Parametric Roll and Ship Design

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ABSTRACT

The variety of vessels which can suffer from parametric roll is large. It has been observed on small fishing vessels sailing in following waves but also on large cruise and container vessels in head and following waves. The occurrence of parametric roll, with high roll angles, is governed by a complex combination of main dimensions, loading condition, hull form, appendage configuration, speed and encountered wave conditions. In this paper the influence of main dimension and of variations on the fore and aft body on the occurrence of parametric roll are investigated. A one degree of freedom motion method and a non-linear time domain simulation method were used. The results were validated with model tests on a C11 container ship. Also, the influence of different roll damping devices on the occurrence of parametric roll is evaluated.

Keywords: *Parametric roll, Ship design, Container vessels, Extreme motions, Roll damping, Transverse stability*

1. INTRODUCTION

In 1998 a post-Panamax, C11 class containership lost 1/3 of her deck containers and damaged another 1/3 in a severe storm. The incident was analysed by means of numerical simulations and model tests. The results confirmed that the vessel suffered from a severe case of parametric roll during the storm (France et al., 2003). Since the publication of these results, ship operators and ship designers have become more aware of the fact that this phenomenon can occur for larger vessels in confused seas and not only for small vessels in regular waves which was thought for many years.

New designs of large vessels, in particular container vessels, are since then more and more checked on their tendency for parametric roll behaviour. Class societies acknowledged the problem and have started incorporating parametric roll in their guides (Shin et al., 2004). Still, model tests are considered as the way to assess the sensitivity for parametric roll and such tests are expensive in a design parameter space. Analytical tools which can predict the phenomenon are not easy to use, not always reliable or available. Therefore hull lines variations or optimisation with regard to parametric roll is seldom performed in an early stage of new designs.

In order to incorporate parametric roll in ship design an understanding of the phenomenon is required. A greater understanding is needed in how main dimensions, hull form changes and appendages configuration alter the probability of parametric roll.

In this paper results of a study on the effect of main dimension variations, hull form variations and different appendages configuration on the occurrence of parametric roll will be presented. In Chapter 2 the phenomenon is explained and a discussion of the minimal requirements for parametric roll to occur is given. In the third chapter a discussion will be given of how the main dimensions and the loading condition of a vessel can result in conditions where parametric roll can occur. Different graphs are presented showing the "critical" combinations of vessel length and loading condition. In Chapter 4 results of the effect of hull form variations on a C11 post-panamax container vessel on the occurrence of parametric roll are presented. Variation of bow flare and stern configuration are investigated. In the last Chapter the effect of different roll reduction devices such as bilge keels and active fin stabilizers is discussed.

2. BACKGROUND

2.1 Theory

The theory behind parametric roll and its consequence has been studied and described by many investigators; see for example Kempf (1938), Graff & Heckscher (1941), Paulling & Rosenberg (1959), Paulling (1961), Oakley et al. (1974), Dunwoody (1989a), Dunwoody (1989b), Dallinga et al. (1998), Luth & Dallinga (1998) or Francescutto & Bulian (2002). Therefore, in this paper only the principles of parametric roll will be described.

In "normal" sailing conditions the ship motions of a vessel are caused by direct wave excitation. Resonant roll motion often occurs in beam waves and stern quartering waves when the combination of wave period, vessel speed and heading leads to a wave encounter period close to the natural roll period of the vessel.

In pure head seas condition, the first order roll wave excitation is zero. Nevertheless, under certain conditions of encounter period, roll motion can be excited in head seas, via a different phenomenon. This phenomenon is referred to as "auto parametrically excited motion" which is usually shortened to "parametric motion" or "parametric roll". The term describes a state of motion that results not from direct excitation by a time-varying external force or moment but from the periodic variation of certain parameters of the oscillating system. The roll motion, once started, may grow to large amplitude, limited by roll damping and, in extreme conditions, may result in danger to the ship or its contents.

For a ship in head or stern seas the uneven wave surface together with the pitch-heave motion of the ship results in a time-varying underwater hull geometry. This varying geometry, in turn, results in time-varying changes in the metacentric height, i.e., in the static roll stability. The variation of the static roll stability can cause instability if it occurs in the appropriate period.

From theory and as validated by model tests (Dallinga et al., 1998, Luth & Dallinga, 1998 and France et al., 2003), parametric roll occurs when the following requirements are satisfied:

(1) The natural period of roll is equal to approximately twice the wave encounter period.
(2) The wavelength is in the order of the ship length (between 0.8 and 1.2 times LBP).
(3) The wave height (thus the GM variations) exceeds a critical value.
(4) The roll damping is low (lower threshold wave height).

2.2 Prediction of Parametric Roll

Nonlinear, time-domain seakeeping computer codes are able to predict the phenomenon of parametric roll (see below). These computations are however not easy to use or available for everyone. In a preliminary design stage a simple and fast method is desirable. For parametric roll Dunwoody (1989a and 1989b) proposed such a method, which is used in this paper. His method is based on a single degree of freedom motion equation for roll, using a time varying restoring coefficient. This motion equation is known as the Mathieu equation, which is presented first.

