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The Whipping Vibratory Response of a Hydroelastic Segmented Catamaran Model

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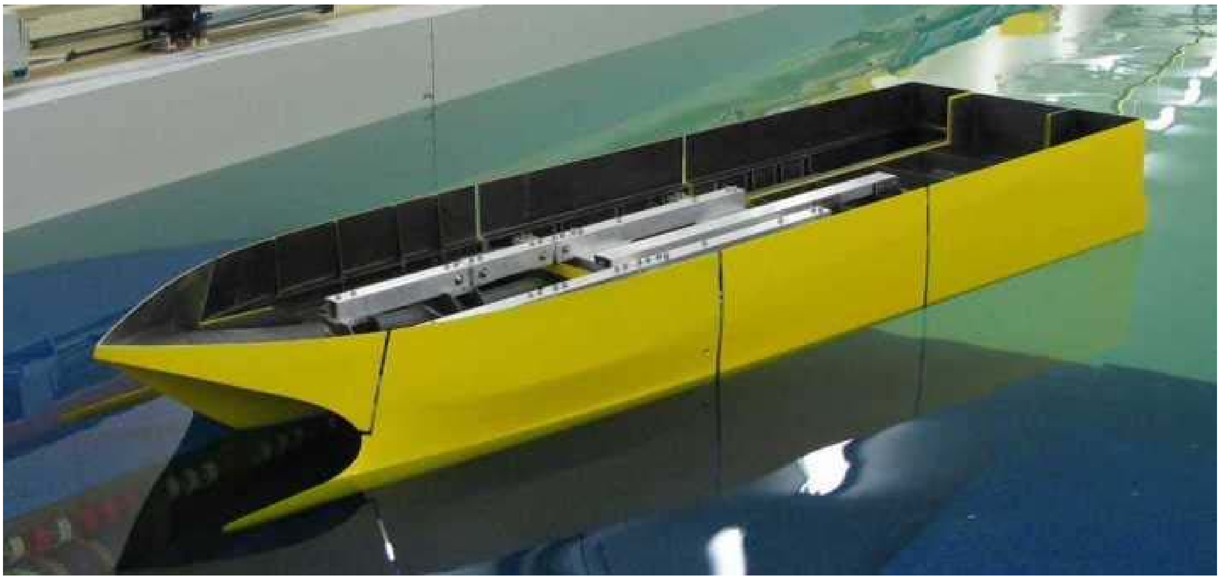

Figure 1: Hydroelastic segmented catamaran model of a 112 m INCAT high-speed wave-piercer catamaran.

ABSTRACT

The parameters affecting the whipping vibratory response of a high-speed catamaran subject to slamming were investigated by hydro-elastic segmented model tests undertaken in calm water. Wet and dry vibration tests at zero speed show significant influences of variations in stiffness and mass of the model on the flexural bending frequency. The zero speed damping ratio was unaffected by changes in stiffness and model mass. The flexural frequencies were predicted with a three degree of freedom theoretical model using an added water mass approximation and gave good correlation with experimental data. The effects of forward speed on the whipping response of the model in calm water were investigated and there were significant increases in the damping ratio with increasing forward speed. There was negligible effect of speed on the whipping frequency.

Keywords

Slamming, whipping, segmented model, hydroelasticity.

1 INTRODUCTION

Increasing demands for high-speed transportation have led to the rapid development of high-speed catamarans and high-speed monohulls for both commercial and military applications. In order to achieve high-speed design targets ship manufacturers have been confronted with the on-going challenge of reducing the weight through the use of light-weight materials and the application of optimal structural design processes. The combination of high-speed and flexible light weight aluminium materials has increased the vulnerability of high-speed vessels in terms of strength and fatigue life. This raises the need to develop a better understanding of the dynamic structural loads on high-speed craft, in particular the slamming and whipping responses with respect to the strength and fatigue life of the vessel.

The development of geometrically similar segmented test models provides an effective method for determining dynamic structural loads (Kapsenberg et al 1999) and whipping responses (McTaggart et al. 1998). The hydroelastic segmented model design facilitates simulation of the full-scale fluid-structure interaction for the purpose of measuring wave slamming loads and replicating whipping vibratory responses (Riska et al. 1994). The techniques used in previous research studies have most commonly applied finite element methods to predict the dry whipping responses of hydroelastic segmented models such as the segmented catamaran model of Hermundstad et al. (1995) and the high-speed monohull of Dessi et al. (2005). The work in this paper seeks to investigate the parameters affecting both the wet and dry flexural responses of a hydroelastic segmented catamaran model through physical experiments and a rigid segment theoretical model. The results of the investigation identify the parameters affecting the whipping flexural modal response of the hydroelastic segmented model providing a basis for further research into the measurement of global wave loads and localised slam loads.

