NUMERICAL SIMULATION OF A FISHING VESSEL IN PARAMETRIC ROLL IN HEAD SEAS

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Abstract

A time-domain numerical simulation method is applied for the investigation of the parametric roll resonance of a fishing vessel sailing in regular head waves. The employed mathematical model captures rationally most of relevant problem parameters hence enables an improved understanding of the complicated non-linear phenomenon. The dynamic behavior of the vessel has been systematically analyzed and mapped to a characteristics pattern describing in a concise way the incurred stability problem. Two modes of possible capsize are identified, namely due to intensive parametric roll and due to loss of stability. Where possible, the numerical results are compared with corresponding available experimental measurements.

1. Introduction

The phenomenon of parametric roll has been extensively studied over the last decades, but it still remains of high interest due to its complicated, highly non-linear character.

Parametric roll is considered as the induced roll motion of a ship sailing in following or in head waves, despite the fact that there is not any oblique wave component to directly excite the roll motion.

The resulting roll is explained through the change of the underwater part of the vessel as it moves through the waves, and subsequently the change in GM and restoring moments. When the vessel is on the wave crest then restoring is generally reduced, whereas on a wave trough the restoring is increased.

Parametric roll is expected to occur when the wave frequency of encounter is close to the double of the natural roll frequency of the vessel. Given this condition, the other parameters of the problem eventually determine the occurrence as well as the intensity of the phenomenon. These are the vessel's forward speed, the hull form and related hydrostatic characteristics, the roll damping effects and the wave amplitude itself.

Several approaches have been employed to analyze and understand the parametric rolling ranging from the non-linear roll equation, as a single degree of freedom system with appropriate parameters, to models of multi degrees of freedom, where roll motion is coupled to the other motions. Parametric roll resonance has been investigated both in regular and irregular seaways as well as in following and head seas. A recent review of the related literature has been carried out in [1].

In this paper, a nonlinear time-domain numerical simulation method, which is implemented with the computer code CAPSIM [2], is applied to explore the multi-dimensional space defined by the above parameters. The method models rationally many of the involved aspects and enables the understanding of the complicated character of the occurring stability problem.

In this paper, a fishing vessel, which has been experimentally tested in model tank in [3], is modeled and numerically investigated. The motion of the vessel is systematically simulated in conditions of parametric roll resonance in regular head waves.

The obtained results permit the understanding the performance of the studied ship in parametric roll and provide an insight to the employed numerical method.

2. The Numerical Simulation Method

The numerical simulation method applied herein for the evaluation of extreme motions of ships in seaways is a time domain numerical method which is based on linear potential theory with respect to the basic hydrodynamics of the problem and considers a variety of non-linear terms like excitation by large amplitude regular of irregular waves, nonlinear body geometry effects, like the effect of the above calm waterline body shape and its impact on the ship's restoring, as well as possible sloshing effects due to internal to the vessel or trapped on deck moving fluids, [2]. The mathematical modeling of the method is implemented with the computer code CAPSIM. The method has been successfully applied in the past to the simulation of ship motions at zero forward speed in damaged and flooded conditions, as well as the estimation of wave loads and particularly of drift forces on floating bodies, [4], and proved very efficient and satisfactory in many practical cases.

The essential characteristics of the method are outlined in the following. The sailing ship is considered rigid and of arbitrary shape moving in six degrees of freedom in response to incident waves. The equations of motion of the body in the 3D space are derived by application of the momentum conservation theorem. The equations expressed in the ship-fixed coordinated system, which moves with respect to a coordinate system traveling with constant velocity Us are given below.

$$m\left(\dot{\vec{U}} + \vec{\omega} \wedge \vec{U}\right) = \vec{F} \tag{1}$$

$$[I]\dot{\vec{\omega}} + \vec{\omega} \wedge [I]\vec{\omega} = \vec{M} \tag{2}$$

where m and $[I]$ are the mass and the matrix of moments of inertia of the ship respectively, U and ω the linear and angular velocity respectively, and F, M are the external to the ship forces and moments.

The external to the body acting forces comprise gravity, hydrostatic and hydrodynamic components; others might be added in a straightforward way (wind & current forces, etc). Following potential theory, the wave effects, as the dominant dynamic part, are further analyzed into Froude-Krylov or undisturbed incident wave and diffraction effects.

