TRANSIENT DYNAMIC SLAM RESPONSE OF LARGE HIGH SPEED CATAMARANS

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SUMMARY

In order to optimise the structural design of lightweight high speed vessels knowledge is required of the effect of sea loads on their structure. Wet deck slam events are of particular importance for high speed catamarans. This paper reports on the use of dynamic finite element analysis to investigate the transient response of a large high speed catamaran to an extreme asymmetric wet deck slam. Such an extreme slam occurred whilst extensive full scale hull stress, motion and wave measurements were being conducted on a 96m catamaran. Dynamic finite element analysis, including the fluid-structure interaction, has been utilised to develop an extreme slam dynamic load case which may be utilised in the design process. Results from this analysis are compared with the stress levels obtained through applying the Det Norske Veritas (DNV) longitudinal bending moment (sagging) rule load case.

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1. INTRODUCTION

Large, fast, lightweight vessels have recently undergone rapid development in order to satisfy the high speed sea transportation requirements of both commercial and military applications. For the structural design of such vessels to be optimised knowledge is required of the effect of sea loads on their structures [1, 2, 3, 4]. Severe wet deck slam events, which occur when the vessel's motion causes an impact between the cross deck structure and the water surface when operating in large waves, are of particular importance for high speed catamarans [5]. As well as causing high local bow pressures, such a slam can impart a large global load onto a vessel's structure. The dynamic response of the structure, observed as a shudder on-board the vessel after a slam, is known as whipping and can have significant implications with respect to the fatigue life of the structure [6].

Finite element (FE) modelling has become a standard tool for naval architects to conduct structural analyses of ship's structures [7]. Traditionally quasi-static analyses have been utilised during the design process [8, 9], whereby the true dynamic nature of the wave and slam impact loading is neglected. However a transient dynamic FE analysis may be utilised to more accurately model the time-varying loading due to slamming. Pegg et al. [10] used pressure loading time history data, measured during sea trials, in a dynamic FE analysis of the bow section of a slow speed monohull. More recently, a localised dynamic FE analysis was conducted by Rothe et al. [11] on the wet deck structure of a catamaran. Local pressures measured during typical wet deck immersions were used to evaluate local stresses at a variety of hot spot locations. However no full scale strain gauge data were available for the purposes of comparison. The dynamic FE analysis of a full vessel was carried out by Murray et al. [12] to investigate the dynamic response of a vessel to ramming ice floes. A vessel ramming an ice floe produces a similar structural response to an impact slam, although the impact loading is somewhat slower than for a slam, with a time of approximately 0.5 seconds to reach maximum load from the initial time of impact (compared with approximately 0.2 seconds for a large slam event). The response was calculated for a number of ramming cases using loads measured in full scale tests. In contrast to the other dynamic FE analyses cited above, the effect of the surrounding fluid was included in their analysis by accounting for its added mass. The calculated bending stresses for the vessel showed reasonable agreement with measured responses. Various studies have been conducted seeking to relate the measured stresses on-board vessels with those predicted by FE analysis [13, 14]. Whilst good agreement has been achieved, all these analyses were conducted utilising a quasi-static wave loading condition.

The derivation of an extreme slam load case, based on the preliminary assumption of a symmetric slamming event, for a 96m Incat catamaran was presented by Thomas et al. [15] at FAST '01. Strain gauge data from full scale slam events was used, in conjunction with a refined FE model, to develop a slam loading scenario for structural design purposes.

This paper reports on an extension of this method to investigate the transient response of a large high speed catamaran to an extreme slam, with the asymmetry of the impact being accounted for. This extreme slam, which caused structural damage, occurred whilst extensive full scale hull stress, motion and wave measurements were being conducted on the vessel during regular ferry services. Dynamic FE analysis, including the fluid-structure interaction, has been utilised to model the asymmetric impact loading. The hydrodynamic added mass of the surrounding fluid was calculated using a two dimensional panel method. The magnitude of the underlying global wave force was found from a wave-induced load model, whilst the impact load was derived from strain gauge measurements conducted on the catamaran ferry.

The stress levels from the full scale results and the FE analysis for this extreme event were then compared with the stress levels obtained through applying the Det Norske Veritas (DNV) longitudinal bending moment (sagging) rule load case to the same vessel.