2 WHIPPING MODAL RESPONSE

Wave induced slamming occurs when the bow of the vessel encounters a wave that causes an impulsive load on the structure resulting in instantaneous impulsive flexure and a whipping vibratory response. Slam impulse loads applied to the bow in head seas most often excite the first longitudinal mode of vibration in the vertical plane as discussed by Thomas et al. (2003a). The slamming and subsequent whipping frequencies measured in full scale vessel trials on 86 m and 96 m INCAT catamarans (Thomas et al. 2003b and 2003c) provide a basis for predicting the first longitudinal mode of vibration of the new 112m catamaran. The predicted whipping frequency of the 112 m catamaran was then used as a basis for estimating the flexural response frequency of a 2.5 m scaled segmented catamaran model. The scaling of the whipping frequency was determined by assuming a constant dimensionless wave encounter frequency between the geometrically similar model and full scale vessel. For the purpose of evaluating the whipping frequency of the scaled segmented model the following scaling relationship is used:

$$\omega_m = \omega_f \sqrt{\frac{l_f}{l_m}}, \qquad (1)$$

where ω_m = model scale modal frequency, ω_f = full scale modal frequency, l_f = full-scale length of vessel, l_m = model scale length. Table 1 shows the results of the natural bending frequencies measured on the 86 m and 96 m INCAT catamarans (Thomas 2003d) and the predicted natural bending frequencies of the 112 m catamaran and the 2.5 m hydroelastic segmented catamaran model.

Table 1: First longitudinal modal frequencies.

Hull	Overall length (m)	Displacement	1st-longitudinal modal frequency (Hz)
045	86	880 t †	3.01
050	96	1100 t †	2.89
064	112.5	2500 t ‡	2.06
HSM	2.5	27.43 kg ‡	13.79

† Light weight ship, ‡ Loaded ship.

Given the results gathered from previous full scale trials, the 112 m catamaran first longitudinal mode of vibration was predicted to be 2.06 Hz for a loaded ship at a displacement of 2500 tonnes. To simulate the whipping response of the 112 m catamaran vessel with a 2.5 m hydroelastic segmented catamaran model required a target model whipping frequency of 13.79 Hz. Although the segmented model concept is an approximation of the full-scale vessel flexural dynamic response, it facilitates a practical simulation of the whipping phenomena observed on the full-scale vessel.

3 THEORETICAL BACKGROUND

The theory and design methodology developed in a previous study undertaken on a NPL 6A hydroelastic segmented model (Lavroff et al. 2006), assisted with the technical development of the hydroelastic segmented catamaran model. The principles used to predict the two degree of freedom wet flexural mode response of the NPL 6A hydroelastic model were used as a basis for developing a three degree of freedom theoretical model used to predict both the wet and dry flexural mode response of the hydroelastic segmented catamaran model. Previous research conducted on hydroelastic models has made extensive use of finite element methods to predict the modal response (Hermundstad et al. 1999). However, the present authors have found the development of a theoretical model based on rigid hull segments connected by elastic links provided an effective means for predicting both the wet and dry whipping flexural mode response of the test model.

A three degree of freedom theoretical model was developed to predict the longitudinal bending frequency of the catamaran model as a function of the effective link stiffness and mass distribution of the physical model. The theory takes into consideration a single catamaran demihull consisting of three separate rigid hull segments joined together by forward and aft torsion springs as shown schematically in Figure 2. During a slam event the forward and aft segments of the hull vibrate in a rotational direction opposite to each other and against the rotational stiffness of the torsion springs to replicate the first longitudinal mode of vibration of a catamaran demihull.

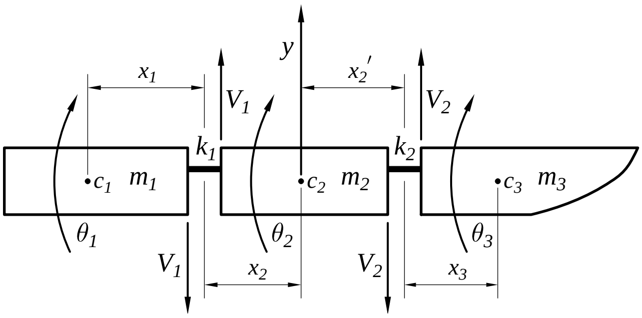

Figure 2: Schematic diagram of the three degree of freedom spring mass system.

Let

- m = mass of complete hull
- m_1 = mass of aft hull segment
- m_2 = mass of mid-ship hull segment
- m_3 = mass of forward hull segment
- k_1 = stiffness of aft torsion spring
- k_2 = stiffness of forward torsion spring
- c_1 = centre of mass of aft hull segment
- c_2 = centre of mass of mid-ship hull segment
- c_3 = centre of mass of forward hull segment
- x_1 = distance from aft torsion spring to centre of mass of aft hull segment
- x_2 = distance from aft torsion spring to centre of mass off mid-ship hull segment
- x_2' = distance from forward torsion spring to centre of mass of mid-ship hull segment
- x_3 = distance from forward torsion spring to centre of mass of forward hull segment
- I_1 = moment of inertia of aft hull segment about centre of mass, c_1
- I_2 = moment of inertia of mid-ship hull segment about centre of mass, c_2
- I_3 = moment of inertia of forward hull segment about centre of mass, c_3
- θ_1 = clockwise rotation of aft hull segment
- θ_2 = clockwise rotation of mid-ship hull segment
- θ_3 = clockwise rotation of forward hull segment
- y = vertical displacement of mid-ship hull segment centre of mass, c_2
- V_1 = shear force at aft torsion spring
- V_2 = shear force at forward torsion spring