Incident wave forces together with the hydrostatic ones are calculated through direct integration of the dynamic and hydrostatic pressure over the instantaneously wetted part of the hull $S_W(t)$, which is defined by the undisturbed incoming wave and the instant position of the ship.

$$\vec{F}_I(t) = \iint_{Sw(t)} p\vec{n}ds \tag{3}$$

where n is the unit normal vector and the pressure p comprises both hydrostatic and dynamic terms, according to the next formula.

$$p = -\rho g z - \rho\frac{\partial\Phi}{\partial t} - \frac{1}{2}\rho\left|\nabla\Phi^2\right| \tag{4}$$

where Φ is the velocity potential, ρ the water mass density, g the gravity acceleration, and z the vertical coordinate, directed upwards.

The radiation forces are herein calculated by use of the added mass and damping coefficients calculated in the frequency domain and properly transformed into the time domain by application of the impulse response function concept introduced by Cummins [5].

$$F_{R,i}(t) = -A_{ij}(\infty)\dot{U}_j - \int_0^{\infty} K_{ij}(\tau)U_j(t-\tau)d\tau\,,\ i,j=1\div 6 \tag{5}$$

where A_{ij} are the added masses at infinite frequency of oscillation and the kernel functions K_{ij} are the impulse response functions corresponding to the damping coefficients calculated in the frequency domain.

For irregular seaway excitation, the elementary diffraction forces, corresponding to the constituent wave frequencies of an assumed irregular seaway spectrum, are taken to be directly proportional to the corresponding elementary diffraction forces calculated in the frequency domain.

Radiation (thus added-mass and damping coefficients) and diffraction forces in the frequency domain were herein calculated by applying the computer code NEWDRIFT, [6]. It is a six-degrees of freedom, three-dimensional panel code program for the calculation of motions and wave induced loads, including drift force effects acting on arbitrarily shaped bodies in regular waves. The code is based on the zero-speed Green function, pulsating source distribution method and accounts for the forward speed effects considering the slender body theory assumptions. The code employs triangular or quadrilateral panels for the modeling of the wetted ship surface.

Viscous effects are taken into account for the roll motions using a semi-empirical linear or quadratic roll velocity model.

The above formulated mathematical model for the present ship-wave (hydro-mechanical) system comprises of a set of six second-order nonlinear differential equations. Integrating this set of equations by a time integration method, the six degrees of freedom motions of the body are obtained in the time domain.

3. Details of Numerical Experiments

The herein tested vessel is a fishing boat with a transom stern arrangement. The details of the vessel have been taken from [3] where it has been tested for the parametric roll in head waves in a model tank. Its main particulars are listed in Table 1 and the body plan in Figure 1.

Length (m)	25.91
Length between perpend. (m)	22.09
Beam (m)	6.86
Depth (m)	3.35
Draught (m)	2.48
Displacement (tons)	170.30

Table 1 Main particulars of the tested vessel

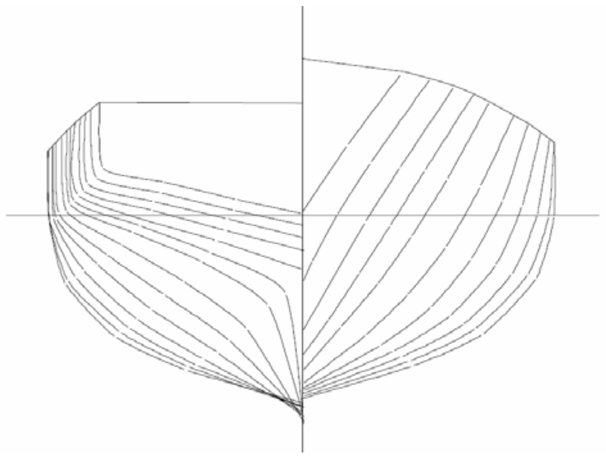

Figure 1 Body plan of tested vessel

The fishing vessel has been numerically modeled close to the experimental conditions applied during the tests in the model tank. However, some differences where accepted for the setup of numerical simulation.

The model in tank was equipped with two lines serving as course keeping arrangement whereas in the numerical simulation the model was studied in constrained surge, sway and yaw, hence omitting the course keeping problem.