2. REFERENCE VESSEL

The vessel used in this study was Incat Hull 050, a 96m high speed aluminium ferry. This ferry is a wave-piercer catamaran with a prominent centre bow, as pictured in Fig. 1. The principal parameters of the vessel are shown in Table 1.

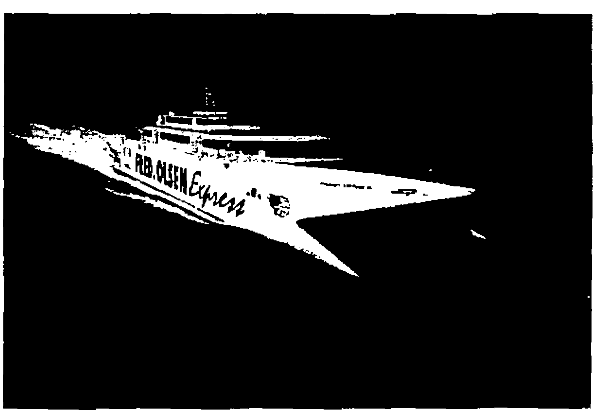

Figure 1: Incat 96m catamaran

Hull Parameter	Dimension
Length waterline	86.0m
Length overall	96.0m
Draft	3.7m
Beam overall	26.6m
Hull beam	4.5m
Deadweight	800 tonnes
Speed, fully loaded condition	38+ knots
Main Engines	Four CAT 3618 marine diesel engines. 7200kW each @ 1050rpm
Propulsion	4 Lips 150 D water jets

Table 1: Hull 050 Principal Parameters

3. EXTREME SLAM EVENT

On the 21st of November 1999 the reference vessel was travelling from Picton to Wellington, across Cook Strait in New Zealand, into a large southerly swell. Full scale wave, motion and hull stress measurements were being taken on board at the time. The measurement system comprised 8 strain gauges located throughout the vessel, see Fig. 2 and Table 2. A TSK on board radar based wave sensor was utilised to give readings of instantaneous wave height. A tri-axial accelerometer was fitted close to the centre of gravity of the vessel to measure heave, surge and sway accelerations. The speed of the vessel was also measured using an on board GPS. Further details of the on board data acquisition system may be found in Thomas et al. [15].

The vessel experienced an extreme slam event which caused some external plate buckling and distortion to several internal frames as shown in Fig. 3. On board weather observations were recorded at the time of the slam and are given in Table 3. The structural stresses were measured, along with the incident wave details and the vessel accelerations. The rapid application of load on the vessel due to centrebow archway impact with the water may be clearly seen in the extreme slam event strain gauge raw data in Fig. 4.

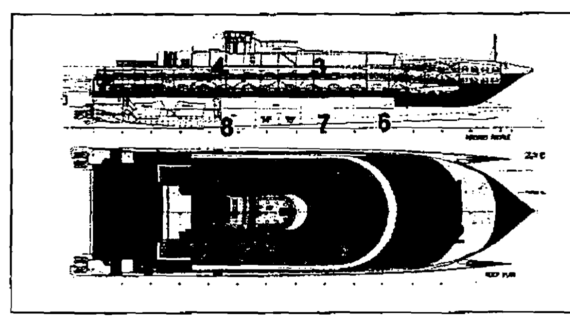

Figure 2: Strain Gauge Locations

	Location
1	Port lower steel post at fr. 63
2	Starboard lower steel post at fr. 63
3	Starboard portal top cross brace at fr. 41
4	Starboard portal top cross brace at fr. 23
5	Starboard steel post on vehicle deck
6	Starboard keel at fr. 49.5
7	Starboard keel at fr. 40.5
8	Starboard keel at fr. 24.5
Note: There are 75 frames in total, numbered from the transom at a spacing of 1200mm	

Table 2: Strain Gauge Locations

This fast increase in load is more visible on the structural members closer to the bow of the vessel and hence closer to the point of slam impact; the asymmetric nature of the slam event can be seen with the forward starboard steel post experiencing significantly larger stresses than the port forward steel post. It was determined from the data records that the wave height for the extreme slam event was approximately 5m and the encounter wave length 80m.