The equations of motion for the three degree of freedom spring mass system are then as follows:

$$V_1 = -m_1\left(\ddot{y} + x_2\ddot{\theta}_2 + x_1\ddot{\theta}_1\right) \tag{2}$$

$$V_2 = m_3\left(\ddot{y} - x_2'\ddot{\theta}_2 - x_3\ddot{\theta}_3\right) \tag{3}$$

$$V_1 - V_2 = m_2\ddot{y} \tag{4}$$

$$V_1 x_1 - k_1\left(\theta_1 - \theta_2\right) = I_1\ddot{\theta}_1 \tag{5}$$

$$V_1 x_2 + V_2 x_2' - k_2\left(\theta_2 - \theta_3\right) + k_1\left(\theta_1 - \theta_2\right) = I_2\ddot{\theta}_2 \tag{6}$$

$$V_2 x_3 + k_2\left(\theta_2 - \theta_3\right) = I_3\ddot{\theta}_3 . \tag{7}$$

Substituting (2) and (3) into (4), (5), (6), (7) and eliminating $\ddot{y}$,

$$-m_1 x_1\left(\frac{m_3 x_2' - m_1 x_2}{m}\ddot{\theta}_2 + \frac{m_3 x_3}{m}\ddot{\theta}_3 - \frac{m_1 x_1}{m}\ddot{\theta}_1 + x_2\ddot{\theta}_2 + x_1\ddot{\theta}_1\right)$$
$$-k_1\left(\theta_1 - \theta_2\right) = I_1\ddot{\theta}_1 \tag{8}$$

$$-m_1 x_2\left(\frac{m_3 x_2' - m_1 x_2}{m}\ddot{\theta}_2 + \frac{m_3 x_3}{m}\ddot{\theta}_3 - \frac{m_1 x_1}{m}\ddot{\theta}_1 + x_2\ddot{\theta}_2 + x_1\ddot{\theta}_1\right)$$
$$+m_3 x_2'\left(\frac{m_3 x_2' - m_1 x_2}{m}\ddot{\theta}_2 + \frac{m_3 x_3}{m}\ddot{\theta}_3 - \frac{m_1 x_1}{m}\ddot{\theta}_1 - x_2'\ddot{\theta}_2 - x_3\ddot{\theta}_3\right)$$
$$-k_2\left(\theta_2 - \theta_3\right) + k_1\left(\theta_1 - \theta_2\right) = I_2\ddot{\theta}_2 \tag{9}$$

$$m_3 x_3\left(\frac{m_3 x_2' - m_1 x_2}{m}\ddot{\theta}_2 + \frac{m_3 x_3}{m}\ddot{\theta}_3 - \frac{m_1 x_1}{m}\ddot{\theta}_1 - x_2'\ddot{\theta}_2 - x_3\ddot{\theta}_3\right)$$
$$+k_2\left(\theta_2 - \theta_3\right) = I_3\ddot{\theta}_3 \tag{10}$$

Substituting $\ddot{\theta}_i = -\omega_n^2\theta_i$ into equations (8), (9), (10) and reducing into matrix form,

$$\begin{bmatrix} a_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & a_{23} \\ a_{31} & a_{32} & a_{33} \end{bmatrix}\begin{pmatrix} \theta_1 \\ \theta_2 \\ \theta_3 \end{pmatrix} = 0 \tag{11}$$

where

ω_n = natural frequency

$$a_{11} = \frac{m_1 m_3 x_3 x_1}{m}\omega_n^2$$

$$a_{12} = m_3 x_3\left(\frac{m_1 x_2 - m_3 x_2'}{m} - x_2'\right)\omega_n^2 + k_2$$

$$a_{13} = \left(m_3\left(\frac{m_2 + m_1}{m}\right)x_3^2 + I_3\right)\omega_n^2 - k_2$$

$$a_{21} = \left(\frac{m_1 x_1^2}{m}\left(m_2 + m_3\right) + I_1\right)\omega_n^2 - k_1$$

$$a_{22} = m_1 x_1\left(\frac{m_3 x_2' - m_1 x_2}{m} + x_2\right)\omega_n^2 + k_1$$

$$a_{23} = \frac{m_1 m_3 x_3 x_1 m}{m}\omega_n^2$$

$$a_{31} = \left(\frac{m_1 m_3 x_2' x_1}{m} - \frac{m_1^2 x_2 x_1}{m} + m_1 x_2 x_1\right)\omega_n^2 + k_1$$

$$\left(m_1 x_2\left(\frac{m_3 x_2' - m_1 x_2}{m} + x_2\right) + I_2\right)\omega_n^2$$

$$a_{32} = -\left(m_3 x_2' \frac{m_3 x_2' - m_1 x_2}{m} - x_2'\right)\omega_n^2 - (k_2 + k_1)$$

$$a_{33} = \left(\frac{m_1 m_3 x_2 x_3}{m} - \frac{m_3^2 x_2' x_3}{m} + m_3 x_2' x_3\right)\omega_n^2 + k_2$$