Intending to test the model in parametric roll the natural roll frequency, respectively period, of the model was first numerically evaluated. Numerical values showed differences compared to the corresponding periods resulting from model testing. As shown in Figure 2, the natural periods where corrected to adjust the experimental data by decreasing the roll radius of gyration to 2.11 m with respect to the value used in tank tests (2.68 m).

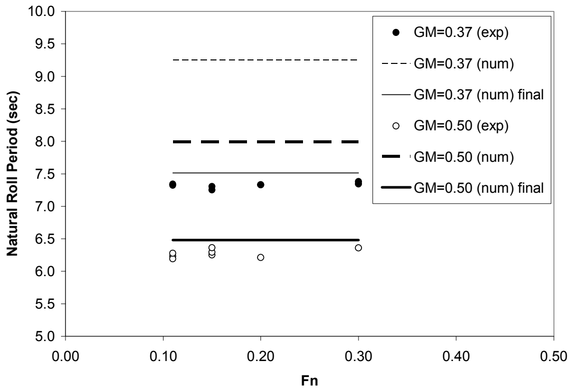

Figure 2 Natural roll period of the tested vessel

The observed systematic overestimation of natural period could be explained either by differences in inertia terms or in hydrostatics. As soon as the later was precluded by conduct of a detailed investigation, and as the hydrodynamic roll inertia is limited, the full difference was finally attributed to the structural roll inertia. As regards the pitch radius of gyration it was kept equal to the experiments 5.52 m.

In absence of roll damping data, the viscous roll damping is approximated with a linear model having a damping parameter varying from 3.0 to 10.0 % of the critical roll damping. In Figure 3 the free roll decay of the vessel for the two values of damping is depicted.

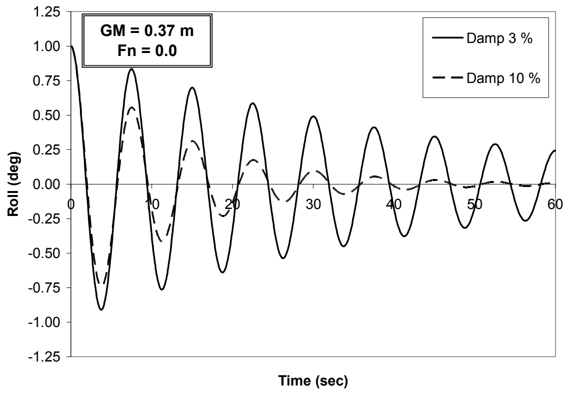

Figure 3 Free roll decay of the studied vessel

4. Results and Discussion

The fishing vessel was tested in parametric roll resonance conditions, namely for all the numerical tests the wave frequency of encounter was almost double the natural roll period of the vessel. For these conditions three forward speeds were considered which correspond to Froude numbers 0.15, 0.20 and 0.30, and two different GM values, 0.37 m and 0.50 m. Taking also into account the parametric variation of the roll damping, a total of 24 cases were investigated.

For each case the performance of the fishing vessel for varying wave amplitude was systematically recorded. The roll response of the vessel as a function of wave amplitude (and indirectly of the wave steepness) is plotted in Figure 5 and up to Figure 10 below.

In these diagrams the roll amplitude of the vessel corresponds to steady state conditions, namely when the transients have died out. Four curves are plotted in each diagram, with each curve corresponding to a different damping parameter. The simulated results are depicted with symbols connected by straight lines.

Reviewing the diagrams, it is evident that the parametric roll of the vessel occurs up to a certain wave amplitude for each case. The roll curve exhibits a hump at the left part of the diagrams. To the right side of these humps, there is always a flat region where no rolling occurs, whereas for higher wave amplitudes a systematic loss of stability and capsize was recorded.

Thus, two different modes of capsize were identified. The first was observed near the crest of the parametric roll hump for the cases with the lower GM=0.37 and the lower roll damping. These capsize results are shown in the left column diagrams (Figure 5 to Figure 7) as a jump out the plot area. The second mode of capsize is observed for the higher wave amplitudes, as evident in the right part of the plots, shown also as a jump out the plot area.

In Figure 4 the roll motion in the two capsize modes is presented. In the first mode the intensive parametric roll leads to capsize as the vessel progressively absorbs more and more energy which eventually leads to capsize. In the second mode of capsize the vessel obviously disposes poor stability characteristics and heels quickly to high angles with no return.