<u>Modelling in one degree of freedom</u> The equation for one degree of freedom roll motions is given below.

$$(I_{xx}+A_{xx})\frac{d^2\phi}{dt^2}+B_{xx}\frac{d\phi}{dt}+C_{xx}\phi=M_x \qquad (1)$$

In head seas condition the roll moment excitation will be zero, similar as in a roll-decay test. The roll motion equation reduce in these situations to:

$$\frac{d^2\phi}{dt^2}+2\upsilon\frac{d\phi}{dt}+\omega_\phi^2\phi=0 \qquad (2)$$

Where the roll period is defined by:

$$\omega_\phi=\sqrt{\frac{C_{xx}}{I_{xx}+A_{xx}(\omega_\phi)}}=\sqrt{\frac{\rho g\nabla GM}{I_{xx}+A_{xx}(\omega_\phi)}} \qquad (3)$$

And where the damping ratio is defined as:

$$\upsilon=\frac{B_{xx}(\omega_\phi)}{2(I_{xx}+A_{xx}(\omega_\phi))} \qquad (4)$$

The restoring moment or static stability GM of the ship when sailing in waves will vary in time and the variation is a function of the actual wetted surface contour and thus depends on the hull lines around the calm water line. The largest variations in restoring occurs when the ship has large pitch motions, so when it sails in a wave length equal to about the ship length. A formula for the time varying restoring force, for the upright ship, in regular waves is given by:

$$C_{xx}(t)=\rho g\nabla\left(GM_m+GM_a\cos(\omega t)\right) \qquad (5)$$

Where GM_m is the mean GM, that is the GM in calm water GM_a is the amplitude of the GM variation and ω is the wave frequency. When equation (5) is used in equation (1) a one-dimensional Mathieu equation is obtained; a linear second order differential equation with periodic coefficients. The damped Mathieu equation is now written as:

$$\frac{d^2\phi}{dt^2}+2\upsilon\frac{d\phi}{dt}+(\omega_\phi^2+a\cos(\omega t))\phi=0 \qquad (6)$$

Parametric resonance conditions for this equation can be found when:

$$\frac{\omega_\phi}{2\pi}=\frac{\omega}{2\pi}\frac{1}{2}n \qquad (7)$$

According to Francescutto (2002) the threshold value for parametric roll in regular waves is:

$$\frac{\delta GM}{GM}=\frac{4\upsilon}{\omega_n} \qquad (8)$$

This threshold has been used and compared with simulations and model test. The results are presented in section 4.4. Furthermore in this paper a one degree of freedom based method proposed by Dunwoody (1989a and 1989b) will be used. The method is shortly described below.

Response to GM fluctuations (Dunwoody, 1989). The method proposed by Dunwoody is based on the following assumptions:

1. The roll motion can be expressed by the differential equation for a single degree of freedom motion method with parametric excitation of the stiffness.
2. That there is a linear relation between roll stiffness excitation (GM fluctuations) and wave height.
3. That a relation can be made between the spectrum of the stiffness fluctuations with the incident wave height spectrum and the speed of the ship.
4. That GM fluctuations produce an effect analogous to a reduction in the roll damping.
5. If the reduction in roll damping is larger than the roll damping from the hull and appendages an unstable situation occurs and the vessel is subject to parametric roll.

Furthermore, Dunwoody demonstrates that for a wide band random process the spectral density can be estimated for a frequency of twice the natural roll frequency.

Given the assumptions above the spectral density of the GM fluctuations can be expressed as the product of the spectral density of the wave encounter spectrum (at twice the natural roll frequency) and the square of the transfer function of the GM fluctuations (at twice the natural roll frequency). The equation is given below.

$$S_{GM} = \left(\frac{GM_a(\omega_\phi)}{\zeta_a} \right)^2 S_{e\zeta}(\omega_\phi) \tag{9}$$

According to Dunwoody the non dimensional damping reduction follows from:

$$\Delta\xi = \frac{\pi g^2 S_{GM}}{4 \omega_\phi^3 k_{xx}^4} \tag{10}$$

Parametric roll will occur if the non dimensional roll damping reduction exceeds the total roll damping B_{total} (made dimensionless by dividing with the critical roll damping B_{crit}). See equation (11).

$$\begin{aligned} &\frac{B_{total}}{B_{crit}} - \Delta\xi \le 0 \\ &B_{crit} = 2\sqrt{(A_{xx} + I_{xx})\rho g \nabla GM} \end{aligned} \tag{11}$$

The threshold wave height (the wave height for which parametric roll will start) can be determined by varying the wave spectrum in equation (9) and determining when equation (11) becomes zero.

Nonlinear, time domain seakeeping code PRETTI The development of a 3D panel code for seakeeping motion prediction has been a point of interest for the Co-operative Research Ships (CRS) since many years. In this joint-industry project a large group of different companies, such as class societies, ship-yards, ship operators , navies, and research/ engineering companies are actively involved in research related to many aspects in the design and operation of ships.

In recent years the CRS has developed a time-domain seakeeping code (Pretti) based on the hydrodynamics as calculated in the frequency domain by a 3D panel code (Precal). The current time-domain code incorporates non-linear excitation by pressure integration over the actual wetted surface. Diffraction forces are considered linear. Hydrodynamic coefficients and oscillatory (manoeuvring) derivatives are specific to sinusoidal motions and in a general theoretical model of a ship manoeuvring in a seaway, the ship motion cannot be considered simply sinusoidal. The motion equation must account for transient and random motions. This problem was initially discussed by Cummins (1962) and this approach of impulse response functions is adopted in Pretti.

The behaviour or a ship travelling in a seaway ‘integrates’ in practice two areas which are traditionally studied as separate problems and have led to two different mathematical approaches: a seakeeping theory assuming small motion amplitudes and the theory of manoeuvring assuming calm water and thus frequency independent hydrodynamics. In Pretti the two theories are combined as discussed by Bailey et al. (1997) or Fossen & Smogeli (2004). This is a challenging area of research since there is an overlap between the two models which require a careful implementation of manoeuvring coefficients. Ideally the aim is that with vanishing wave height the manoeuvring capabilities of the ship are found, and that the seakeeping hydrodynamics are captured in moderate wave conditions.

In large wave conditions with large amplitude motions the assumptions behind both the seakeeping and manoeuvring theory are violated since large variations in wetted surface are not accounted for when the basic coefficients in the models are calculated. Model tests are an essential guidance for the user of non-linear time domain simulation tools to gain experience in the use of a unified model.