Evaluating the determinant of the matrix defined in (11) and excluding the solution $\omega_n = 0$ (for rigid body motion) leads to the theoretical evaluation of the natural frequencies of the three degree of freedom spring mass system for which the determinant is zero. The total mass, centre of mass and radius of gyration of each hull segment was determined by physical measurements and formed part of the input parameters to the theoretical model. The theoretical evaluation of the wet flexural mode response of the segmented model considered the total mass of each hull segment including the measured mass and the theoretical added water mass. The added water mass was estimated using the mass of a semi-circle of water with a diameter equal to the beam of the waterline at each section along the length of the catamaran model demihull (Newman, 1977). The total mass moments of inertia of the catamaran model were evaluated by integrating the hull mass moments of inertia and the added water mass moments of inertia along the length of the catamaran model demihull, given by the integral

$$I = \int_L \left(\frac{dm}{dx}\right) x^2 dx. \qquad (12)$$

The results of the integration led to values for the total mass moments of inertia and the centre of mass of each hull segment. The stiffness was evaluated experimentally by measuring the flexibility and effective stiffness of each torsion spring on the physical model. Following the input of the measured parameters, the whipping flexural frequency was determined when the determinant of the matrix was equal to zero, Det[A] = 0 (equation 11). Figure 3 shows the results of the matrix determinant as a function of frequency for the evaluation of the theoretical flexural mode response of the segmented model.

It is observed from the results shown in Figure 3 that the natural frequencies of the three degree of freedom system are 13.65 Hz and 29.97 Hz and represent the first and second modes of longitudinal vibration. Based on the matrix derived in (11), the relative angular deflections of the catamaran model hull segments and vibration modes were determined at the excitation frequencies that returned a matrix determinant equal to zero. Figure 4 shows a schematic diagram of the segmented model vibration mode shapes and angular deflections relative to the stern segment for the first two modes of longitudinal vibration.

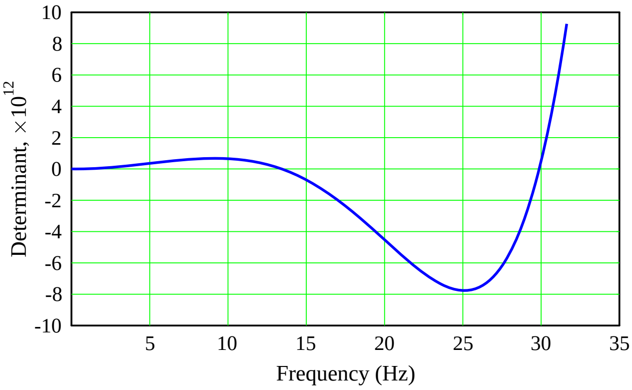

Figure 3: Matrix determinant of the three degree of freedom spring mass system for the theoretical evaluation of the flexural response frequency of the 2.5m test model.

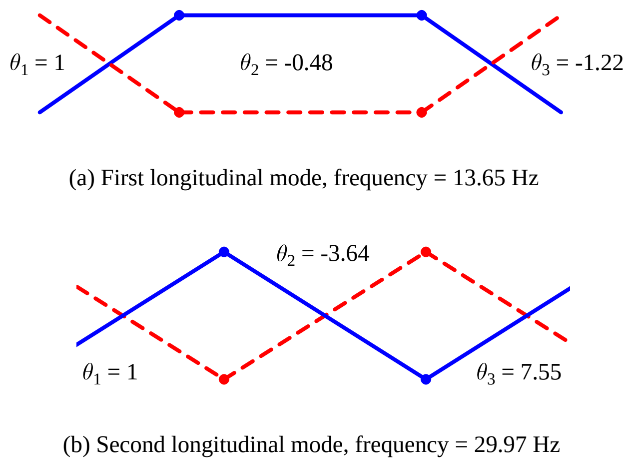

Figure 4: Three degree of freedom spring mass system mode shapes and angular deflections (relative to the stern) for the first two modes of longitudinal vibration.

4 HYDROELASTIC CATAMARAN MODEL DESIGN

The hydroelastic segmented catamaran model was developed as a geosim of the 112 m INCAT high-speed wave-piercer catamaran for the primary purpose of measuring dynamic structural loads. The experimental model was based on the segmented model concept as previously adopted by Hermundstad et al. (1995, 1999), Kapsenberg et al. (1999) and McTaggart et al. (1997). The 2.5 m catamaran model demihulls were cut at two locations along each demihull to form three rigid demihull segments. This configuration allows for the measurement of bending moments at two points along each demihull girder. The centre bow was also designed as a segmented item so as to isolate the slam loads acting on the bow of the model. Each catamaran model hull segment was constructed from carbon fibre and

Divinycell[tm] foam sandwich to produce a highly stiff and light-weight composite structure. The demihull segments were glued to an aluminium square hollow section backbone beam and each rigid hull segment was joined together by specially designed elastic hinges (torsion springs) instrumented with strain gauges to measure the wave induced response.

