the concept, however to fully analyse the whole concept for implementation then this would have to be undertaken at a later date.

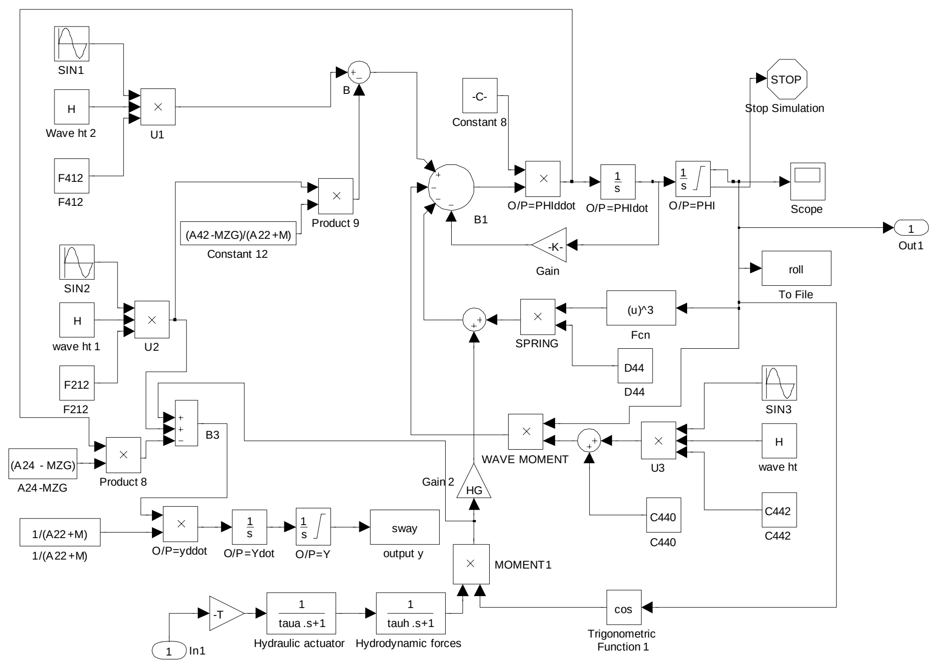

Fig. 1. Simulink model

3 Mathematical model

Linear equations of motion for coupled roll-yaw motions of a ship in a seaway with a non-linear term for roll moment. These agree in form with those given by Lloyd[20] and Spyrou[27] but miss out the equations in heave and surge. This limitation is a fair approximation in the open sea but close to land it would not be sufficient. The evaluation of the coefficients uses a more refined strip model than used in [38]. The form of wave input is a rational simplification of that used by Spyrou. The more recent analysis of Spyrou includes the surge equation, which enables the surf riding condition to be evaluated. These equations are non-linear and there is a softening spring term in the roll restoring moment and its time variation. The GZ term was computed from experimental data by allowing for the increased wave height.

$$\mathrm{GZ} = C_{44}\varphi + D_{44}\varphi^3, \tag{1}$$

where D_{44} is given a negative value to make $GZ = 0$ at a specified angle of heel. The oscillation due to wave passage is introduced by setting:

$$C_{44} = C_{440} + C_{441}\sin(\omega t + r) \tag{2}$$

making

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

where 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

$$(A_{44} + I_{44})\ddot{\varphi} + B_{44}\dot{\varphi} + (C_{440} + C_{441}H\sin(\omega t + r))\varphi + D_{44} + (A_{42} - MZG)\ddot{y} = H(F_{41} + jF_{42})e^{j\omega t}. \tag{4}$$

Sway

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

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.

To add damping a simple hydrofoil stabiliser pair was added to the ship simulation model (Fig. 1). If we apply strip theory to this foil[20] then:

$$\mathit{\Delta} L = \frac{1}{2}\rho V^2 C_L c\mathrm{d}r, \quad \mathit{\Delta} L = \frac{1}{2}\rho V^2 a(\frac{r\dot{\varphi}}{V})c\mathrm{d}r, \tag{6}$$

$$M_f = \rho {V_t}^2 ca(\frac{\dot{\varphi}}{V})\int r^2\mathrm{d}r, \quad M_f = \rho {V_t}^2 c(\frac{\dot{\varphi}}{V})[{r_t}^3 - {r_r}^3]\frac{a}{3}. \tag{7}$$

4 Simulation

The simulation was produced using SIMULINK© as shown in the model of Eq. (1-5) in Fig. 1. The integration routine was a Runge-Kutta order 4 with a step size of 0.01 seconds to match the earlier work. The real ship was travelling at a speed 10 knots with quartering waves 30^o 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.

The data for the ship is given in Tab. 1-3. Terms are included to enable the fin stabilisation or cycloidal propeller and controller to be simulated, with a similar transfer function representation of the hydraulic actuator.

5 Stabilisation

The main feature of these ship equations is the small amount of damping present in the system. If this could be improved then the catastrophic capsize behaviour could be reduced significantly. In the original work a simple hydrofoil stabiliser pair with controlled elevons was added to the ship model to add damping.