The current non-linear time domain code of CRS focuses on course-keeping of the ship in 6 degrees of freedom (6 DOF), in which the interaction with the manoeuvring model is already essential. Especially the sway hydrodynamics will influence roll motions and this means that seakeeping (roll damping) and manoeuvring is to be combined. Yaw manoeuvring forces and a PID controlled rudder are furthermore essential to keep the ship on track and course.

3. INFLUENCE OF MAIN DIMENSIONS ON PARAMETRIC ROLL

In the theoretical background four criteria for parametric roll to occur were given. The first two (the natural period of roll is equal to approximately twice the wave encounter period and the wavelength is on the order of the ship length) can be described with the following equations:

$$T_\phi \approx 2T_{pe} \tag{12}$$

$$\lambda \approx L_{pp} \tag{13}$$

Equation (14) gives an approximation for the natural roll period (assuming 10 % added mass). Equation (15) gives a relation between wave length (λ) and wave period for deep water.

$$T_\phi = \frac{2.2\pi k_{xx}}{\sqrt{g\,GM}} \tag{14}$$

$$T_p = \sqrt{\frac{2\pi\lambda}{g}} \tag{15}$$

For zero speed equations (14) and (15) can be substituted in (12). GM can then be expressed in the following way (for zero speed).

$$GM = 0.605\pi\frac{k_{xx}^2}{\lambda} \tag{16}$$

For non zero speed the following relation between peak wave period and peak encounter wave period should be used (V_s being the ship speed and μ the wave direction, 180 being head waves).

$$T_{pe} = \frac{T_p}{1-\dfrac{2\pi V_s\cos\mu}{T_p\,g}} \tag{17}$$

Equation (16) gives a relation between GM, radius of gyration and wave length. By substituting also the second criterion for parametric roll (see equation (13)) the relation becomes purely dependent of geometry and main dimensions.

When the radius of gyration is not known, standard values for k_{xx}/B can be used. By also using standard values for L/B, equation (16) can be rewritten in the following way.

$$GM = 0.605\pi\frac{L^2}{\lambda}\left(\frac{k_{xx}}{L}\right)^2 \tag{18}$$

In the table below some values for k_{xx}/L are given for a range of L/B and k_{xx}/B values.

k_{xx}/L for a range of L/B and k_{xx}/B values					
	L/B				
k_{xx} /B	5.6	5.8	6	6.2	6.4
0.38	0.0679	0.0655	0.0633	0.0613	0.0594
0.39	0.0696	0.0672	0.0650	0.0629	0.0609
0.40	0.0714	0.0690	0.0667	0.0645	0.0625
0.41	0.0732	0.0707	0.0683	0.0661	0.0641
0.42	0.0750	0.0724	0.0700	0.0677	0.0656

Using equation (18) and the above table the two graphs (one for zero speed and one for 10 knots) given below can be made. They show the combinations of GM and Lpp for which parametric roll instability might occur. In other words where the first two criteria for parametric roll to occur are met.

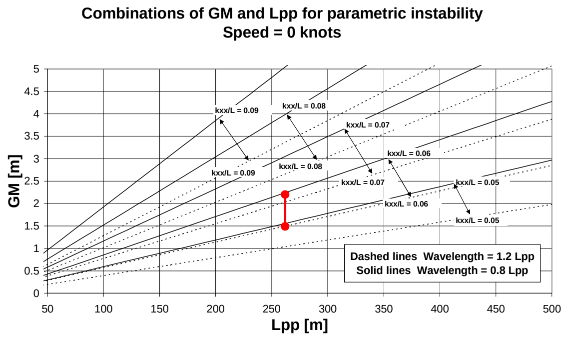


Figure 1 Combinations of GM and vessel length resulting in parametric instability (0 knots)

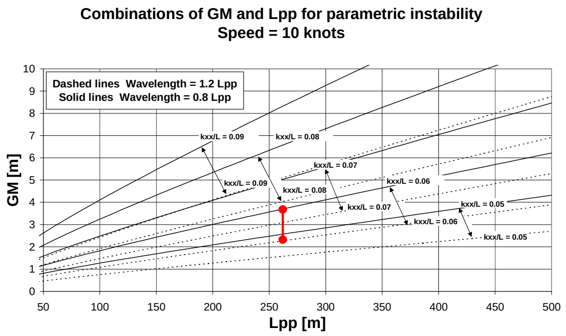


Figure 2 Combinations of GM and vessel length resulting in parametric instability (10 knots)

The figures show the relation between GM and Lpp for an upper limit of wavelength/Lpp = 1.2 (dashed lines) and a low limit of wavelength/Lpp = 0.8 (solid lines). For a given k_{xx}/Lpp ratio the area between the dashed and solid lines give the GM range for a certain Lpp for which the criteria are met.

With these graphs it is easy to investigate in an early stage if the vessel can be subject to parametric roll. To further illustrate how to use these graphs an example is given below.

C11 class container vessel:

Lpp	=	262	m
B	=	40	m
k_{xx}/B	=	0.40	

Thus: $k_{xx}/L = 0.061$

In the figures the range of GM's which will give parametric instability is indicated. At zero speed the range will be between GM = 1.5 m and GM = 2.2 m and at 10 knots speed between GM = 2.2 m and GM = 3.8 m. This doesn't mean the vessel will actually be subject to parametric roll. This depends on other factors like the amount of parametric excitation (GM fluctuations in waves) and the amount of roll damping. These factors depend on the hull form and the appendages.

4. HULL FORM VARIATIONS

4.1 Selected Case

In order to investigate the influence of the hull form on the occurrence of parametric roll a hull form needed to be selected. As a case the post-Panamax, C11 containership which encountered the storm, as described in the introduction, is used. It is a logical starting point since model tests were previously performed on the same vessel at MARIN.