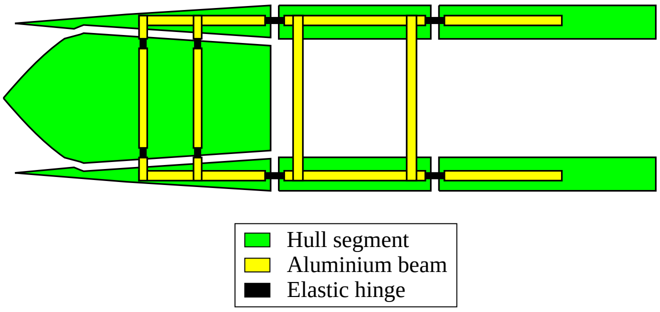

Figure 5: Schematic diagram of the 2.5 m hydroelastic segmented catamaran model.

Table 2: Main particulars of the hydroelastic segmented catamaran model.

Description	Specification
Length overall	2.5 m
Waterline length	2.3 m
Overall beam of demihull	0.129 m
Overall beam of model	0.68 m
Displacement	27.43 kg
Vertical centre of gravity (from keel)	0.091 m
Longitudinal centre of gravity (from AP)	0.948 m
Pitch radius of gyration	0.64 m

The demihull elastic hinges were made interchangeable to allow for the variation of the stiffness and whipping frequency of the segmented model. The port and starboard midship demihull segments were joined by transverse beams and the centre bow was attached to the forward demihulls by transverse beams also instrumented with strain gauges for the purpose of measuring centre bow slam loads. A photograph of the hydroleastic segmented catamaran model is shown in Figure 1 and a schematic diagram is presented in Figure 5. The model main particulars are shown in Table 2.

5 MODEL EXPERIMENTS

The hydroelastic segmented catamaran model tests were carried out at the towing tank facility of the Australian Maritime College in Launceston, Tasmania. Vibration experiments were undertaken in both air and water to investigate the dry and wet modes of longitudinal vibration. The strain gauges mounted on the elastic hinges of the model recorded the differential strain on the top and bottom surfaces of the elastic link beam element at each of the connection points along the catamaran model demihull. The strain gauge analog signals were acquired at a sampling rate of 100 Hz by a National Instruments CompactRIO running Labview FPGA and were later transferred via Ethernet to a laptop PC running Labview. Vibration tests were also performed in the towing tank at speed in calm water to investigate the effect of forward speed on both the frequency and damping of the response.

Wet and dry whipping experiments were undertaken to investigate the influences of effective stiffness and model mass on the whipping frequency of the vibratory response at zero forward speed. Dry vibration tests were performed by suspending the model in air using long, soft elastic straps and applying impulse loads by hand to the centre bow of the model. A similar process was followed for all wet vibration experiments with the model located unrestrained in still water. The whipping flexural responses were observed clearly during all tests undertaken on the catamaran model. The differential strain measured at each of the elastic hinges showed a decaying vibratory response. This was similar to the whipping phenomena observed during full-scale trials reported by Thomas et al. (2003c) and also as observed in the segmented model tests undertaken by Lavroff et al. (2006) and Dessi et al. (2005). Figure 6 shows the results of a typical strain gauge response during dry whipping experiments in air. Spectral analysis was performed on the whipping data gathered to identify the natural frequency of the first mode of longitudinal vibration.

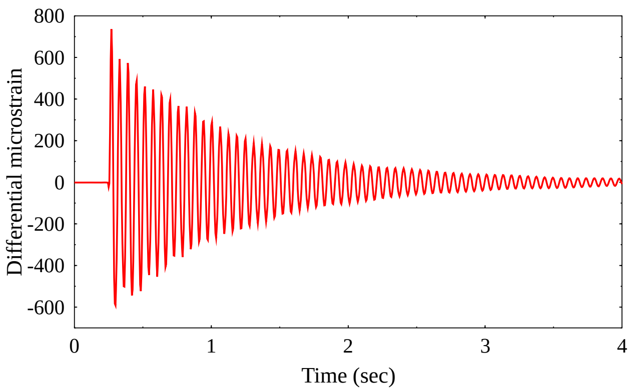

Figure 6: Dry flexural mode response of the hydroelastic segmented catamaran model during vibration experiments undertaken in air.

A Fast Fourier Transform (FFT) algorithm was used to determine the power spectral density of the response and to identify the peak frequency within a resolution of 0.008 Hz.

Figure 7 shows the power spectral density results of a FFT performed on a set of raw strain gauge data for vibration tests undertaken in still water with an elastic hinge effective stiffness of 7,162 Nm/rad.

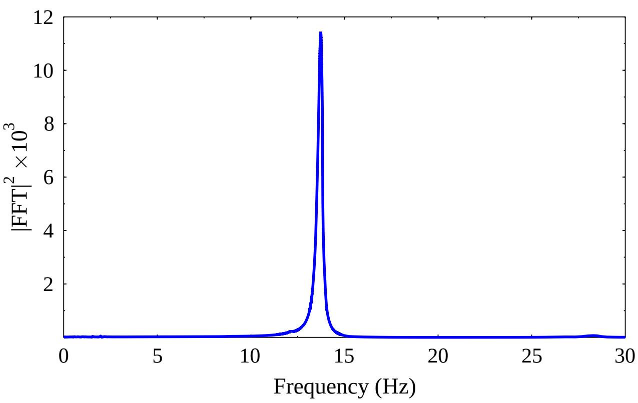

Figure 7: Power spectral analysis used to identify the natural frequency of the first mode of longitudinal vibration.