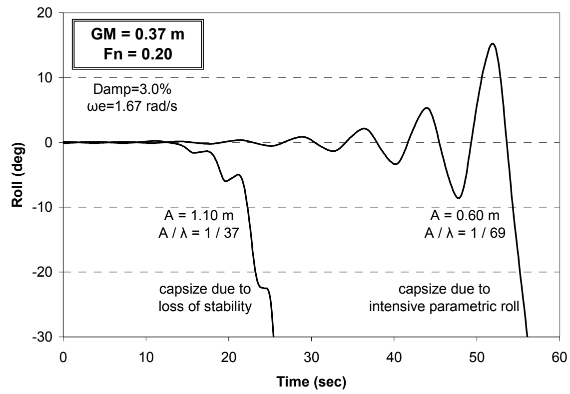

Figure 4 Capsize in different modes

The numerical results (Figure 5 to Figure 10) are plotted together with relevant measurements from model tests published in [3]. Although experimental data are not enough to support the overall numerical mapping, it is interesting to observe the vanishing of rolling at intermediate wave amplitudes, namely between the parametric roll hump and loss of stability areas, which is met both with the experimental and numerical results. In this range of wave amplitudes the vessel practically moves in heave/pitch without experiencing roll motion.

The effect of viscous damping on the performance of the vessel is always to decrease the final roll motion, whereas the characteristic pattern of the response is practically kept unchanged.

