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PREDICTION OF SLAMMING OCCURRENCE ON CATAMARAN CROSS STRUCTURES

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ABSTRACT

In this paper, we study the problem of wave slamming on the cross structures of both fast ferry type catamarans and ocean going racing sailing catamarans. The emphasis is given to the prediction of the statistical distributions of slamming occurrence and slamming pressure magnitudes in a random seaway. A partly non-linear high-speed strip theory sea-keeping program is used to calculate the vessel motions and the relative motions between any part of the hull and the sea surface, including slamming impact velocity. Impact velocities are classified in 5 groups, and slamming pressures calculated for each group. To calculate vessel motions of heeled sailing catamarans a strip method for an asymmetric multi-hull is developed; the theory and initial results are presented. An investigation into the effect of sail forces on motions and slamming occurrence is also performed. The sail forces are found to be an important factor in predicting motions of sailing catamarans.

The procedure proposed in this paper gives the necessary information to estimate the maximum slamming pressures the vessel is likely to encounter and equally importantly the expected frequency of lighter slams, which is useful for fatigue calculations.

INTRODUCTION

Although not a new problem slamming of the ship hull against the waves has become a more important problem with the increasing popularity of high-speed catamarans in the recent years. A special concern for catamarans is the structure between the hulls, i.e. the wetdecks or the crossbeams. This structure is often flat or nearly flat, and is located a distance above the waterline. Slamming on this flat, large area cross-structure can potentially give very high slamming pressures. High-speed catamarans add to the problems with their high speed and usually also light weight. Light weight and high speed often leads to higher vertical displacements, velocities and accelerations. Light weight is the result of a carefully optimized structure and modern materials, often with a smaller tolerance against failure than conventional ships. Slamming is clearly recognized as a problem by shipmasters, and they routinely reduce speed or abort operation when the vessel slams frequently. Fast ferries have recognized slamming as an important problem for quite a long time, and slamming is now usually taken into account in the design, construction and operation of fast ferries. Nevertheless damage still occurs, both to local plating and reinforcements and global structure.

Slamming has also been a problem for sailing catamarans, and with the recent trend for large high-speed ocean-crossing sailing catamarans slamming has quickly become one of the major concerns for this type of boats. Compared to high-speed ferries, sailing catamarans are considerably lighter and in some conditions sails with a speed comparable to that of a fast ferry. In a recent non-stop race around the world, THE RACE, most of the yachts suffered damages from slamming. One yacht had to make a stop in Cape Town and later New Zealand to repair a delaminated main beam, damaged by fatigue from slamming. A sailing catamaran can be operated in two modes; when the sail generated heeling moment is under a threshold value the boat will sail with both hulls in the water, with a slight heel angle generating a righting moment. When the heeling moment increases above this threshold value the windward hull will lift clear of the water and the righting moment curve will reach its maximum value and flatten out. This is not a stable condition, but a well-balanced boat with a good crew can sail the boat with a hull flying at a nearly steady heel angle.

Traditional way of predicting slamming occurrence employs linear theory in the frequency domain to calculate the ship RAOs. Given the ship RAOs and a sea spectrum the probability of slamming assuming a Rayleigh distribution can be calculated by formulas given by [1], hereafter referred to as Ochi's method. The probability of slamming is calculated as the joint probability of water entry and relative velocity exceeding a threshold value. The threshold value has been empirically determined for conventional monohull ships when slamming-induced buckling was a major concern.

In the present study we introduce a direct approach that enables the calculation of the motions of the ship in the time domain, in theory without any assumptions of linearity of the ship motions or Rayleigh distributed peak values. The direct approach is performed in three steps. In the first step the global rigid body motions are calculated in an irregular sea. From the time series all downwards crossings of the sea surface and the accompanying relative vertical velocity is calculated at several positions on the cross structure. Finally in the third step the local slamming pressures are calculated using the relative vertical velocity. Details about the calculations are given in the following sections. The advantage of this method is that it fully utilizes the accuracy in the calculation of motions and slamming pressures by e.g. non-linear calculations or full 3D calculations. It also enables the calculation of slamming occurrence at all severity levels. The effect of slamming on ship motions is not included. It could be argued that the slamming effect on the motions is not vital to the prediction of slamming occurrence, as the slam will only affect the motion during and after each slam. Given a reasonable time between each slamming impact the ship will adjust itself into its natural pattern again. In the slamming calculations the impact velocity is assumed constant, which is an extreme case.

Sailing catamarans can be analyzed in much the same way as power catamarans. There are however some extra considerations caused by the steady heeling angle and both steady and unsteady forces from the sails. Two methods are used in this paper to predict motions of sailing catamarans. A simplified approach allows for the use of non-linear high-speed strip theory while a more complete modeling of a heeled catamaran hull including the effect of sail forces will be presented based on a linear theory.