The effective stiffness of the demihull backbone was varied to study the effects on the dry and wet flexural mode response of the hydroelastic segmented catamaran model. Three elastic hinges of varying stiffness were interchanged to vary the stiffness of the demihull backbone beam. Bending tests performed on the model had confirmed that the measured stiffness of the hull and the backbone combined beam were considerably less than the theoretical stiffness of the elastic hinge as described in previous studies conducted by Holloway et al. (2006).

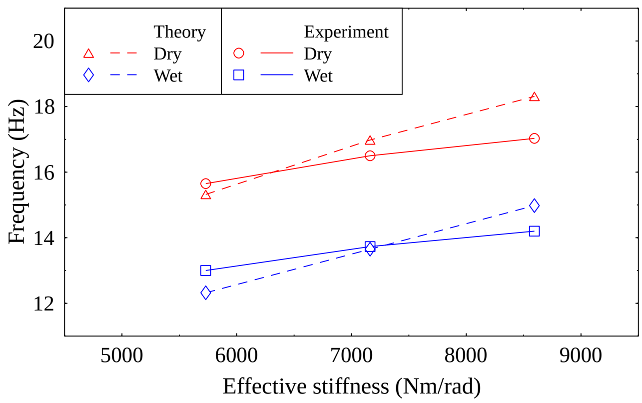

Figure 8: Experimental and theoretical whipping frequencies as a function of the effective stiffness of the catamaran model for both wet and dry tests at zero speed.

Figure 8 shows the results of both the experimental and theoretical whipping frequencies as a function of the effective stiffness for both wet and dry tests. It is evident from the results presented in Figure 8 that increases in the effective stiffness of the model corresponded with increases in the flexural response frequency. By variation of the model stiffness the modal frequency was adjusted until the target wet modal frequency of 13.65 Hz was achieved, this then simulating the first longitudinal mode of vibration of the full-scale INCAT 112m catamaran vessel. It is evident that the theoretical model provided good correlation with experimental measurements and that the added water mass causes a significant reduction in the frequency between dry and wet modes of vibration. This result reinforces the validity of the theoretical model and in particular the effectiveness of the added mass model that is commonly applied in these forms of analysis (Holloway et. al., 2004).

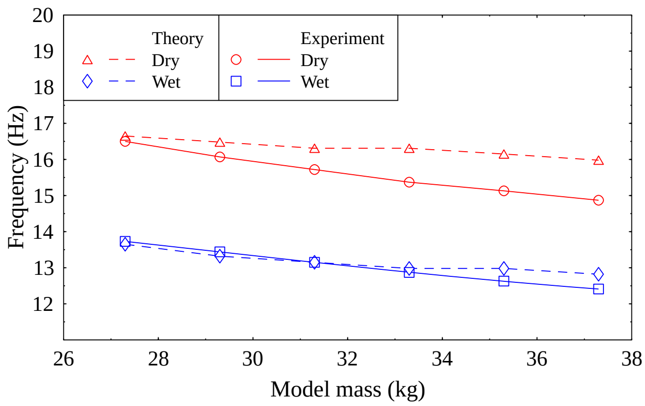

Figure 9: Experimental and theoretical whipping frequencies as a function of the catamaran model mass for both wet and dry tests at zero speed at an effective stiffness of 7,162 Nm/rad.

The catamaran model total mass was increased to investigate the effect of mass on the whipping frequency of the response. Figure 9 displays the frequency results as a function of increasing model mass (mass was added at the LCG). The theory and experimental results once again show good correlation with an overall reduction in frequency as a function of increasing mass. The influence of mass on the response frequency of the model is also evident in the data measured on full-scale catamaran vessels reported by Thomas (2003d). Increases in model mass from 27.43 kg to 37.43 kg reduced the measured wet natural frequency from 13.65 Hz to 12.82 Hz. However, reduction of the effective stiffness from 8,954 Nm/rad to 5,730 Nm/rad had a greater impact on the response, resulting in a natural frequency reduction from 14.95 Hz to 12.32 Hz. The damping ratio of the catamaran model flexural response remained relatively unaffected by these parameter variations. The damping ratio of the response was calculated in terms of the overall equivalent viscous damping ratio as defined in Rao (1995)

$$\zeta = \frac{\ln\left(\frac{x_1}{x_{m+1}}\right)}{2\pi m} \qquad \text{for } \zeta << 1, \quad (13)$$

where x_1 and x_{m+1} represent the amplitudes of the signal at times t_1 and t_{m+1} separated by m cycles. Figures 10 and 11 show the experimental results for damping ratio as a function of the effective stiffness and model mass. We see that there is negligible variation of the damping ratio for both wet and dry tests at zero speed.