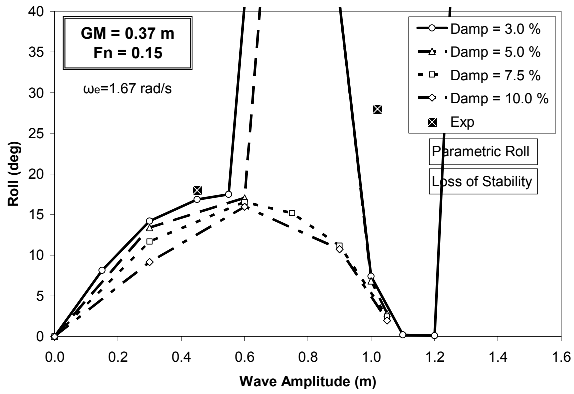

Figure 5 Roll amplitude vs wave amplitude for GM=0.37 and the Fn=0.15

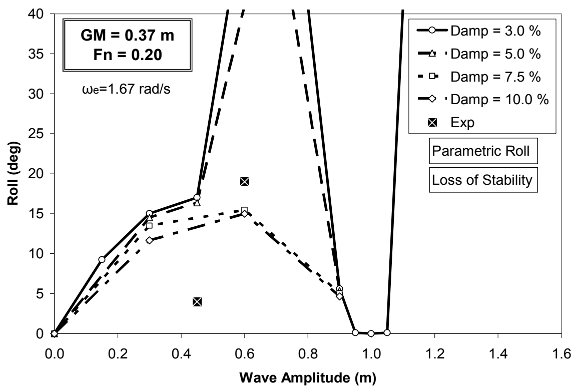

Figure 6 Roll amplitude vs wave amplitude for GM=0.37 and Fn=0.20

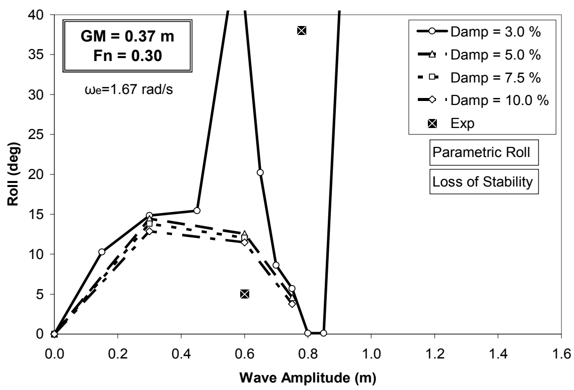

Figure 7 Roll amplitude vs wave amplitude for GM=0.37 and Fn=0.30

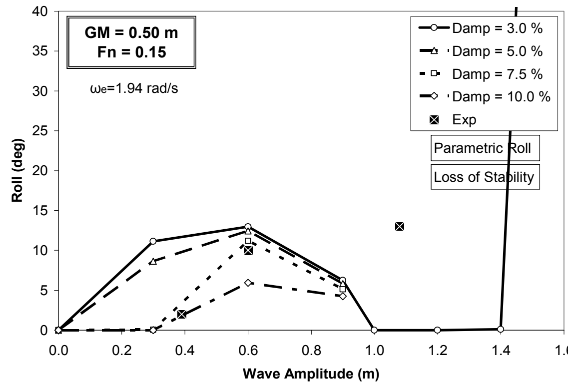

Figure 8 Roll amplitude vs wave amplitude for GM=0.50 and Fn=0.15

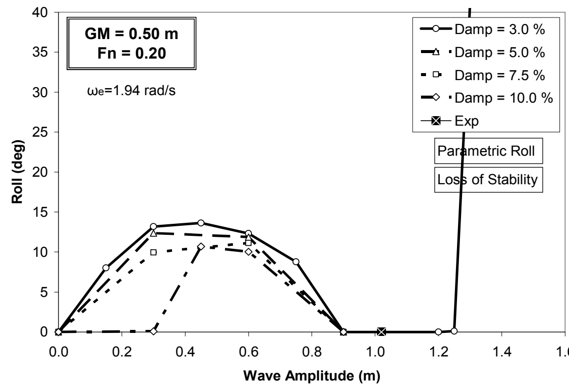

Figure 9 Roll amplitude vs wave amplitude for GM=0.50 and Fn=0.20

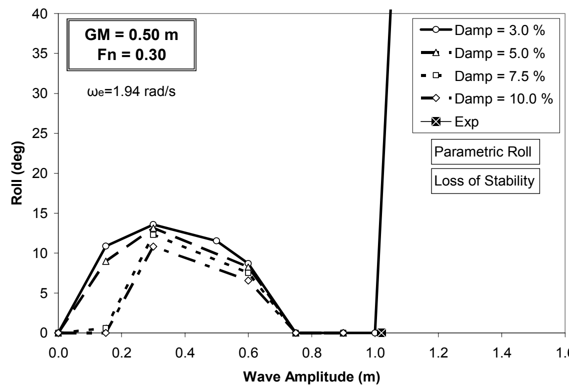

Figure 10 Roll amplitude vs wave amplitude for GM=0.50 and Fn=0.30

Some secondary observations from these diagrams are the effects of forward speed and metacentric height. With an increase of the forward speed the hump of parametric roll is shifted to the lower wave amplitudes and the region of loss of stability occurs for even lower waves. Hence, the vessel suffers less from parametric roll but more from loss of stability.

The effect of GM on the roll behavior is more drastic. When the GM is lower then both the hump of parametric roll and the loss of stability area become wider. Furthermore, when low GM is combined with weak damping effects, then capsize due to intensive parametric roll is more likely.

The other two motions of the vessel those of heave and pitch are presented in Figure 11 and Figure 12. The curves for the lower roll damping of 3% should be enough to show the dependence of these motions on the wave characteristics. Both motions are almost linear for the lower wave amplitudes whereas they reach a maximum response for large waves. The heave motion clearly depends on the forward speed, as heave increases in the speed increase. To the contrary pitch motion seems independent on the forward speed effects. It should be reminded that in these diagrams, to the contrary to the previous, the wave slope for a certain wave amplitude is a function of froude number.