NOMENCLATURE

CG = the boats center of gravity
η = displacement at CG
ζ = wave elevation
Z_R = relative displacement between the ship and wave surface
ω = circular frequency
ω_e = circular encounter-frequency
V = boat speed
U = wind speed
U_1 = wind speed parallel to x axis
U_2 = wind speed parallel to y axis
β_T = true wind angle
β = wave heading angle
A = added mass
B = damping
C = restoring coefficient
F = exciting force
a = sectional added mass
b = sectional damping
c = sectional restoring coefficient
f = sectional Froude-Krylov force
h = sectional diffraction force
Hs = significant wave height
Tp = wave spectra peak period
xc = sail strip chord length
$\hat{\eta}_4$ = roll amplitude
$\hat{\eta}_5$ = pitch amplitude

SLAMMING OF FAST FERRY TYPE CATAMARANS

Methodology

For the motion simulations a non-linear high-speed strip theory program is used [2-4]. The use of a non-linear sea keeping code is important as slamming usually occurs during large amplitude motions, and it also enables the effect of bow flare on slamming to be studied. Although not a fully linear program, the non-linear version of VERES accounts for nonlinear added mass, damping, Froude-Krylov, diffraction and restoring forces as a modification to the forces obtained from linear theory. The effect of slamming on ship motions is neglected.

Given the displacements in heave, pitch and roll it is straightforward to calculate the vertical motions at arbitrary positions on the vessel. Local origin is positioned at center of gravity horizontally, and in plane with the still water line vertically. The right handed coordinate system has the X-axis positive aft, and the Z-axis positive upwards.

The vertical displacement at a specific location i may be calculated as

$$\eta_{3i} = \eta_3 - X_i * \sin(\eta_5) + Y_i * \sin(\eta_4) \qquad (1)$$

Vertical velocity may be calculated in a similar manner using the velocities at CG obtained by differentiation of the displacement with respect to time.

Knowing the individual wave components used to simulate the irregular sea in the motion simulation the wave elevation at an arbitrary point X_i, Y_i is calculated as

$$\zeta_i(t) = \sum_{j=1..n} \zeta_{a\,j} \sin(\omega_{e\,j} t - k_j (X_i \cos(\beta) + Y_i \sin(\beta)) + \varepsilon_j) \qquad (2)$$

Similarly the wave vertical velocity is obtained by taking the total derivative of the wave elevation with respect to time,

$$\dot{\zeta}_i(t) = \sum_{j=1..n} \omega_j \zeta_{a\,j} \cos(\omega_{e\,j} t - k_j (X_i \cos(\beta) + Y_i \sin(\beta)) + \varepsilon_j) \tag{3}$$

The relative vertical displacement at an arbitrary point X_i, Y_i, Z_i is calculated as

$$Z_{R\,i} = Z_i + \eta_{3i} - \zeta_i \tag{4}$$

In the calculation of the relative vertical velocity there will be a contribution due to forward speed and pitch, $V \sin(\eta_5)$.

$$\dot{Z}_{Ri} = \dot{\eta}_{3i} - \dot{\zeta}_i - V \sin(\eta_5) \tag{5}$$

In the present analysis, relative vertical motions are calculated for each time step, and all downwards zero crossings of the relative displacement Z_R are identified as slamming events. A minimum impact velocity of $0.093\sqrt{gL}$ as proposed by Ochi [5] has been widely used, but since this was empirically determined for a conventional type of ship it is doubtful whether it is applicable for all ships. Instead, in this study the relative vertical velocity and the slamming pressure are divided into 5 categories, based on relative vertical velocity squared. This gives the designer the necessary information to estimate the maximum slamming pressure the vessel is likely to encounter, as well as the number of lighter slams important for fatigue calculations.

The slamming pressure is calculated for each category of impact velocity and each location considered,

$$P_{SLAM} = \frac{1}{2} \rho \dot{Z}_{Ri}^{\,2}\, k_{SLAM} \tag{6}$$

A 2-dimensional non-linear slamming theory [6] is used to calculate the slamming pressure coefficient k_{SLAM} for each location considered. The wet-deck or cross structure is divided into longitudinal strips, and the effect of transverse flow is neglected. Hydro-elastic effects are also neglected. This is obviously a simplification, but calculation of slamming pressure on catamaran wetdecks is a field where few calculation methods have reached a mature state. Most catamaran wetdecks have no or very small transverse curvature justifying the 2D approach. Slamming usually occurs in the forward part of the wetdeck due to the strong influence of pitch motions near the bow. Due to this fact most catamarans have a raised wetdeck height near the bow, introducing a longitudinal curvature. It is believed that this curvature reduces the effect of hydro-elasticity, and that a rigid body approach can be justified for the present purpose.

Results for a 30m fast ferry

In this section computational results are presented for a typical 30m fast ferry hull, with lines plan and main particulars shown below.