Parameter	Observed Value
Beaufort Sea Scale	4
Beaufort Swell Scale	SSE 5
Heading Direction	Waves 40 degrees on starboard bow
Significant Wave Height	3.7m
Vessel Speed	Engine 700rpm and 15 knots

Table 3: Slam Event On-board Observations

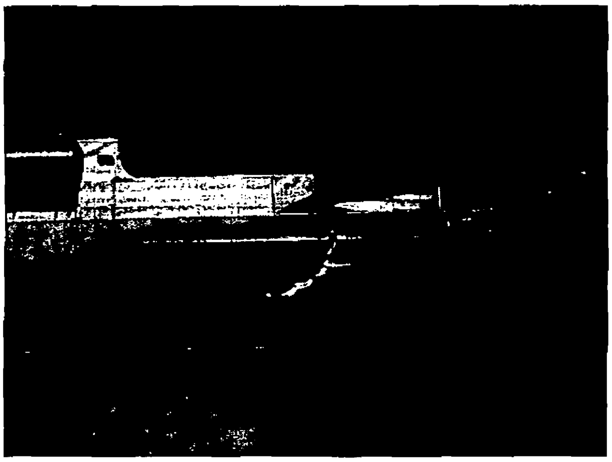

Figure 3: Hull 050 Damage from Extreme Slam Event - External Plating at Frame 51. View showing Buckled External Plating and Sponson

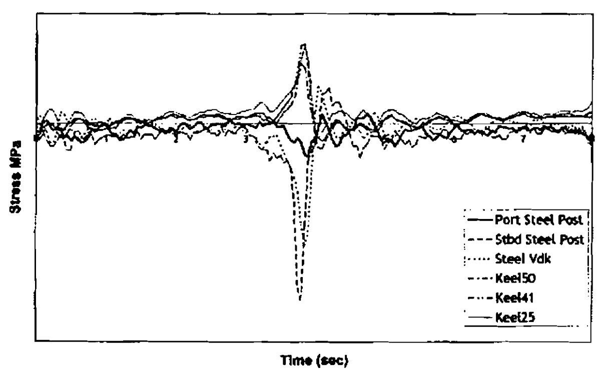

Figure 4: Extreme Slam Event Raw Strain Gauge Data

4. DYNAMIC STRUTURAL ANALYSIS

FE dynamic analysis differs from static analysis in two basic aspects. Firstly, dynamic loads are applied as a function of time and secondly, this time-varying load application induces time-varying responses (displacements, velocities, accelerations, forces and stresses). Whilst these time-varying characteristics make dynamic analysis more complicated it also provides a more realistic solution than static analysis for cases such as a slamming event which is a highly dynamic scenario.

The development of a dynamic extreme slam load case allows the dynamic structural response of the vessel to a slam to be simulated more realistically. The results from the dynamic analysis may therefore be utilised in the design process to aid structural optimisation.

4.1 DYNAMIC FE ANALYSIS THEORY

In a direct transient response analysis the structural response is calculated by solving a set of coupled equations using direct numerical integration [16]. The equation of motion in matrix form is as follows:

$$[M]\{\ddot{u}(t)\}+[B]\{\dot{u}(t)\}+[K]\{u(t)\}=\{P(t)\} \quad [1]$$

where: $[M]$ = mass matrix
$[B]$ = damping matrix
[K] = stiffness matrix
$P(t)$ = time varying force
u = displacement

The fundamental structural response (displacement) may then be solved at discrete times with a fixed integration time step Δt. The damping matrix $[B]$ represents the energy dissipation characteristics of the structure, with the structural damping being included by means of converting it to an equivalent viscous damping.

4.2 FE MODEL

The PATRAN/NASTRAN FE model of Hull 050 was constructed by importing the geometry from CADKEY, see Fig. 5.

The model consisted of predominantly plate and bar elements with the exception of laminate elements used to model the honeycomb material in the mezzanine ramps. The model included the superstructure which was connected to the main hull via elements modelling the connecting rubber mountings.

4.3 FLUID-STRUCTURE INTERACTION

For a dynamic FE analysis the added mass of the surrounding fluid needed to be included. The added mass represents the effective inertia of the water surrounding the oscillating hull, and may be defined as the component of force in phase with the body's acceleration exerted by the hull on the water for a unit amplitude acceleration of the hull. It was calculated utilising a steady periodic Green function panel method proposed by Doctors [17] and further developed by Holloway [18].