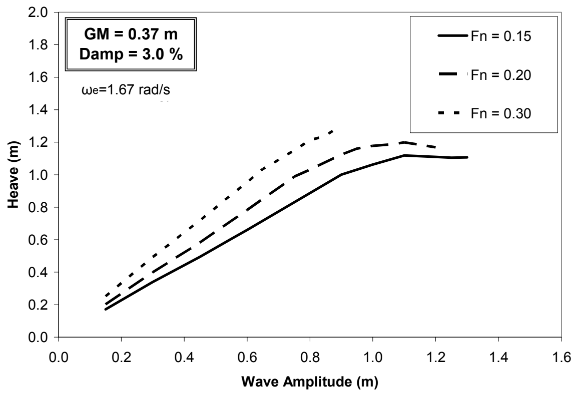

Figure 11 Heave motion vs wave amplitude

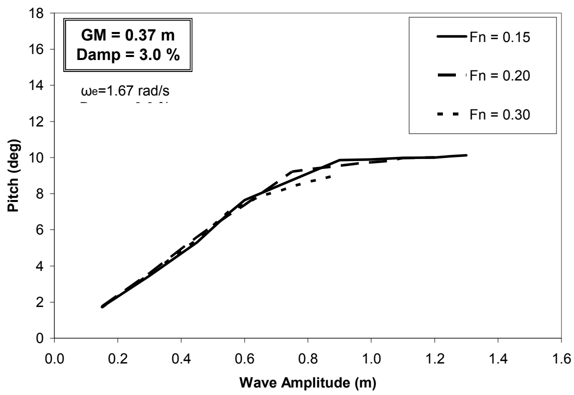

Figure 12 Pitch vs wave amplitude

The above presented behavior was also investigated with respect to a variation of the initial conditions. It was confirmed that as soon as a transient response, at the initial stages of simulation, died out the model eventually responded independently of the initial conditions. In Figure 13 two tests are shown that differ with respect to the initial roll angle. Both responses converge to the same curve at the steady state conditions, as shown at the right end of the diagram.

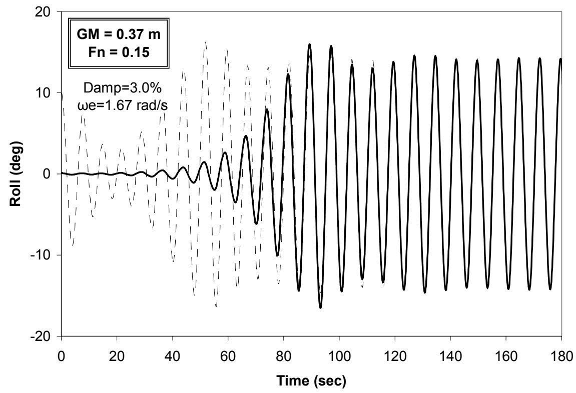

Figure 13 Roll response with different initial conditions

Another investigation for the sensitivity of results was conducted in terms of the encounter wave. The recorded roll response pattern in head waves was confirmed for varying frequencies of encounter in the range of ±4% of the resonance frequency.

The next Figure 14 presents schematically a generalization of the roll response of the vessel in parametric roll conditions in regular head waves.

The complicated relation of ship's roll motion with the wave amplitude in parametric roll appears to have the following pattern:

- ✓ At the range of lower wave amplitudes a parametric roll hump is formed. A hump crest can be identified unless combined low GM and damping generally lead to dynamic instability, thus to intensive parametric roll and possible capsize.
- ✓ For medium wave amplitudes the vessel is riding the waves without roll motion. The range of this pattern increases when the GM is high, and this performance is shifted towards lower wave amplitudes when the roll resonance occurs at higher vessel speeds.
- ✓ As the wave amplitude becomes even higher the vessel generally loses transverse stability and capsizes. This instability is favored in cases of low GM and higher speeds.

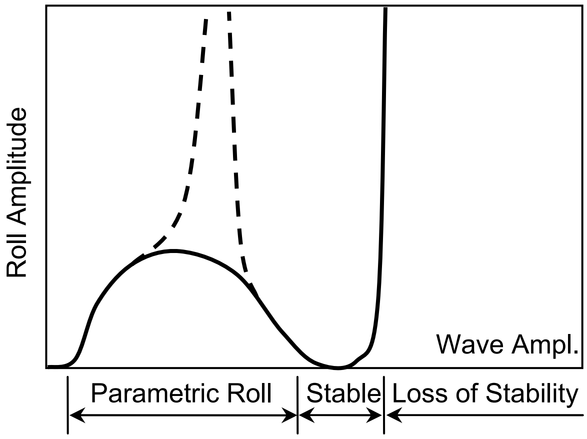

Figure 14 Generalized roll response

This generalized roll response in parametric resonance of Figure 14 has been repeatedly met in the numerical simulation results; however the available experimental data are not enough to verify this complex pattern. Nevertheless, some similarity to the roll pattern presented earlier in [7], which has been produced by use of a one-degree nonlinear roll equation model, can be identified.

5. Conclusions

The motion of a fishing vessel in parametric roll conditions in regular head waves was explored with the use of a nonlinear 6 DOF time domain numerical simulation method.

The application of the method produced quite interesting results and enabled the generation of a general response pattern for the roll motion in parametric resonance.

For these conditions two modes of dynamic instability and subsequent capsize were recorded. Capsize due to intensive parametric roll and due to loss of stability.

Findings need further investigation as only limited experimental data were available for the verification of this highly complex behavior.

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