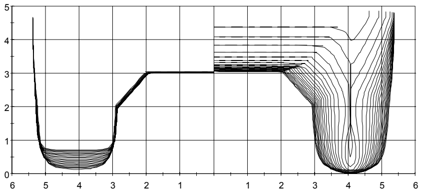

Fig. 1 Lines plan of a 30m fast ferry
Aft station shown left, fwd stations shown right

Loa	30m
Boa	11m
Displacement	112t

Table 1 Main particulars of the ferry

The calculations are performed in long-crested irregular seas simulated by a Bretschneider spectrum [7]. The slamming pressure coefficient $k_{SLAM} = 79.6$ has been calculated and averaged over a 2m panel in the forward part of the wetdeck. In the slamming analysis the wetdeck was rotated bow down corresponding to a pitch angle that would give impact in calm water. In Figs 2 and 3, the results from the direct non-linear simulation are compared with a linear frequency domain calculation using Ochi's statistical approach (denoted linear in fig. 2). The latter predicts a drastically higher slamming frequency than the direct approach. The average slamming pressure shows a similar trend as the slamming frequency, and it is worth noting that reducing speed from 40 to 30 knots makes the situation worse in some cases. The distribution of slamming events is as expected dominated by lighter slams, but a number of severe slams do occur, see Fig. 4.

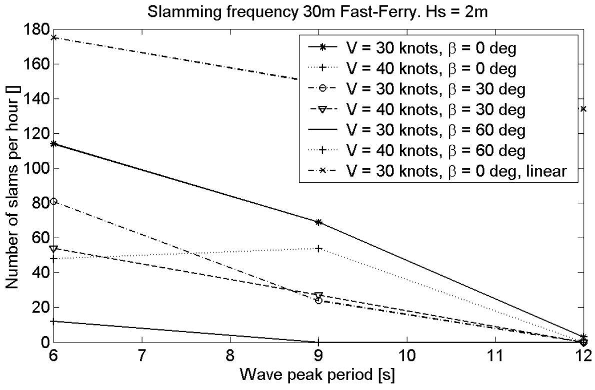

Fig. 2 Wetdeck average slamming pressure vs peak period

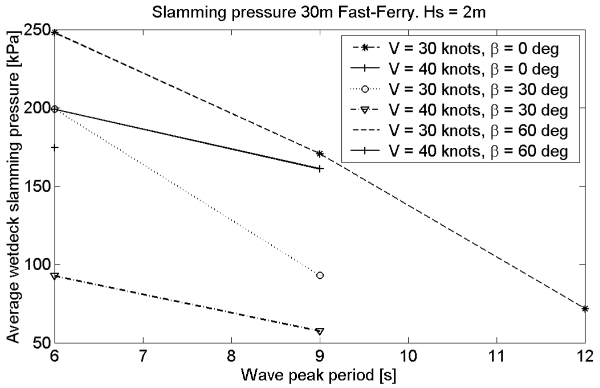

Fig. 3 Wetdeck average slamming pressure vs peak period

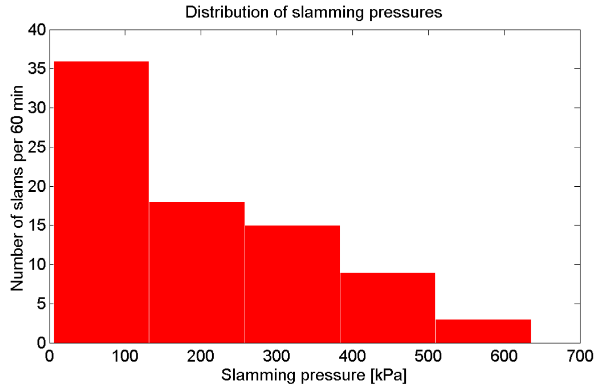

Fig. 4 Main beam slamming pressure distribution, Hs = 2m, Tp = 6s, V = 30 knots, Heading = 30 deg

SLAMMING OF SAILING CATAMARANS

Non-linear calculations

To avoid asymmetry the catamaran can be analyzed in two extreme sailing modes, either as a symmetric catamaran sailing at zero heel angle or as a symmetric monohull when flying the windward hull. Roll motion is highly dependent on sail forces and crew action but is neglected in this analysis. The variation of sail forces from surge and pitch motion is also neglected. This enables the prediction of motions using non-linear high-speed strip theory and prediction of slamming occurrence following the same procedure as described for fast ferry catamarans.