The method of Salvesen, Tuck and Faltinsen [19] has been generalised to express the sectional added mass with forward speed in terms of the local hull deflection, slope and curvature due to hull flexure. Assuming negligible damping it can be concluded that there is no phase difference between the deflection, slope and curvature. This gives a forward speed added mass of

$$a_u = a\left(1 - \frac{U^2}{\omega^2}\frac{\eta''}{\eta}\right) + \frac{U}{\omega^2}\left\{\frac{db}{dx} + \frac{\eta'}{\eta}\left(2b - U\frac{da}{dx}\right)\right\} \quad [2]$$

where a is the zero speed added mass and b the zero speed damping. If η is the local vertical displacement amplitude of a point on the hull, $\eta' = d\eta/dx$ and $\eta'' = d^2\eta/dx^2$.

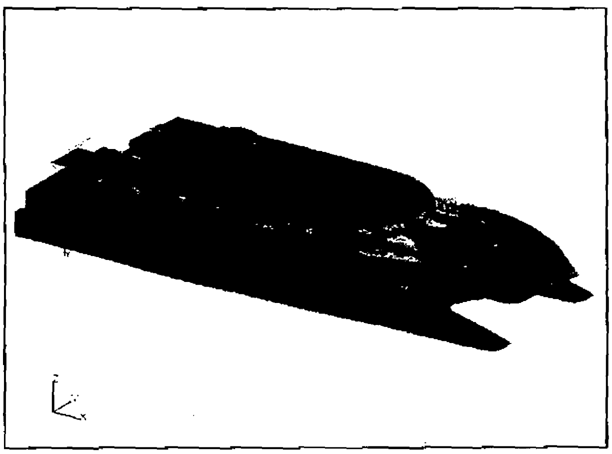

Figure 5: FE Model of Hull 050

4.4 WAVE-INDUCED LOAD MODEL

In order to provide global wave load information for the FE model, a method for estimating the wave-induced load of the underlying wave that the vessel encountered during the slam event was needed. Since a simple yet effective method was required, the Froude-Krylov exciting force was utilised to determine the vessel's heave and pitch in a regular wave and consequently the vessel loading. The Froude-Krylov force results from the integration over the vessel's surface of the underwater pressure, assuming that the presence of the hull has no effect on the waves [20]. This approximation uses only the incident wave potential in estimating the total wave exciting force and the effect of wave diffraction by the body is therefore not included. The appropriateness of this approximation increases in accuracy as the incident wave wavelength increases relative to the length of the vessel [19]. The motions of the vessel are taken into account in the proposed model by including, in the weight of each section of the vessel, an additional dynamic inertia component. This component was determined from the measured full scale vertical accelerations at the longitudinal centre of gravity and forward perpendicular. That the hydrostatic portion of the wave-induced loading is dominant may be seen as the reconciliation is good when the model was tested by comparing the measured strain gauge stress readings with those predicted by FE analysis for a purely global wave loading situation [15].

The model was developed to consider regular sinusoidal waves at all heading angles. The vessel's hulls were modelled through Bonjean curves representing the immersed area of each transverse section for the local draft. The moment generated in the single hulls by the asymmetry of the wetted transverse area was neglected and the draft was considered to be equal on both sides of the same hull. However, for a non-head or non-following sea condition the draft at the same longitudinal position on each hull was different and accounted for.

The vessel is therefore balanced on the wave, for a given wave length, wave height and heading angle, with the vessel's weight forces and moments balanced by the vessel's buoyancy forces and moments. The sinkage and trim of the vessel are iteratively varied until the equilibrium position is determined. The vessel is assumed to maintain a zero heeling angle in the wave environment. Although in oblique seas a catamaran would be expected to exhibit an angle of heel, the effect on the buoyancy forces was assumed to be small. The global wave force at each frame may thus be calculated utilising the wave-induced load model.