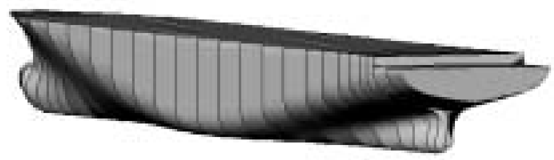


Figure 3 C11 Hull Form

A rendering of the C11 hull form is presented in Figure 3. Typical of post-PANAMAX containerships, the C11 has extensive bow and stern flare. The ship's natural roll period as used in this study was 25.7 seconds. This corresponds to the estimated roll period of the ship at the time of the incident. The GM in calm water was about 2 m.

The original hull form and variations at the bow and aft of the hull form were investigated. In the aft a Pram type hull form and a hull form with higher dead rise were taken. At the bow hull lines with a less pronounced bow flare (55 deg) and a more pronounced bow flare (45 deg) were used. This resulted in four variations of the original hull form.

In order to have a fair comparison between the different variations it is necessary to keep the draft, GM and natural roll period the same. Because of the different under water hull forms it means that the displacement and KG will change slightly.

4.2 Model Tests

In order to have validation material model tests were performed. These were performed in the Seakeeping and Manoeuvring Basin (SMB) at MARIN. The basin measures 170 x 40 x 5 m in length, width and depth respectively. Not all hull form variations were tested. The original hull form (model 8004-1) and the pram aft body (model 8004-2) variation were tested.

The models were self propelled during the tests and completely free sailing. The only connection between the model and the carriage consisted of an umbilical for power and data transmission.

The model tests were performed for an average speed in waves of 5 knots. The tests were done for head sea conditions and a variation of wave heights and wave periods were tested. For each model the tests started with a test in high waves (Hs = 7.5 m) for which several wave periods were tested. This was done to determine the most critical wave period. For this critical wave period a series of tests with increasing wave heights were done. For all these tests the realisation of the encountered waves was kept the same. Only the height of the waves was increased. The same realisation of the waves was used for both the models. This means that wave group effect do not influence the comparison of the two vessels. From the model test results it is possible to determine the wave heights for which parametric roll will start (threshold wave height).

Figure 4 Original C11 (upper) and Pram type (lower) aft ship

4.3 Model Tests Results

In Figure 5 the roll damping of the two hull forms (determined from roll decay tests) is compared. One can see that the roll damping of the pram aft shape is slightly higher, which is according to expectations. Both models were equipped with 40 cm high 76.54 m long bilge keels.

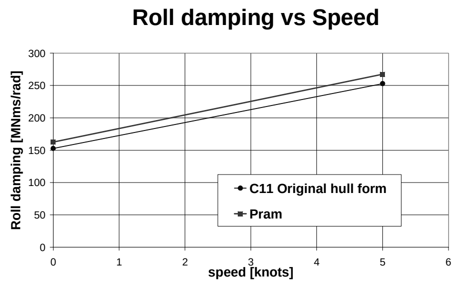


Figure 5 Roll damping comparison

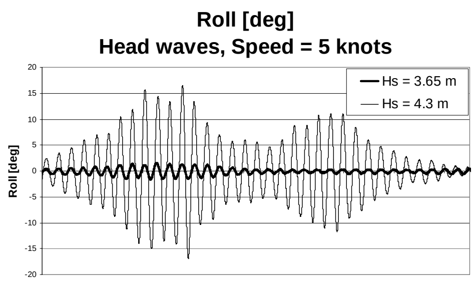


Figure 6 Time trace roll motion of C11 Original hull form

In Figure 6 a sample of the roll motion time traces are shown for the C11 original hull form in head seas at 5 knots speed. Results for two wave heights are given. They show the roll time trace for the vessel within the same wave realisation (same wave but only different amplitude). As can be seen the difference in wave height is small (Hs = 4.3 m versus Hs = 3.65 m) the difference in roll response is however big. The figure illustrates quite well the threshold behaviour of parametric roll.

The model tests results are summarized in Figure 7. The figure shows the mean of the 1/10 highest roll motions (A1/10+) as function of the wave height. Each dot in the figure represents a test in irregular head seas.

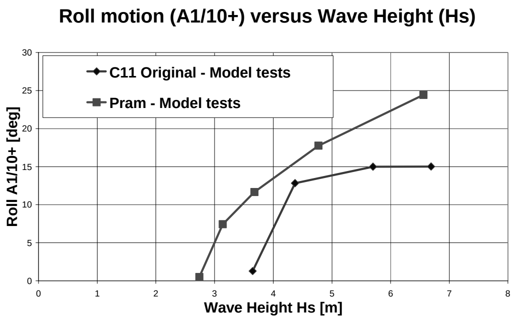


Figure 7 Model tests results

The model tests results show that the pram hull form results in higher roll angles. It also shows that the roll motions due to parametric roll start earlier meaning that the threshold wave height is lower.

From Figure 5 it could be seen that the damping of the pram is slightly higher than the original v-shape. The results presented in Figure 7 however show that the roll response is higher. It means that the excitation (GM variations) of the pram hull form is higher which is also according to expectations. The difference in GM variations will be discussed in the next section.

Using a criterion for parametric roll of 10 deg A1/10 one can determine the threshold wave height from Figure 7. For the C11 original hull form the threshold wave height becomes Hs = 4.3 m and for the pram Hs = 3.5 m.

4.4 Approximation Methods

The first step of the calculations was to validate the different approximation methods. If a good enough agreement was found with the model test the second step was to perform calculations for the other hull form variations which were not tested. Calculations were performed using a simplified one degree of freedom method (Dunwoody, 1989a and 1989b) and a non linear time domain program PRETTI (see also section 2.2).