Results from time-domain simulation

Crowther Multihulls, Sydney provided the case study design used in the simulations. Crowther hull 318 is a racing catamaran of the type Southern Ocean 50. The main particulars are

Loa	15.2m
Boa	11m
Displacement (sailing condition)	3700kg

Table 2 Main particulars of the sailing yacht

The results shown in Figs 5-9 are all performed with incoming waves 45 deg off the bow, with one hull flying. A total of 60 simulations are performed, each with a run time of 30 min. All simulations are made in a long-crested irregular sea simulated by a Bretschneider spectrum. The slamming pressure coefficient $k_{SLAM} = 40.3$ has been calculated and averaged over a .4m panel of the main beam. The slamming frequency is as expected higher for shorter waves, as the number of waves encountered per hour is higher. It is notable that it is the two lowest speeds that is most affected by slamming, at least for shorter waves. The average slamming pressure shows a similar trend; again low speed in short waves is the worst condition.

The distribution of slamming pressures in a single run is a particularly interesting result. The pressure distribution shown in Fig. 9 is the slamming pressure averaged over an area of the main beam, calculated by SLAM2D. The lighter slams clearly dominate the results, but a few slams have an impact pressure of up to 4 times the average slamming pressure.

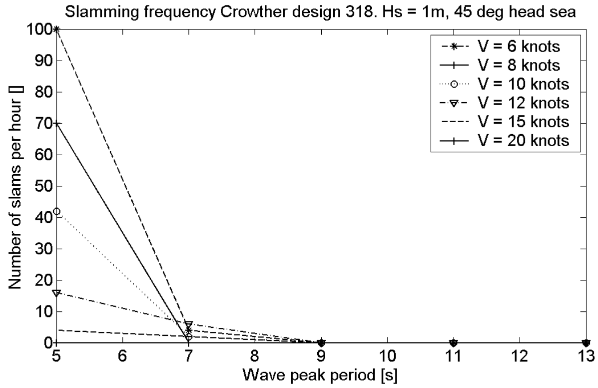

Fig. 5 Main beam slamming frequency vs peak period

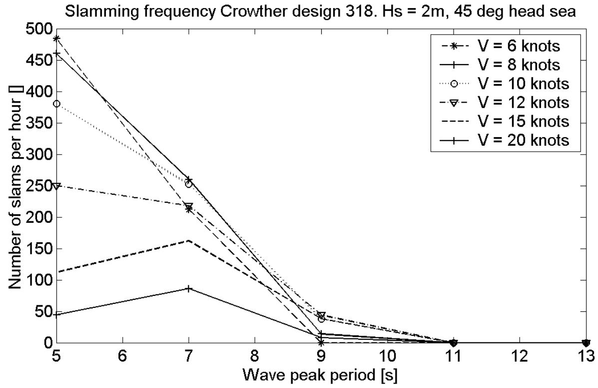

Fig. 6 Main beam slamming frequency vs peak period

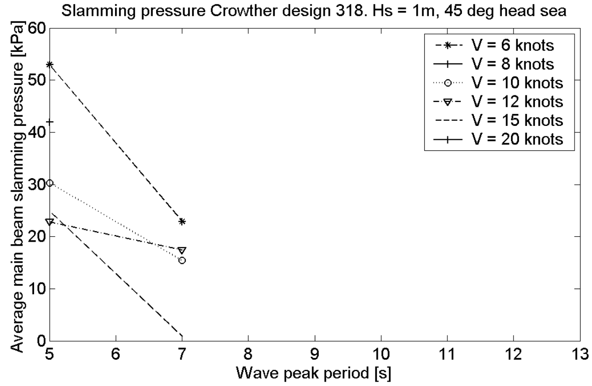

Fig. 7 Main beam average slamming pressure vs peak period

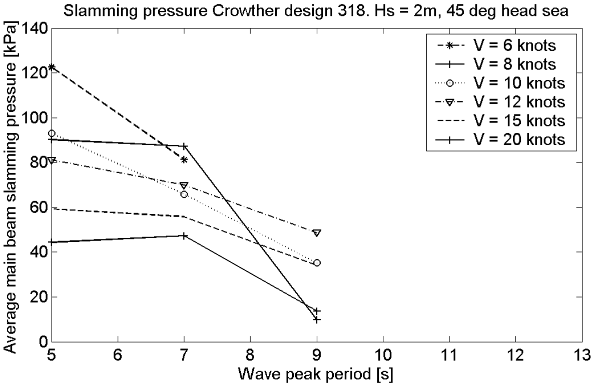

Fig. 8 Main beam average slamming pressure vs peak period

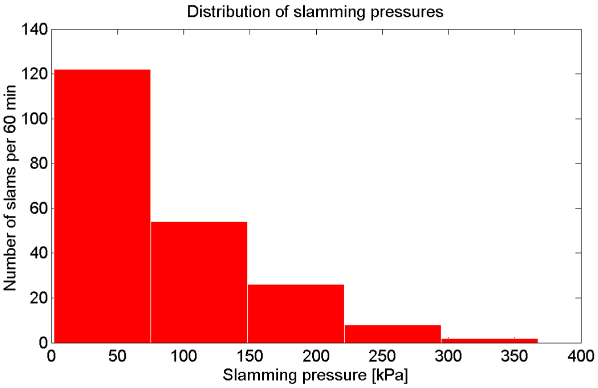

Fig. 9 Main beam slamming pressure distribution, Hs = 2m, Tp = 7s, V = 6 knots

Asymmetric strip theory

The simplified treatment of a sailing catamaran is a useful and effective method to investigate slamming and seakeeping of catamarans when flying a hull. However, some catamarans spend a lot of time in a steady heeling but non-flying condition, sailing as an asymmetric multihull. To investigate the seakeeping properties of a heeled catamaran a linear frequency domain strip theory program capable of handling asymmetric multihulls is developed based on the strip theory proposed by Salvesen, Tuck and Faltinsen [8] (hereafter referred to as STF). STF outlines the theory first without any assumptions of symmetry, but simplifies the calculation of added mass, damping, stiffness and excitation forces by assuming the ship to be symmetric. For a symmetric ship heave and pitch can be calculated independently from sway, roll and yaw motions. In the general asymmetric case they cannot be considered independent. Limiting the motions of interest to heave, roll and pitch a coupling between heave and roll as well as coupling between roll and pitch must be considered.