The data from the extreme slam event was utilised to develop a dynamic asymmetric slam load case. The underlying wave loading was determined by utilising the wave-induced load model for a wave of length 80m, height 5m and heading angle 140 degrees. The vertical acceleration of the vessel was also taken into account when calculating the buoyancy forces using the acceleration levels of 1.9g measured at the LCG and 3.0g measured at the bow during the slam event. The global load time history was estimated from the encountered wave data recorded at the time of the slam, which gave an average peak to peak encounter period of approximately 5.1 seconds. The maximum global load was taken to occur at the time instant for the maximum slam load, with the global load varying sinusoidally with a period of 5.1 seconds. The global load time history utilised is shown in Fig. 6.

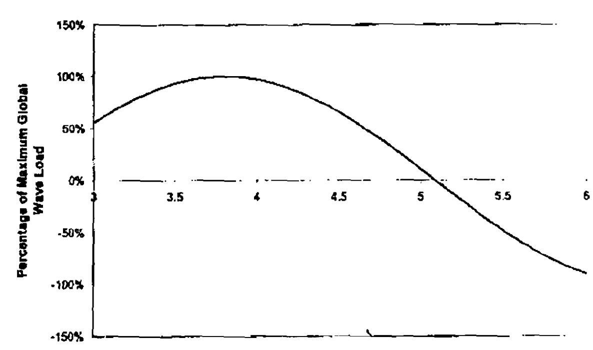

Figure 6: Global Wave Load Time History for Extreme Slam Event

4.5 IMPACT SLAM LOAD

In addition to the underlying global load, a load was required to simulate the slam impact force on the bow of the vessel. The slam load time history was derived from the strain gauge results for the forward steel post which was the gauge closest to the point of slam impact. The change in slam load with time was taken to match the change in stress with time, as shown in Fig. 7. A time step of 0.05 seconds was utilised which matched the sampling rate of the strain gauge data.
The magnitude of the extreme slam impact, and its transverse and longitudinal distribution, were systematically altered until an acceptable correlation with the maximum values of the full scale strain gauge data was achieved. The match between the FE results and the full scale results was attained when the average difference between the FE and full scale stress results was minimised. The result of the study was that a slam load of 1025 tonnes was distributed over the starboard side of the centrebow and archway to account for the impact force. The distribution of the slam impact load on the centrebow is shown in Fig. 8. Note that the frames are numbered from the transom.