GM variations As discussed earlier (see section 2.2) the GM variations in head waves represent the excitation for parametric roll. The static stability in "frozen" longitudinal waves is a good indication of the GM variations of a vessel sailing in waves. These were calculated for the original C11 hull form and the different hull form variations. For this purpose the program SHCP (NSSC, 2003) was used. The input for the program are the hull form, loading condition, wave height, wave length and the longitudinal position of the wave crest with respect to the hull. A range of wave conditions and roll angles can be entered. For each condition the pitch-heave static equilibrium is solved (thus preserving equilibrium of weight, buoyancy and trim moments). The righting arms calculated for each condition can be used to determine GM and thus the GM difference between the sagging and hogging conditions. Using this method one assumes that the dynamic pitch and heave motion do not influence the GM variations. Sample results are given in the figure below.

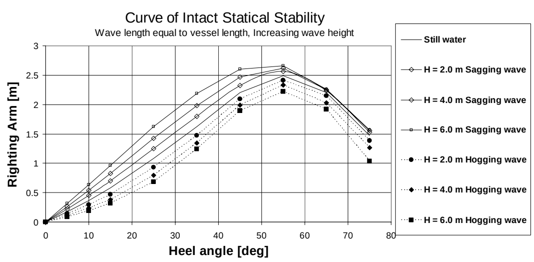


Figure 8 Curve of static stability C11 class container vessel

From Figure 8 the difference in GM between the sagging in hogging conditions can be determined. This can be done for different wave lengths. In Figure 9 the GM variations for a range of wave lengths are given for a wave height of H = 5.0 m. The figure shows the GM in sagging condition, hogging condition, the difference between the two (Delta GM) and the difference divided by the wave height (Delta GM/H) which is plotted on the second y-axis (right).

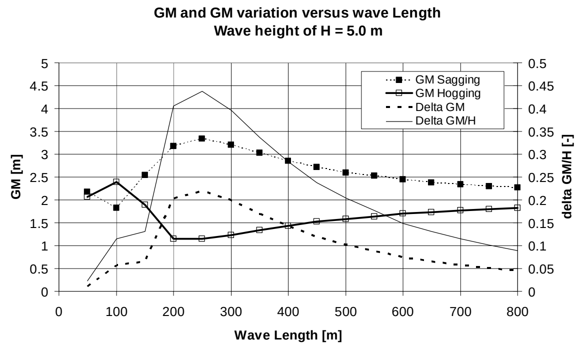


Figure 9 GM and GM variations versus wave length for C11 class container vessel

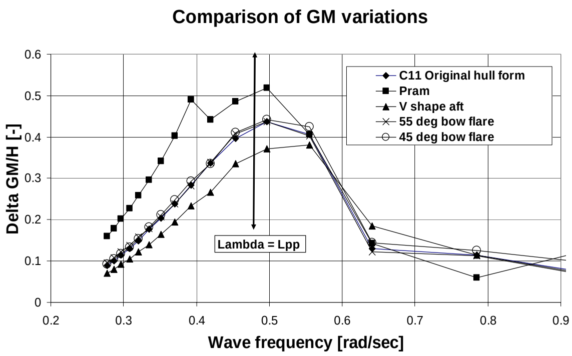


Figure 10 Influence of hull shape on GM variations

In Figure 10 the linearised GM variations (DeltaGM/H) are given as function of wave frequency for the C11 hull form and the four hull form variations. The frequency corresponding to a wave length equal to the length of the vessel is indicated in the figure.

The following observation can be made from the results presented in Figure10. First it can be observed that the changes to the bow flare do not seem to influence the GM variations. Secondly, the V shape aft hull form shows smaller GM variations than the original C11 hull form and the pram aft shape shows larger GM variations as the C11 hull form. The latter is in accordance with the model tests results presented in Section 4.3.

Threshold value for parametric roll in regular waves Using equation (8) an estimate of the required GM fluctuations in regular waves can be made. This was done using the following data at 5 knots forward speed: GM = 2.0 m; B_{xx} = 2.5 10^5 Nms/rad, I_{xx} = 2.12 10^7 kNms^2; A_{xx}= 3.0 10^5 kNms^2, which leads to $\delta GM = 0.16$ m in regular waves of 12.8 s. This is the threshold value at zero roll amplitude. Model tests in regular were not performed with the C11, so that these values can not be verified. However, compared to the observations in irregular seas, the GM fluctuations are about a factor 10 larger before the C11 shows a steep increase in roll.

Response to GM fluctuations (Dunwoody) The results presented in Figure 10 were used in Dunwoody's method (see formula 7, 8 and 9). For the damping the results from roll decay tests (see Figure 5) were used. It was assumed that the different hull form variations had the same roll damping as the original C11 hull form. For each hull form variation the threshold wave height was determined for a speed of 5 knots and a wave period of T_p = 14 s. The results are given in Table 1.

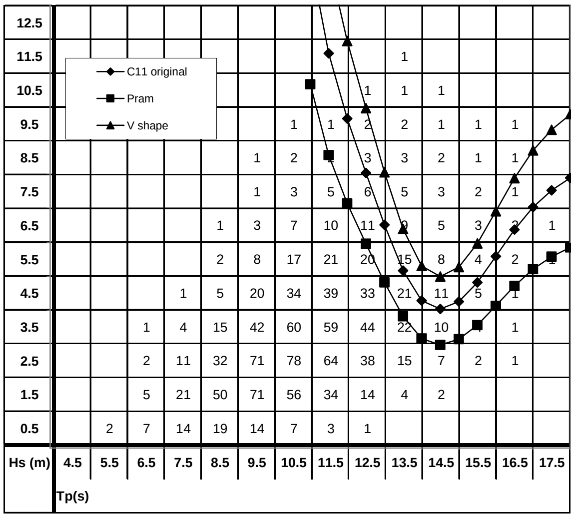
Table 1 Threshold wave height for different hull variations

	Threshold wave height	
	Dunwoody	Model tests
C11 original	4.3 m	4.3 m
Pram aft	3.2 m	3.5 m
V shape aft	5.3 m	-
Large bow flare (45 deg)	4.3 m	-
Small bow flare (55 deg)	4.3 m	-

The results using Dunwoody's method show very good comparison with the model tests performed. Furthermore the results from Dunwoody's method confirm the results found from the GM variation calculation. The bow flare does not seem to have any influence on the threshold wave height. The aft shape of the vessel has some influence. A very pronounced V shape aft gives the best performance with regard to parametric roll.