a) Equations of motion

The general equations of motions in heave, roll and pitch can be written as

$$\sum_{k=3}^{5}\left[(M_{jk}+A_{jk})\ddot{\eta}_k+B_{jk}\dot{\eta}_k+C_{jk}\eta_k\right]=F_je^{i\omega t} \quad ; \; j=3\,..\,5 \qquad (7)$$

The only coefficients to be considered in the mass matrix M when Cg is located at (0, ycg, zcg) is

$$M=\begin{bmatrix} - & - & - & - & - & - \\ - & - & - & - & - & - \\ - & - & m & my_{cg} & mz_{cg} & - \\ - & - & my_{cg} & mr_{44} & mr_{45} & - \\ - & - & mz_{cg} & mr_{54} & mr_{55} & - \\ - & - & - & - & - & - \end{bmatrix} \qquad (8)$$

b) Calculation of hydrodynamic coefficients

The total added mass, damping and excitation forces can be calculated as

$$A=A^0+A^F+A^T\,,$$
$$B=B^0+B^F+B^T\,,$$
$$F=F^0+F^F+F^T \qquad (9\text{ a,b,c})$$

Calculation of forward speed terms (superscript F) and transom terms (superscript T) are outlined in STF, and not repeated here. In what follows, zero-speed added mass A^0, zero-speed damping B^0 and stiffness C are calculated by integrating sectional properties along the waterline. Exciting forces are calculated by integrating Froude-Krylov and diffraction forces along the waterline.

c) Calculation of sectional hydrodynamic properties

The calculation of sectional properties may be simplified by assuming each hull to be symmetric about its own center-plane. The roll moment about each hull's own axis is also assumed negligible compared to the total roll moment of the ship, an assumption valid when the demihull separation is high compared to the beam of the individual hulls. The sectional properties in heave, roll and pitch can then be calculated from the sectional properties in heave for each hull. This simplification is valid for small roll angles, consistent with linear theory.

The sectional properties in heave can be easily calculated from the individual sectional properties of each hull in heave. Superscript j denotes sectional property of a single hull.

$$a_{33} = \sum_{j=1}^{2} a_{33}^{j} \qquad b_{33} = \sum_{j=1}^{2} b_{33}^{j} \qquad c_{33} = \sum_{j=1}^{2} c_{33}^{j} \qquad \text{(10 a,b,c)}$$

$$f_3 = \sum_{j=1}^{2} f_3^{j} * e^{-ik(x\cos\beta - y^{j}\sin\beta)},$$

$$h_3 = \sum_{j=1}^{2} h_3^{j} * e^{-ik(x\cos\beta - y^{j}\sin\beta)} \qquad \text{(11 a,b)}$$

Similarly for pitch,

$$a_{55} = \sum_{j=1}^{2} x^2 a_{55}^{j} \qquad b_{55} = \sum_{j=1}^{2} x^2 b_{55}^{j} \qquad c_{55} = \sum_{j=1}^{2} x^2 c_{33}^{j} \qquad \text{(12 a,b,c)}$$

$$f_5 = \sum_{j=1}^{2} x f_3^{j} * e^{-ik(x\cos\beta - y^{j}\sin\beta)},$$

$$h_5 = \sum_{j=1}^{2} x h_3^{j} * e^{-ik(x\cos\beta - y^{j}\sin\beta)} \qquad \text{(13 a,b)}$$

roll,

$$a_{44} = \sum_{j=1}^{2} y^2 a_{33}^{j} \qquad b_{44} = \sum_{j=1}^{2} y^2 b_{33}^{j} \qquad c_{44} = \sum_{j=1}^{2} y^2 c_{33}^{j} \qquad \text{(14 a,b,c)}$$