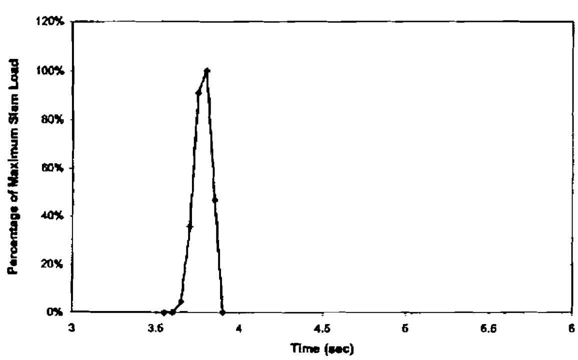

Figure 7: Slam Load Time History for Extreme Slam Event

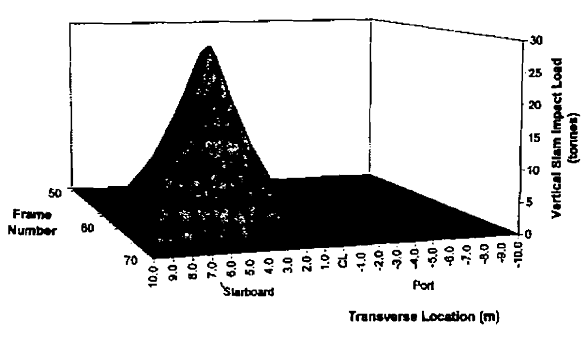

Figure 8: Distribution of Slam Impact Load for Extreme Slam Event

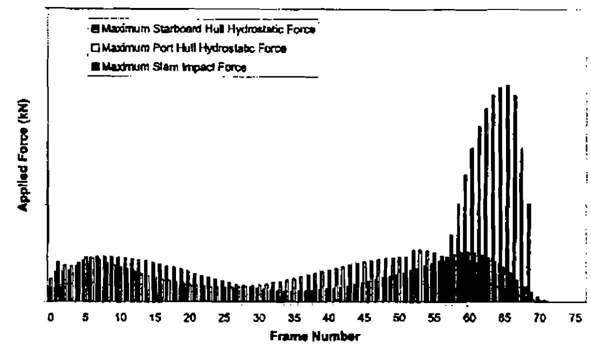

Figure 9: Longitudinal Distribution of Applied Force for Extreme Slam Event

4.6 FE ANALYSIS

The dynamic FE analysis was carried out using PATRAN/NASTRAN direct transient response dynamic analysis. For the analysis the FE model was subjected to one set of buoyancy forces distributed along each demihull (acting at 3 nodes for each frame per hull) and centrebow (acting at 4 nodes for each frame per hull) plus a vertical inertial force equivalent to that determined from the full scale data. Fig. 9 shows the relative magnitudes of the global wave load and slam impact load, and their longitudinal distributions.
A value of 0.5 was utilised for G (the overall structural damping coefficient utilised by NASTRAN which has no units) to account for the structural and hydrodynamic damping. This value was adopted after investigating the level of damping in the vessel through full scale exciter tests [21]. The frequency of interest for the conversion of overall structural damping into equivalent viscous damping, was taken as 18.85 rad/sec to match with the 1st longitudinal mode of vibration identified through modal analysis [21].

5. FE ANALYSIS RESULTS AND COMPARISON WITH FULL SCALE MEASUREMENTS

An example of the output from the FE analysis is shown in Fig. 10. The plot shows the dominance of distortion in the starboard bow region due to the slamming impact force. The detailed image of stress on the starboard side, frames 55 to 60, in Fig. 11, shows the concentration of stress in the region where damage was experienced by the vessel following the extreme slam event.

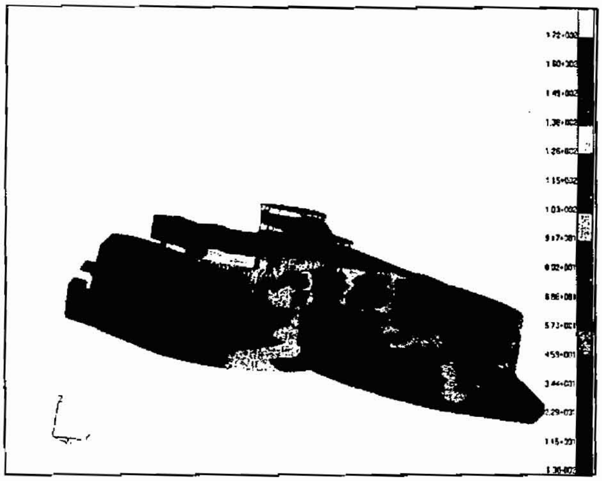

Figure 10: Exaggerated Deflection and Stress Plot for Dynamic Extreme Slam Load Case at Time of Maximum Slam Load

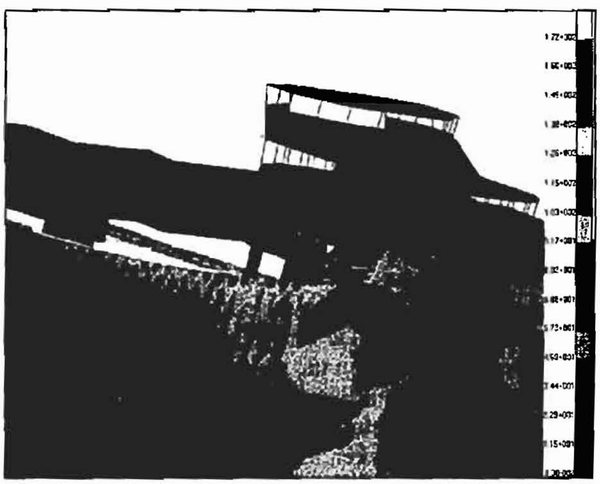

Figure 11: Stress Plot for Dynamic Extreme Slam Load Case - Starboard Side, Frames 55 to 60 at Time of Maximum Slam Load

Examples of the stress results for the strain gauges for the dynamic slam event FE analysis are shown in Figs. 12 to 14. These plots show that good correlation was achieved, as a function of time, for the initial slam impact peak stresses. This indicates that the slam time history utilised for the FE loading was realistic, therefore for this slam event it may be concluded that the slam loading took a time of 0.2 seconds to reach a maximum value, and the slam load then expired within a further 0.1 seconds. The level of damping utilised in the FE analysis gives an indication of the reduction in dynamic response of the structure during the whipping after the slam event. It should be noted that the levels of damping measured during full scale trials after slam events varied with cycle number and strain gauge location. The whipping periods shown in Figs. 12 to 14 are approximately 3 Hz, which correspond with the 1st longitudinal mode of vibration as identified previously [21].