This modifies the value of B_{44}. 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 as high as 7 m for the passive fins. This passive damping still leaves the ship quite badly rolling at higher wave heights (Fig. 4), and doesn't prevent capsize at much larger values of wave height. If active damping is introduced into the system (Fig. 5) with a full span elevon coupled to a roll angle detector and driven via a PID controlled hydraulic actuator, which is modelled as a simple lag then the results are spectacularly better. The assumption of a simple time lag representation is reasonable as the behaviour of the system is dominated by the rotary inertia of the fins and it is a good first order approximation. The PID controller is an industry standard represented by the three terms as in Eq. (8).

$$\text{fin\_actuator\_signal} = \left(P + \frac{I}{s} + Ds\right)\text{error\_in\_roll\_angle}, \tag{8}$$

where s is the Laplace transform. The controller has three terms, a gain proportional to the error, a term depending on the rate of change of error and a term depending on the integral of the error. With these terms the overshoot, rate of acceleration and overall steady state error can be altered. The controller was tuned manually using the simulation to give minimum integral of time multiplied by absolute error in roll angle (ITAE). This only took a few minutes.

In a practical PID controller there are limits to the amount of actuation possible. Since this is a feasibility study these practical limits were not included here. As shown in Fig. 5 the reduction is about 90% in angle of roll and for larger wave heights of order 10 m, which would overwhelm the passive system, the active control still reduces the angle of roll to about 2^o!

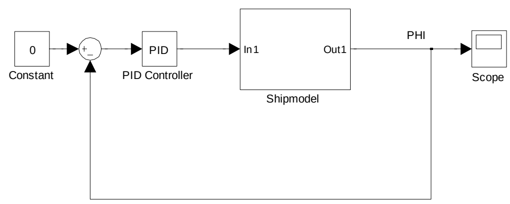

Fig. 2. Control schematic

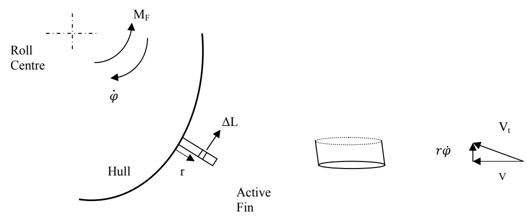

Fig. 3. Fin schematic

The second actuating device examined was a Voith Schneider (VS) cycloidal propeller. This consists of a number of aerofoil blades rotating about a common axis mounted so that the length of the blades is parallel to the axis of rotation and usually mounted vertically in the ship. The pitch of each blade is controlled cyclically to produce thrust in a given direction[1, 2, 13]. Since the thrust is applied underneath the vessel it is a powerful actuator for roll control. The question to be answered in the case of stern seas is whether it can be made to act fast enough to effect control. The model used here is to assume that the thrust response could be represented by a time constant dictating the time to achieve 67% of the nominal maximum thrust. This time constant is shown in the model Fig. 1. This was also controlled via a PID controller for simplicity since only the principle is being examined here. The hydraulic actuation is represented by a simple lag transfer function as indicated by [20, 24] with a time constant τ_a. The design charts[33, 37] for these propellers indicated that two 16R5 EC/120-1 units would be suitable to propel this ship. It is assumed that the propeller rotational speed would be unaltered and the sideways force would result from changing the cyclic angle of attack. The hydrodynamic forces generated by the propellers do not happen instantaneously and this time delay, τ_h is represented again by a lag function.

The results in Fig. 4,5 and 6 show that the installation of the cycloidal propeller can control the ship roll very effectively reducing the passive damping roll to about 6^o at 5 m wave heights. For 10 m waves the angles of roll are about 12^o. This is not as good as the active elevon but the use of the cycloidal propeller has other advantages. It doesn't alter the width of the ship, it doesn't add inertia higher up in the ship reducing the metacentric height and most importantly it can be used at low speed when the dynamic fin is not effective.

Figure 6 illustrates the effect of changing the reaction time $\tau(= \tau_a + \tau_h)$ of the Voith Schneider propeller. If it only takes 3 seconds to achieve 67% of the maximum side thrust then the control is very effective reducing the roll angle to 12^o. When the reaction time rises to 8 seconds then the roll angle at a wave height of 10 m is around 27^o. When the reaction time rises to 13 seconds then the system cannot prevent capsize after 65 seconds.

This indicates the limits of the mechanical servo system and hydrodynamic forces required for a successful application of a cycloidal propeller to control roll in this particular ship subject to a severe stern quartering sea. Data was not available from the manufacturer to indicate whether this is possible. Voith Schneider themselves have proposed such a system[3, 32] showing a substantial reduction in roll angle for a naval vessel application. It does not appear to be such a severe case as that considered here.