$$f_4 = \sum_{j=1}^{2} y f_3^{j} * e^{-ik(x\cos\beta - y^{j}\sin\beta)},$$

$$h_4 = \sum_{j=1}^{2} y h_3^{j} * e^{-ik(x\cos\beta - y^{j}\sin\beta)} \qquad \text{(15 a,b)}$$

pitch-heave coupling,

$$a_{35} = -\sum_{j=1}^{2} x a_{55}^{j} \qquad b_{35} = -\sum_{j=1}^{2} x b_{55}^{j} \qquad c_{35} = -\sum_{j=1}^{2} x c_{33}^{j} \qquad \text{(16 a,b,c)}$$

$$a_{53} = a_{35} \qquad b_{53} = b_{35} \qquad c_{53} = c_{35} \qquad \text{(17 a,b,c)}$$

roll-heave coupling,

$$a_{34} = \sum_{j=1}^{2} y a_{33}^{j} \qquad b_{34} = \sum_{j=1}^{2} y b_{33}^{j} \qquad c_{34} = \sum_{j=1}^{2} y c_{33}^{j} \qquad \text{(18 a,b,c)}$$

$$a_{43} = a_{34} \qquad b_{43} = b_{34} \qquad c_{43} = c_{34} \qquad \text{(19 a,b,c)}$$

and roll-pitch coupling,

$$a_{45} = \sum_{j=1}^{2} yx a_{33}^{j} \qquad b_{45} = \sum_{j=1}^{2} yx b_{33}^{j} \qquad c_{45} = \sum_{j=1}^{2} yx c_{33}^{j} \qquad \text{(20 a,b,c)}$$

$$a_{54} = c_{45} \qquad b_{54} = c_{45} \qquad c_{54} = c_{45} \qquad \text{(21 a,b,c)}$$

The coupling terms for roll-heave and roll-pitch will all be zero for a symmetric hull, but must be included for an asymmetric hull-shape. An effect of the asymmetric coupling with only academic interest for sailing catamarans is the possibility of roll motion in pure head sea!

Results from the new strip method

No data are currently available to validate the asymmetric part of the theory, but here a comparison is made with the slender-body theory program VERES for a symmetric catamaran at zero heel angle or a symmetric monohull in flying mode with 5 deg of heeling angle. The results shown in Figs 10-12 are a comparison with the linear high-speed strip theory version of VERES. The agreement is very encouraging. An even better comparison was found to the low speed version of VERES but not included in this presentation. The results from the 2 deg heeled, asymmetric case are also plotted. The large difference in the RAO curves from the different heel angles can be attributed to the special hull shape of the case study ship. The hydrodynamic properties of the hull changes rapidly with changing draft, and consequently heel-angle.