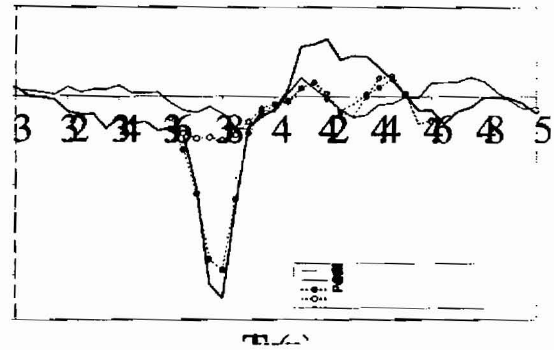

Figure 12: Time History of Steel Post Stress Results for Extreme Slam Event

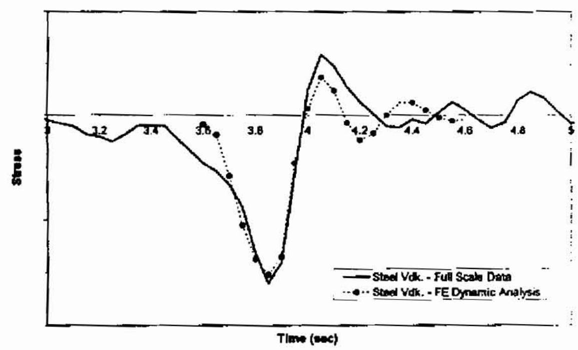

Figure 13: Time History of Steel Vehicle Deck Stress Results for Extreme Slam Event

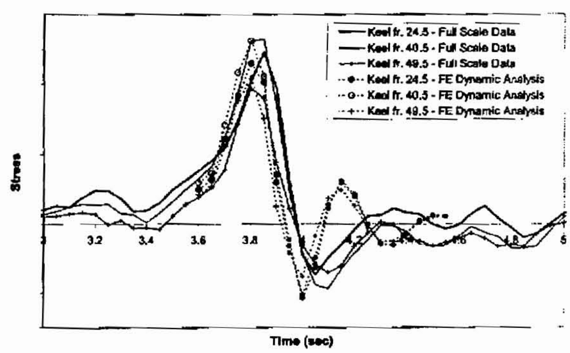

Figure 14: Time History of Keel Stress Results for Extreme Slam Event

Fig. 15 shows that good correlation was achieved for the FE dynamic analysis maximum stress results when compared with the strain gauges for the extreme slam event. The stress results were within 21.1% of the full scale measurements except for the gauge on the port steel post. The average error for all the strain gauges was 11.1% with the major discrepancy in the results being the level of stress in the port steel post. It was difficult to reduce the level of stress in this structure whilst maintaining sufficient load to retain the required stress levels at the other strain gauge locations.

Especially since the steel posts were very susceptible to the localised slam loading. It is noted that while the relative error for the port post is high, it is not significantly different from the error at other locations in absolute terms. The slam loading increased in magnitude very rapidly during a slam event, particularly in the forward region of the vessel, and since the sampling rate for these strain gauges was only 20 Hz the peak for the port steel post may have been missed which could account for the disparity in results for this location. Also, local effects, or other types of loading not considered may have contributed, for example the accuracy of the heading angle, which was an on board visual observation, and the lack of information on wave spreading may have affected the underlying global wave loading and account for a portion of the disparity in the correlation with the strain gauge results.

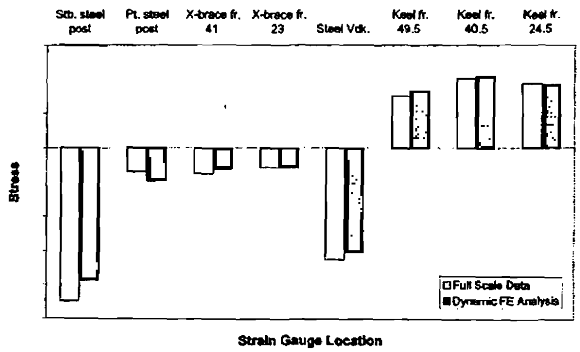

Figure 15: Comparison of Maximum FE Analysis and Full Scale Data for Dynamic Extreme Slam Impact

The primary outcome from this analysis is data on whether the design meets ultimate strength requirements when subjected to an extreme slam event. The output also includes information on the stress time history and whipping frequencies which may be important for any subsequent fatigue life estimates.