It must be mentioned that although the bow flare does not directly influence the occurrence of parametric roll it can indirectly influence it. A more pronounced bow flare will result in more slamming events which in turn will make a vessel master decide to reduce the speed earlier than a vessel with a less pronounced bow flare. Because the roll damping at low speed is smaller it is generally more vulnerable to parametric roll than at high speed. So, indirectly the bow flare can have influence on the occurrence of parametric roll.

With Dunwoody's method it is very easy to perform variations in wave conditions. The threshold wave height can then be determined for a variation of wave periods. By combining these lines with a wave scatter diagram it is then possible to evaluate the effect of the hull form on the probability of occurrence of parametric roll. This is shown in Figure11.



Hs (m)	4.5	5.5	6.5	7.5	8.5	9.5	10.5	11.5	12.5	13.5	14.5	15.5	16.5	17.5
12.5														
11.5										1				
10.5									1	1	1			
9.5							1	1	2	2	1	1	1	
8.5						1	2	2	3	3	2	1	1	
7.5						1	3	5	6	5	3	2	1	
6.5					1	3	7	10	11	9	5	3	2	1
5.5					2	8	17	21	20	15	8	4	2	1
4.5				1	5	20	34	39	33	21	11	5	1	
3.5			1	4	15	42	60	59	44	22	10	4	1	
2.5			2	11	32	71	78	64	38	15	7	2	1	
1.5			5	21	50	71	56	34	14	4	2			
0.5		2	7	14	19	14	7	3	1					
	Tp(s)													

Figure 11 Threshold wave heights within North Atlantic scatter diagram

In Figure 11 the limiting wave height is shown for the original C11 hull form and the two aft shape variations (pram and sharp V). The probability of each combination of H_s and T_p is given in thousands. The wave scatter diagram used was area 9 (North Atlantic) all directions from the Global Wave Statistics (Hogben et al., 1986).

From Figure 11 it can be determined that the V shape aft hull form has a 10% smaller number of occurrences of parametric roll (at 5 knots speed in head waves) in the North Atlantic. The results presented in the figure can not be used to estimate the exact number of occurrences of parametric roll for a given ship. In order to make that estimation the probability of the speed, wave heading and loading condition need to be incorporated. A method to determine this was presented by the author in 2003 (Levadou & Palazzi, 2003).

4.5 Simulation Methods

The non-linear time domain simulations have been performed in 5DOF (sway, heave, roll, pitch, yaw) for the lower sea states and in 3 DOF (heave, roll, pitch) for the more extreme sea states. It was verified that the results for intermediate sea states were comparable. The reason to limit the number of DOF in large sea states is a practical one: the Pretti code does not account for large yaw variations. Besides, in larger sea states the numerical model is more difficult to control, in particular due to the relative low forward speed which makes the rudder rather inefficient. Future enhancements are expected to overcome these limitations. Surge degree of freedom was neglected as well, for similar practical reason. The surge balance requires a dedicated implementation of added resistance, ship resistance and propulsion plant control, which is currently in development.

The viscous roll damping in Pretti is based on a time-domain implementation of Ikeda's method. For the bilge keel damping the water velocities at the bilge keel are assessed at each time instance and with a Keulegan-Carpenter number the drag on the bilge keels are calculated. Lift en eddy damping follows Ikeda's empirics.

All simulations were performed in the same 'relative' sea state realisation. This means that the wave component phase and relative amplitudes were kept identical when the wave height was varied. To obtain reliable statistics the simulations were performed for a duration of 3 hours. The computer time required (CPU) is about half the simulation time. This means that a parametric study as presented in this paper is feasible in a design stage, although significant computer CPU is required for several days.

A mesh of the C11 container ship as used in the simulations is presented in Figure 12.

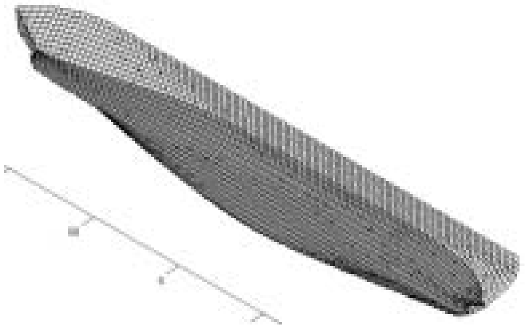


Figure 12 C11 container ship mesh for time-domain simulations

Model tests were carried out for two different hull shapes, the original C11 container ship and a modified one with the pram aft-body, denoted SIM Pram aft in Figure 13. The significant roll amplitude compares very well between simulations and model tests for the lower sea states. In the higher sea states the simulations predict larger parametric roll motions than the model tests show. The trend is however well predicted and the significant wave height at which parametric roll starts agrees well. The pram aft body might be beneficial for calm water resistance, but makes the design more sensitive for parametric roll. Striking is the fact that the original design shows less parametric roll in the higher sea states than the modified design, while this is opposite in the simulations. We consider this an effect of the not fully developed numerical models and the assumptions made in PRETTI. But, most likely non-linear hydrodynamics or more sophisticated non-linear (viscous) damping is required.