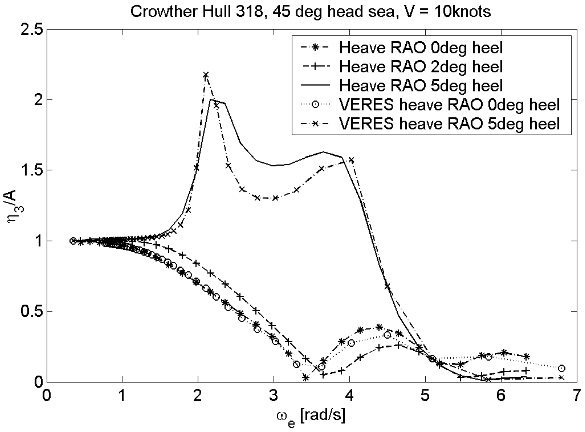

Fig. 10 Heave RAO at different heel angles

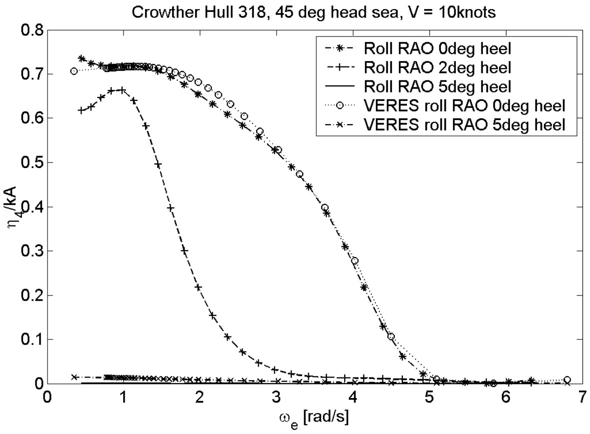

Fig. 11 Roll RAO at different heel angles

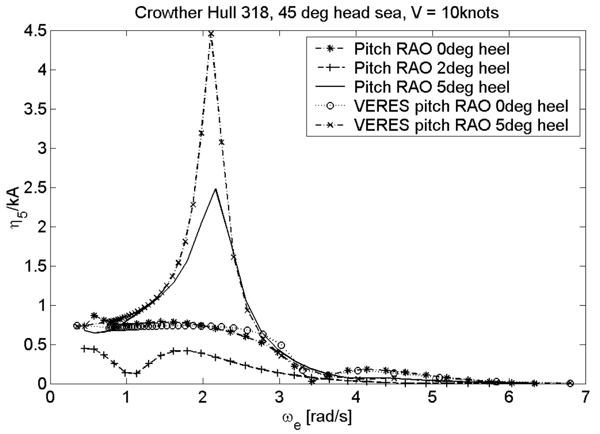

Fig. 12 Heave RAO at different heel angles

Sail forces

A simple model to investigate the effect of sail forces on the motion may be implemented in the linear frequency domain strip theory. To this end, linearized sail damping coefficients are obtained based on a simple time domain model of the sail forces. The relative wind is composed of the true wind, the boat speed and an unsteady contribution from roll and pitch motion. Contributions from surge and sway are neglected, and the heel angle is assumed small. Summing the velocity components in longitudinal and transverse directions we obtain for a strip located at height z over the waterline

$$U_1(z,t) = U_0 \cos \beta_T + V + V_{pitch}(z) \tag{22}$$

$$U_2(z,t) = U_0 \sin \beta_T + V_{roll}(z) \tag{23}$$

$$V_{roll}(z,t) = \dot{\eta}_4 * z = \omega_e \hat{\eta}_4 z e^{i\omega t} \tag{24}$$

$$V_{pitch}(z,t) = \dot{\eta}_5 * z = \omega_e \hat{\eta}_5 z e^{i\omega t} \tag{25}$$

The relative wind is now given by

$$U_{rel}(z,t) = \sqrt{U_1^{\;2} + U_2^{\;2}} \tag{26}$$

and

$$\beta_R(z,t) = a\tan\frac{U_2}{U_1} \tag{27}$$

For upwind sailing the drag force is an order of magnitude smaller than the lift force, and is neglected. Assuming small variations in the relative wind angle the side force and driving force can be calculated using a constant lift coefficient, supported by wind tunnel data given by [9]. Any hysteresis effect from the oscillatory motions is neglected.

$$C_S(z,t) = C_L \cos(\beta_R) \tag{28}$$

$$C_D(z,t) = C_L \sin(\beta_R) \tag{29}$$

The side force and drive force for a strip is now calculated as

$$S(z,t) = \frac{1}{2}\rho_{air} xc(z) C_S U_{rel}^{\;2} \tag{30}$$

$$D(z,t) = \frac{1}{2}\rho_{air} xc(z) C_D U_{rel}^{\;2} \tag{31}$$

where xc is the chord length of a strip

and total heel and pitch moment can be calculated as

$$M_{roll}(t) = \int_{z_{\min 1}}^{z_{\max}} S(z) * z * dz \tag{32}$$

$$M_{pitch}(t) = \int_{z_{\min}}^{z_{\max}} D(z) * z * dz \tag{33}$$

with the integration performed over the total height of the sail.

The forces are unfortunately non-linear, and must be linearized in order to be used in a linear frequency domain theory. The damping forces are calculated as

$$B_{roll} = \sqrt{\frac{\int_0^{\frac{2\pi}{\omega}} M_{roll}^2(t)dt}{\int_0^{\frac{2\pi}{\omega}} \sin^2 \omega t dt}} \tag{34}$$

$$B_{pitch} = \sqrt{\frac{\int_0^{\frac{2\pi}{\omega}} M_{pitch}^2(t)dt}{\int_0^{\frac{2\pi}{\omega}} \sin^2 \omega t dt}} \tag{35}$$

The damping coefficients can finally be calculated,

$$B_{sail44} = \left.\frac{B_{roll}}{\hat{\eta}_4}\right|_{\hat{\eta}_5=0} \qquad B_{sail45} = \left.\frac{B_{roll}}{\hat{\eta}_5}\right|_{\hat{\eta}_4=0}$$

$$B_{sail54} = \left.\frac{B_{pitch}}{\hat{\eta}_4}\right|_{\hat{\eta}_5=0} \qquad B_{sail55} = \left.\frac{B_{pitch}}{\hat{\eta}_5}\right|_{\hat{\eta}_4=0}$$

(36 a,b,c,d)

Results incorporating sail forces

The influence of the sails is found to be very important, with similar influence in both pitch and roll. A comparison is made for linear strip theory with and without sails against results from a non-linear high-speed strip theory code. To compare the results the RAOs from the frequency domain calculations were used to generate time series, with the same wave components as used in the non-linear computation. Results shown in this section are made for Crowther design 318 at 10 knots in a long-crested Bretschneider irregular sea, with $H_s = 1m$, $T_P = 5s$ and 45 deg head sea, see Fig. 13. To compare the results with the non-linear code all calculations are performed for the flying mode, with roll motion locked. It is clear that the modeling of sail forces is important, with a greater impact on the motions than non-linear and high-speed effects. The sail is very effective in smoothing the motions in pitch, but it does also have an effect in heave. No data are presently available for validation.