There are however a number of drawbacks of the method, including the following:

- The dynamic FE analysis method is very computationally intensive. For a large global model, computer memory difficulties may be encountered when running a dynamic analysis for a large number of small time steps.
- Realistic damping data is required for the dynamic FE analysis. This is difficult to source in the literature.

6 COMPARISON OF DYNAMIC FE EXTREME SLAM LOAD CASE WITH DNV RULES

The dynamic extreme asymmetric slam load case was compared with the DNV longitudinal bending moment (sagging) load case as prescribed by their Classification Rules [8]. Fig. 16 shows the comparison of the maximum dynamic FE analysis stress results for the extreme slam load case with the stress levels obtained through applying the DNV sag rule moment load case to the same vessel.

The comparison of stress results shows that the stress levels, for the dynamic slam load case FE analysis, were greater than the DNV sag rule moment for every location, except the steel vehicle deck bracing and port steel post. This is borne out by the maximum bending moment curves shown in Fig. 17, where the curves have been normalised by the maximum values determined from the DNV sagging rule moment, showing actual bending moments in the forward half of the vessel to be substantially higher than specified in the DNV rules.

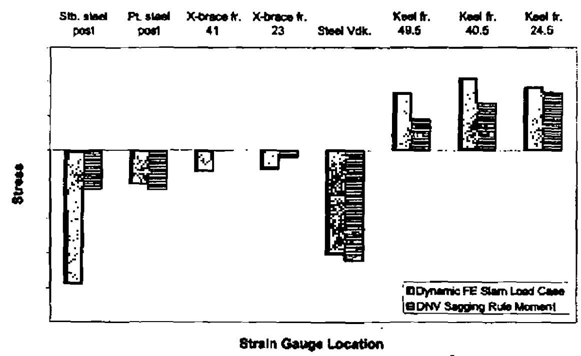

Figure 16: Comparison of Dynamic FE Analysis for Extreme Slam Load Case and DNV Sagging Rule Moment

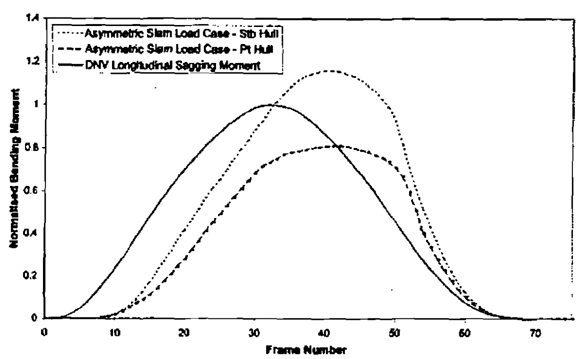

Figure 17: Comparison of Maximum Bending Moment Curves for Extreme Slam Dynamic FE Load Case and DNV Sagging Rule Moment

7 CONCLUSIONS

Dynamic finite element analysis has been utilised to investigate the transient response of a large high speed catamaran to an extreme asymmetric wet deck slam. A realistic load case for an asymmetric extreme slam event has been developed for Incat Hull 050, a 96m catamaran. This was achieved by correlating the measured strain gauge readings measured during an extreme slam event with predictions from an FE model. When compared with the DNV design sagging rule, the maximum bending moment for the extreme slam load case, applied to the vessel in the design condition, has a greater maximum value on the critical hull and its peak is further forward than for the DNV sag rule moment. This design load case is a significant improvement on the current DNV sagging load case design rule for this type of vessel.

However, the dynamic FE analysis method is very computationally intensive and realistic damping data is required. Such factors may limit the current practicality of its use.

8 ACKNOWLEDGEMENTS

This work has been partly supported by an Australian Research Council SPIRT grant.

The authors would like to acknowledge the help of their colleagues at the University of Tasmania, Revolution Design Pty. Ltd. and Incat Tasmania.

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