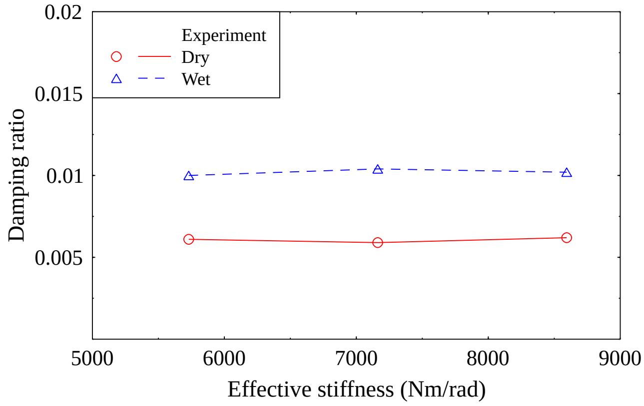

Figure 10: Catamaran model experimental damping ratio as a function of the effective stiffness for both wet and dry tests at zero speed.

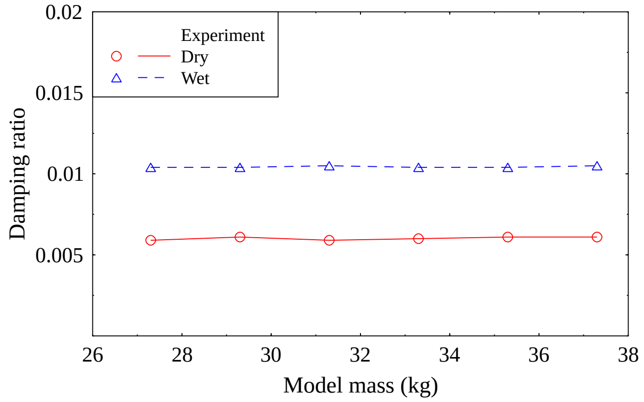

Figure 11: Catamaran model experimental damping ratio as a function of the model mass at an effective stiffness of 7,162 Nm/rad for both wet and dry tests at zero speed.

The wet damping ratio of the catamaran model was on average 0.01 and was found to be significantly less than the full-scale catamaran average damping ratio of 0.03 (Thomas 2003d, at zero speed). This suggests that additional damping effects are present on the full-scale catamaran and may be due to other factors, such as the internal fit out.

The forward speed effect on the frequency and damping ratio of the hydroelastic segmented catamaran model were identified during towing tank tests in calm water. The voids separating the rigid hull segments were tested as open voids (water filled) and with latex seals (dry voids) to study the influence of the void condition on the modal response. Figures 12 and 13 show the frequency and damping ratio as a function of increasing Froude number for test cases with open voids and with latex seals.

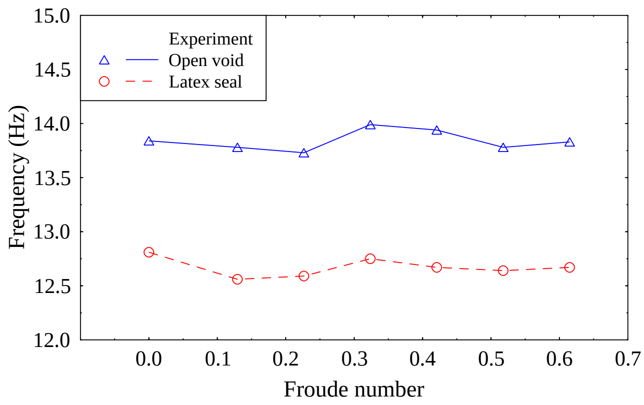

Figure 12: Catamaran model whipping frequency in calm water with increasing Froude number at an effective stiffness of 7,162 Nm/rad.

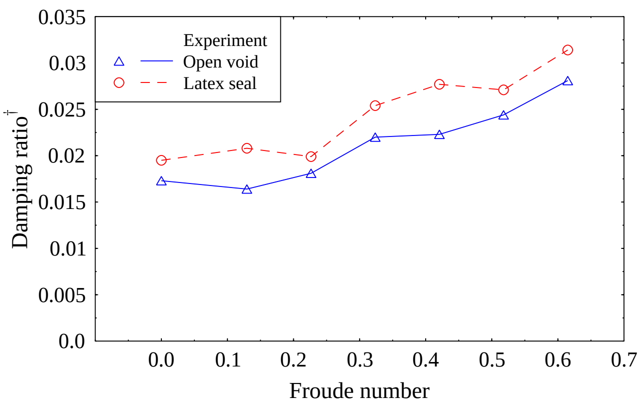

Figure 13: Catamaran model damping ratio in calm water with increasing Froude number at an effective stiffness of 7,162 Nm/rad. †Increases in zero speed damping ratio due to introduction of tow post bearing friction.