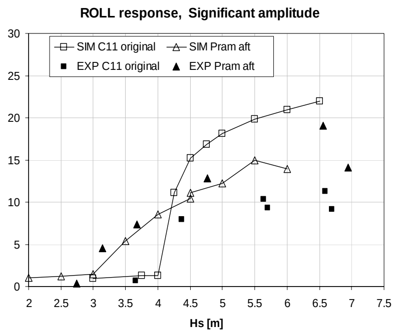


Figure 13 Significant roll amplitude from model experiments (EXP) and from non-linear simulations (SIM) for the C11 and Pram modified hull.

Following these results three other hull shape variations were investigated. The results are summarised in the figure below.

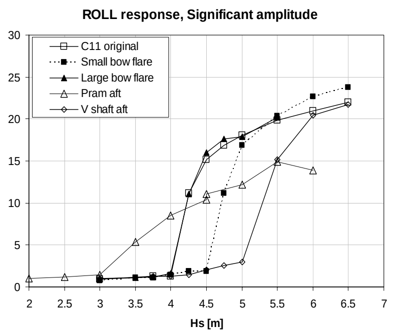


Figure 14 Significant roll amplitude in Jonswap wave spectra for the C11 and 4 modified hull shapes.

The numerical results are summarised in the following table 2, which compares the threshold value with Dunwoody. A threshold significant roll amplitude of about 5 degrees was used.

Table 2 Comparison Dunwoody and Pretti simulations

	Threshold wave height	
	Dunwoody	Simulations
C11 original	4.3 m	4.2 m
Pram aft	3.2 m	3.5 m
V shape aft	5.3 m	5.1 m
Large bow flare	4.3 m	4.2 m
Small bow flare	4.3 m	4.6 m

The comparison between Dunwoody and direct simulations is good, which is a striking conclusion. The dynamic GM variations in the simulations are comparable but too some extent different from the GM variations in an assumed static wave due to the fact that the equilibrium buoyancy condition in waves is not obtained; heave and pitch motions will change the buoyancy of the ship. The numerical simulations show a stronger influence of the bow flare than the static simulations, which could be due to the mentioned reason; the position of the ship in waves will be different in simulations than in a static approach.

5. ROLL STABILISATION

In the previous chapter the influence of the hull form on the parametric roll threshold wave height has been demonstrated. One other way to influence the threshold wave height is to change the roll damping of the vessel. Usually vessels have a small potential damping. Therefore, adding appendages (bilge keels, fin stabilizers) or anti roll tanks can increase the roll damping drastically. The influence of bilge keels and active fin stabilizers on the threshold wave height have been investigated by using Dunwoody's method.

5.1 Bilge Keel

The bilge keel damping is often associated with the energy dissipated by the drag forces of the bilge keel (Dallinga et al., 1998). Within this concept the damping is proportional with the roll velocity amplitude. Equation (17) gives the increase of roll damping per roll velocity amplitude change.

$$\frac{\partial B_{BK}}{\partial \dot{\phi}} = \frac{4}{3\pi} \rho \, 2 \, l_{BK} \; h_{BK} \; (C_{HF} \; r_{BK})^2 \, r_{BK} \; C_{Dbk} \qquad (17)$$

According to Ridjanovic (1962) the effective drag coefficient depends on the amplitude of the transverse flow and the bilge keel height (see equation 18).

$$C_{Dbk} = 22.5 \frac{h_{BK}}{\pi \, r_{BK} \, C_{HF} \, \phi} + 2.4 \qquad (18)$$

Using these equations the bilge keel damping was calculated for several bilge keel heights. The contribution as a fraction of the total damping linearised for 10 deg roll amplitude is given in the first row of table 3 for 5 knot speed.

Table 3 Roll damping contribution of bilge keel

	Bilge keel height			
	0 cm	20 cm	40 cm	60 cm
Fraction	0	0.256	0.428	0.547
Total Damping [MNms/rad]	145	195	253	320

As can be seen the roll damping contribution of the bilge keel is very large. Using these fractions on the total roll damping determined from the roll decay tests the damping for a variation of bilge keels can be determined. These values are indicated in the second row of the table.

Using Dunwoody's method the threshold wave height was determined for the different bilge keel heights. The results are given in. As can be seen a 40 cm bilge keel raises the threshold wave height by Hs = 1 m. Adding 20 cm raises the threshold by about 0.6 m.

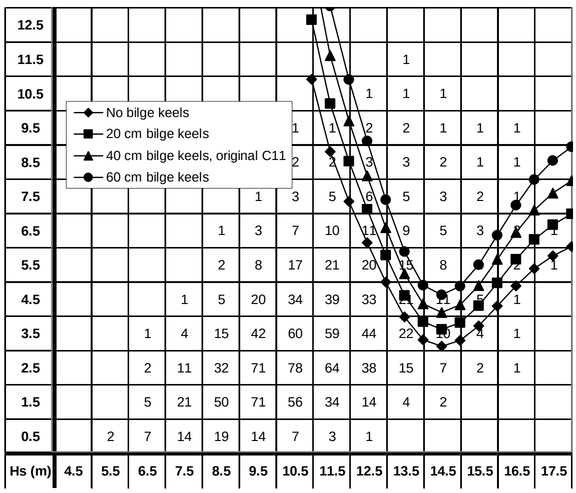


Figure 15 Influence of bilge keel height on threshold wave height

5.2 Active Fin Stabilizer

The influence of active fin stabilizers can be estimated using the same approach. The roll damping (passive and active part) was estimated using the following equation (Dallinga, 1993 and Dallinga et al., 1998).

$$B_{Fin} = 2r_{fin}\frac{1}{2}\rho V_s^2 A_{fin} C_{FH}\left[\frac{\partial C_L}{\partial \alpha_{pas}}\frac{r_{fin} C_{HF}}{V_s} + \frac{\partial C_L}{\partial \alpha_{act}} b_c\right] \qquad (19)$$

Using this equations the damping of fin stabilizers was calculated for several fin stabilizers area. The damping linearised for a 10 deg roll amplitude is given in Table 4.