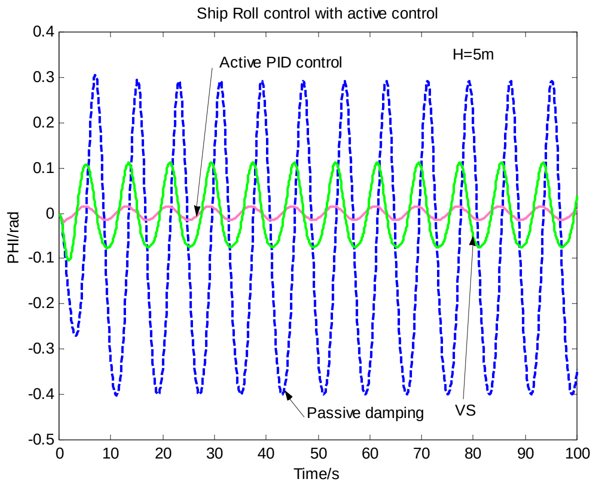

Fig. 4. Comparison of passive damping, elevon active control and Cycloidal propeller actuation for $H = 5$ m

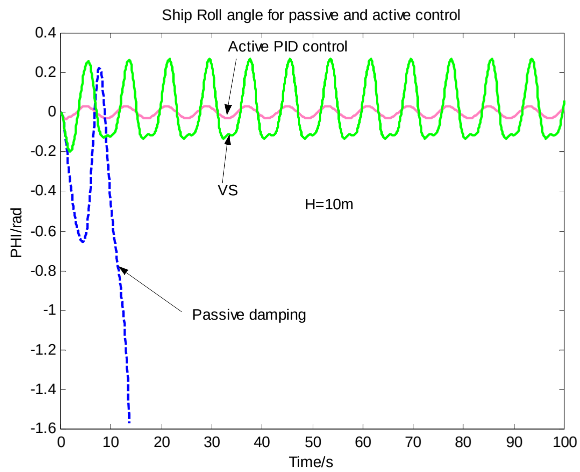

Fig. 5. Comparison of passive damping, elevon active control and Cycloidal propeller actuation for $H = 10$ m

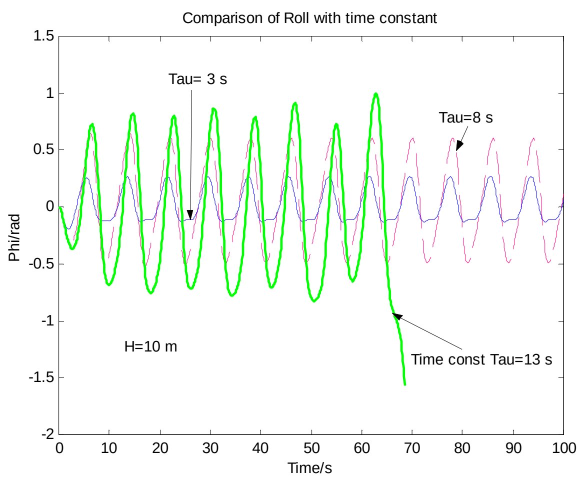

Fig. 6. Cycloidal propeller actuation for different time constants

6 Conclusions

The capsize of a small tanker has been modelled using strip theory for the hydrodynamic forces and Hydrodynamic moments. Static stability is represented as a softening spring. The original model of White et al. was used as this was a validated case of a real ship with comparison model basin tests and analogue computer simulation to confirm the general approach and illustrate the results of the non-stabilised system.

(1) A simple hydrofoil passive damping system would have stabilised roll motions up to at least 7 m in wave height.

(2) An active control system using full-span elevons on a fin stabiliser produces even greater reduction in roll angle and is still very effective at wave heights of 10 m.

(3) A second system is proposed using a Voith Schneider cycloidal propeller acting to produce lateral thrust

to control the roll.
(4) This system is shown to reduce the roll angle to a small enough amount if the time constant to produce thrust is as small as 3 seconds. However if this time constant reaches a value of 13 seconds then it would be incapable of preventing capsize in this stern quartering seas example.
(5) The fin/elevon, although better when the ship is at speed, does not prevent capsize at low speeds, below 5 knots.
(6) The Voith-Schneider propeller system has the main advantage that it works at low speeds and importantly when the ship is stationary. The installed machinery is overall probably less in weight for the VS system and the metacentric height would not be reduced.
(7) The cycloidal propulsion is better suited to larger ships for roll control since the inertias are larger and the roll frequencies likely to be lower.

Future work, as a matter of design, would entail a more exact representation of the mechanical elements necessary for the implementation of the drive systems on a given ship for the two different actuation mechanisms. Simulation with a more precise model of the hydrodynamics and investigation of other controllers are also desirable to confirm the situation described here. The simple PID controllers illustrated here show sufficient merit for further analysis.

7 Nomenclature

Roman

a	Lift curve slope;
A_{22}	Hydrodynamic mass in sway;
A_{24}	Hydrodynamic coefficient roll into sway;
A_{42}	Hydrodynamic 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_{41}	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;
$\varDelta L$	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;
τ_a Hydraulic actuator time constant;
τ_h Hydrodynamic force time constant.

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 aft	1.75 m
Trim aft	1.52 m
Metacentric Height	0.64 m

Table 3. GZ curve particulars

	$\mathrm{GZ}_{\max}$	$\mathrm{GM}_{\max}/\mathrm{deg}$	GM/m
IMCO	0.2	25	0.15
Ballast (capsize condition)	0.032	27	0.64
Loaded	0.29	46	0.65

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