It is seen from Figure 12 that the flexural frequency remained relatively constant as function of increasing Froude number for both cases, with open voids and with latex seals. The latex seals caused a reduction in the overall response frequency and it thus appears that the presence of water in the voids increases the effective bending stiffness. In contrast, the segmented model tests undertaken by Dessi et al. (2005) showed an increase in the natural frequency with increasing forward speed, whereas Riska et al. (1994) observed a decrease in the natural frequency with increasing forward speed. The results for the damping ratio are shown in Figure 13 and we see that there is a far greater sensitivity to forward speed. This speed effect was also observed by Dessi et al. (2005). Both the latex and the open void results clearly showed an increase in damping ratio with increasing forward speed. The latex seal produced the highest damping ratio at all speeds. By comparison the damping ratio measured on full-scale vessels (Thomas 2003d) was found to remain relatively constant with increasing forward speed. However, the damping ratio of the 2.5m test model at the highest speed was similar to the damping ratio observed in the full-scale trails (~0.03).

6 CONCLUSIONS

The results of this investigation have demonstrated the parameters that affect the whipping flexural modal response of a high-speed catamaran model subject to slamming. Based on the full-scale vessel trials the first longitudinal mode of vibration was represented at model scale so as to simulate the whipping response of the full-scale vessel. A three degree of freedom theoretical model was developed to predict the first and second modes of longitudinal vibration of the catamaran model using an added water mass approximation. Whipping experiments undertaken on the catamaran model demonstrated the effects of stiffness and mass on the flexural response frequency and the theoretical model gave good correlation with the experimental data. The response damping of the catamaran model at zero speed showed negligible effect of the effective stiffness and model mass on the damping ratio. Towing tank tests were undertaken in clam water to determine the effect of forward speed on the whipping response of the model. The whipping frequency remained relatively constant and the damping ratio increased significantly as the forward speed increased. At the highest Froude number tested the damping ratio was similar to that observed in full scale trials. A basis has thus been established for future research by model testing to establish global wave loads and wave slamming loads.

7 ACKNOWLEDGEMENTS

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REFERENCES

Dessi, D., Mariani, R., Coppottelli, G., Rimondi, M. (2005). 'Experimental identification of wet bending modes with segmented model tests'. 8th International Conference on Fast Sea Transportation, FAST'05.

Hermundstad, O.A., Aarsnes, J.V., Moan, T. (1995). 'Hydroelastic Analysis of a Flexible Catamaran and Comparison with Experiments'. 3rd International Conference on Fast Sea Transportation, FAST '95.

Hermundstad, O.A., Aarsnes, J.V., Moan, T. (1999). 'Linear Hydroelastic Analysis of High-Speed Catamarans and Monohulls'. Journal of Ship Research, **43 (1)**, pp.48-63.

Kapsenberg, G. K., Brizzolara, S. (1999). 'Hydro-elastic effects of bow flare slamming on a fast monohull'. 5th International Conference on Fast Sea Transportation, FAST '99.

Holloway, D.S., Thomas, G., Davis, M.R. (2004). 'Added mass of whipping modes for ships at high Froude number by a free surface boundary element method coupled with strip theory'. Australian and New Zealand Industrial and Applied Mathematics Journal, **45** (E), pp.C831-C844.

Holloway, D.S., Davis, M.R., Lavroff, J. (2006). 'Global structural design of hydroelastic models'. 5th International Conference on High Performance Marine Vehicles, HIPER '06.

Lavroff, J., Davis, M.R., Holloway, D.S., Thomas, G. (2006). 'Experimental Analysis of the Wet Flexural Mode Response of an NPL 6A Hydroelastic Segmented Model'. 5th International Conference on High Performance Marine Vehicles, HIPER '06.

McTaggart, K., Datta, I., Stirling, A., Gibson, S., Glen, I. (1997). 'Motions and Loads of a Hydroelastic Frigate Model in Severe Seas'. Transactions of the Society of Naval Architects and Marine Engineers **105**, pp.427-453.

Newman, J. N. (1977). 'Marine Hydrodynamics'. MIT Press.

Økland, O.D., Moan, T., Aarsnes, J.V. (1998). 'Structural response in large twin hull vessels exposed to severe wet deck slamming'. Elsevier Science pp.69-78.

Rao, S. S. (1995). 'Mechanical Vibrations'. Addison-Wesley, 3rd edition, pp.135-137.

Riska, K., Kukkanen, T. (1994). 'Speed dependence of the natural modes of an elastically scaled ship model'. Hydroelasticity in Marine Technology, pp.157-168.

Thomas, G., Davis, M.R., Holloway, D.S., Roberts, T.J. (2003a). 'The whipping vibration of large high-speed catamarans'. Transactions of the Royal Institution of Naval Architects, **145** (A4), pp.13-29.

Thomas, G., Davis, M.R., Holloway, D.S., Roberts, T.J. (2003b). 'Transient Dynamic Slam Response of Large High-speed Catamarans'. 7th International Conference on Fast Sea Transportation, FAST '03.

Thomas, G., Davis, M.R., Holloway, D.S., Watson, N.L., Roberts, T.J. (2003c). 'Slamming Response of a Large High-Speed Wave-Piercer Catamaran'. Marine Technology, **40** (2), pp.126-140.

Thomas, G. (2003d). 'Wave Slam Response of Large High-Speed Catamarans'. PhD thesis, University of Tasmania, Australia.