Table 4 Roll damping contribution of fin stabilizers

	Fin stabilizer damping		
	0 m^2	10 m^2	21 m^2
Fin Stab damping [MNms/rad]	0	87	173
Total Damping [MNms/rad]	253	340	426

Using Dunwoody's method the threshold wave height was determined for the different fin stabilizers area. The results are given in Table 4 for 5 knot speed.

As can be seen a 10 m^2 fin stabilizer raises the threshold wave height by Hs = 0.7 m. Adding a 21 m^2 fin stabilizer raises the threshold by about 1.0 m. The results show that the fin stabilizers give approximates the same increase in threshold wave height as bilge keels for 5 knot speed. However, for speeds higher the influence of the fin stabilizers will be higher. This is due to the fact that the fin stabilizer damping increasing with the square of the speed.

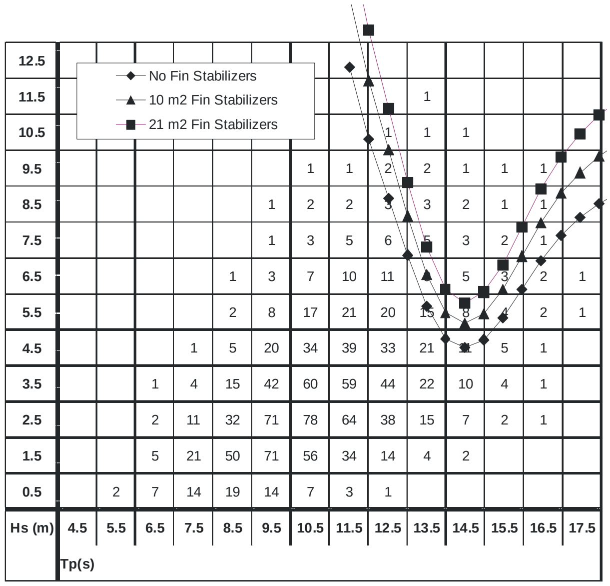


Figure 16 Influence of fin stabilizers on threshold wave height

5.3 Results summary

The results of the simulations and model tests have shown that the C11 original hull form results in a lesser probability of parametric roll to occur than the pram aft body. The pram aft body is however beneficial with respect to calm water resistance. Using the results presented in the previous chapters an estimation can be made of the bilge keel height or fin stabilizer area needed on the pram aft body in order to have a comparable threshold wave height as the C11 hull form. This estimation is given in Table 5.

Table 5 Bilge keel height or Fin stabilizer area needed for same threshold wave height

	BK height [cm]	Fin Stab [m^2]
C11 original	40	-
Pram aft body	80	-
Pram aft body	40	21

The results show that in order to achieve the same threshold wave height the bilge keel must be twice as big or the pram aft body must be equipped with a 21 m^2 fin stabilizer. These results are valid for 5 knots speed. For higher speeds the fin stabilizer will give more roll damping than the 80 cm bilge keel.

6. CONCLUSIONS

In the paper, the influence of main dimensions, hull form and roll stabilization on the occurrence of parametric roll is discussed. A one degree of freedom motion method and non linear time domain simulations were used and validated with model tests. Regarding the results the following conclusion may be drawn.

A relatively simple one degree of freedom motion method can be used in the preliminary stage of a design. An important factor on the results is the used roll damping. Empirical models for the roll damping, tuned with model tests, can be used in a preliminary stage. The method gives an idea of the threshold wave height for which parametric roll will start but not the actual roll angles associated with the rolling.

Non linear time domain simulations can be used to determine the threshold wave height and to determine the roll angles associated with the parametric roll. Here also tuning of the roll damping is needed in order to get reliable results.

The model tests and calculations on a C11 type container vessel have demonstrated that the aft body configuration has more influence on the occurrence of parametric roll than the bow flare. A V-shaped aft body is preferable to a pram type aft body.

Finally, the influence of bilge keel height on the occurrence of parametric roll has been shown. It has also been shown that active fin stabilizers can be used at low speed in order to increase the threshold wave height.

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8. NOMENCLATURE

Axx : added mass
Afin : fin stabilizer area
b_c : damping gain
B : width of the ship
Bxx : damping
Btotal : total damping
Bcrit : critical damping
Cxx : restoring
C_{Dbk} : effective drag coefficient
C_{HF} : magnification of the flow over the bilge coefficient
C_{FH} : fin to the hull coefficient
$\frac{\partial C_L}{\partial \alpha}$: lift slope
g : gravity
GMT : transverse metacentre height
h_{BK} : bilge keel height
H, Hs : wave height, significant wave height
Ixx : ship inertia
kxx : radius of gyration of roll
L or L_{pp} : length between perpendiculars
l_{BK} : bilge keel length
Mx : first order wave excitation
r_{BK} : arm of bilge keel to centre of gravity
r_{fin} : arm of fin stabilizer to centre of gravity
S_{GM} : spectral density of GM fluctuations
$S_{e\zeta}$: spectral density of the wave encounter
T_ϕ : roll natural period
Tp : wave peak period
T_{pe} : encounter wave peak period
Vs : ship speed
ϕ, $\dot{\phi}$: roll motion, roll velocity
ρ : density of water
∇ : displacement
ϖ_ϕ : natural roll frequency

ζ_a : Wave amplitude
$\Delta\xi$: Non dimensional damping reduction
λ : wave length