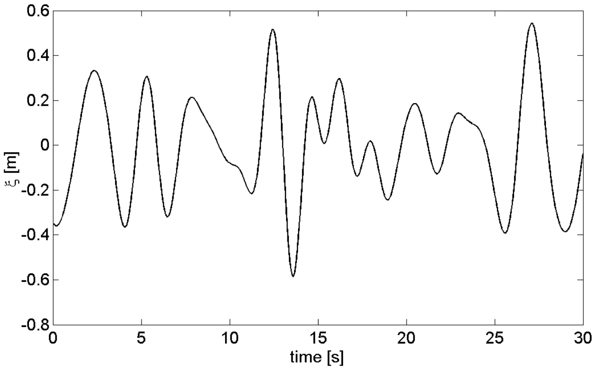

Fig. 13 Irregular wave time series

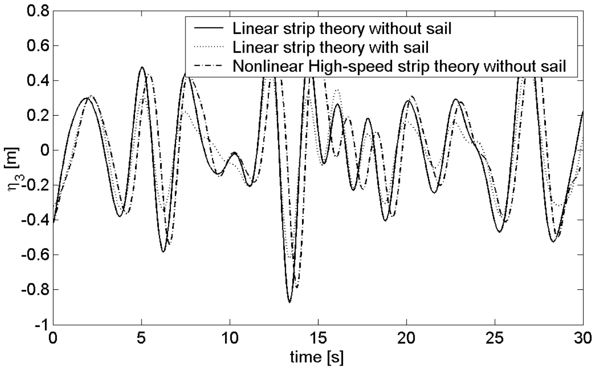

Fig. 14 Heave time series with and without sail

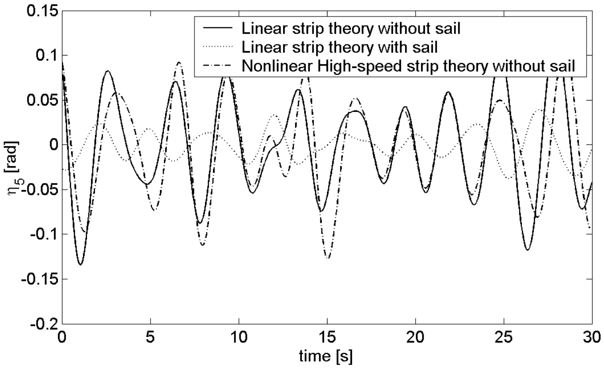

Fig. 15 Pitch time series with and without sail

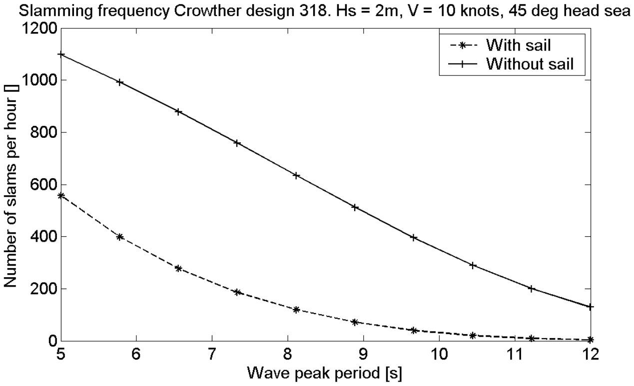

Fig. 16 Effect of sail forces on slamming frequency

Using the statistical approach (Ochi's method[5]) seems to predict significantly higher slamming frequency than the direct approach. This is partly a result of using linear theory in motion prediction but it is also believed that the two methods of calculation give slightly different results. The assumption that the relative vertical velocity and relative vertical displacement is statistically independent in Ochi's method could be one reason, but no further investigation has been done. It is however clear that the effect of a sail is remarkable, effectively reducing the slamming frequency by 50% for this condition, see Fig. 16.

CONCLUDING REMARKS

The proposed procedure for statistical quantification of slamming occurrence and severity on catamaran cross structures has shown to be a promising one to enhance the design and operation of such vessels. The method enables the estimation of both slamming-induced extreme loads and fatigue loads. It may be applied to conventional monohulls.

The proposed strip theory for asymmetric multihulls is a novel method for heeled sailing vessels. Preliminary validation has confirmed its applicability.

The proposed simplified method for the effect of wind on motions and slamming occurrence of sailing vessels enables a real scenario being analyzed for such vessels. One validation is being made through full-scale measurements.

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