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Deep Water Entry of High Speed Ferry Bows

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Abstract

High speed multi-hull ferries can experience strong vertical motions near the bow when encountering heavy seas and risk bow damage due to deep entry into encountered waves. The risk of water passing over the top of the bow has been virtually eliminated in the INCAT Tasmania Wave Piercer design which incorporates a centrebow with a keel on or above the waterline. However, confinement of displaced water in the connecting archways to port and starboard of the centre bow leads to large hydrodynamic loads on the bow structure. The time dependant water entry problem for a high speed wave piercer vessel is extremely complicated owing to the relatively complicated form of the bow areas of the hulls. Determination of the hydrodynamic loads which can occur has therefore been based primarily on experimental testing at full scale and at model scale, but CFD methods are also being investigated. As the bow enters an encountered wave, water is at first displaced by the centre bow and ejected over the top of the forward ends of the demi-huills. However as the water entry event develops the rising water surface can ultimately fill the arch cross section and at that stage the hydrodynamic loads can become extremely large. Maximum loads in excess of the total displacement of the vessel (that is in excess of 2500 tonnes at full scale typically) have been observed and have durations of about 0.4 seconds at full scale.

Introduction

The hydrodynamic design of high speed ferries is influenced by two particular problems with regard to the response to encountered waves. The first is that owing to the relatively high Froude Number, $F_r = \sqrt{\frac{v}{gL}}$, which can reach 0.85 or more based on waterline length, the heave and pitch motions become large and can exceed wave height and wave slope by a factor of two or more [1,2]. The result of this is that vertical motions in the bow area become large and unsuitable for passengers who are generally not located in the deck areas near to the bow. The second problem is that in following seas high speed vessels can often overtake the moving surface waves and so become exposed to deck diving. This occurs when the bow enters the back of a wave and can cause damaging flow of large water masses over the top of the bow deck structure. In conventional single hull vessels the bow design generally incorporates strong outward flare of the bow above the waterline to prevent this, and in any case the exposure is relatively less owing to the lower Froude number of conventional vessels. For multi-hull catamaran designs bow flare is generally much less and so there is a risk of bow entry into the encountered wave and consequent structural damage.

In response to the problem of large motions near the bow in head seas a number of designs have removed bow flare in the main buoyant demi-hulls, as in the INCAT Tasmania Wave Piercer design, or else have completely submerged the demi-hull bow, as in semi-SWATH designs such as the STENA HSS (www.stenaline.nl/en/ferry/ships/stena-explorer) or US Navy X-Craft (www.globalsecurity.org/military/systems/ship/x-craft). The aim of all these designs is to reduce the water plane area near the bow and thus reduce the large motions in the bow area in head seas. In the case of the STENA HSS this has made it possible to extend the passenger cabin right to the bow. However, in achieving the aim of reduced bow vertical motions at high Froude Number, these designs have to be vertically soft near the bow. This means that when encountering the back of a following sea wave the risk of deck diving is increased and has led to significant bow damage to semi-SWATH vessels. This issue has been resolved in the INCAT Tasmania design (www.incat.com.au)by the addition of a centre bow with a keel on or above the calm water line (Figure1). This bow produces large vertical forces when encountering following seas and essentially eliminates deck diving exposure. However, in large head seas rapid water filling of the arch cross section between the centre bow and demi-hulls generates large vertical forces of rather short duration, peaking at close to the instant at which the cross section fills completely. This causes a slam event which is particular to the INCAT Tasmania Wave Piercer design and which is quite different to bottom slamming and bow flare slamming of conventional vessels.

Figure 1. INCAT Tasmania 112m Wave Piercing Catamaran, showing centre bow keel and demi-hull bows (both will be on the water line when loaded).

It is evident that the water entry of a wave piercer bow constitutes a complicated three dimensional transient problem. Whilst the slam event is relatively localised in hydrodynamic terms it is of course influenced by and indeed is largely due to the ship motions. Therefore to identify conditions under which slams occur and the severity of slams the complete dynamic response of the vessel needs to be considered. This complicates the whole problem and in the first instance leads to the adoption of full scale and physical model testing to delineate the kinematics, occurrence and severity of slamming. Once the overall ship motion kinematics are understood then there is clear potential for CFD to be applied to the one off solution for a single relatively short slam event.

Full scale trials observations of slamming

A number of vessels have been fitted with motion sensing, wave sensing and strain gauge instrumentation and over a period of years this has yielded information regarding slam events. Thomas et al [3] analysed an extreme slam on an INCAT 96 m vessel whilst in passenger service. The vessel encountered a wave on the starboard bow quarter of approximately 5.0 m height at a ship speed of 15 knots. This is a wave height greater than the usually adopted limit of 3m for passenger vessels of this class. Analysis of the records of strain gauges on the keel and the internal bow bracing structure indicated a peak load of 1300 tonnes in the starboard arch between the centre bow and starboard demi-hull, with peak vertical accelerations of 3g at the bow and 1.9 g at the centre of mass. The relative motion of bow to the water surface reached approximately 7m/sec and the maximum bending load in the starboard hull exceeded the design load case by about 20%. The vessel, of a now superseded INCAT Tasmania design, suffered some relatively minor structural damage on the bow and starboard side but was subsequently repaired and remains in service.

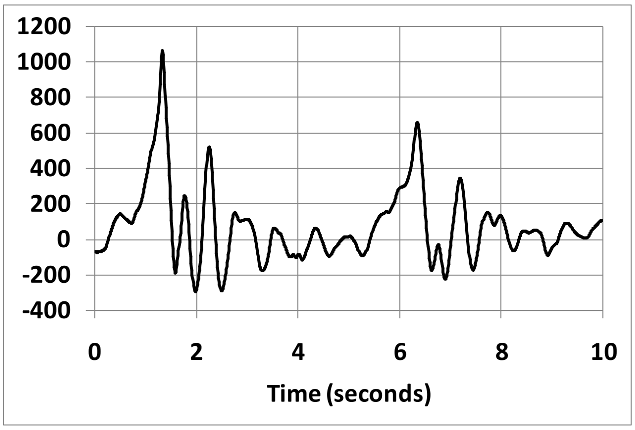


Figure 2. Strain gauge record of two wave slam events on successive waves on INCAT Tasmania 98m WP catamaran (strain gauge located on port demi-hull keel 56% overall length ahead of transom in 3.6m head sea at 20 knots speed)

A more extensive series of dedicated sea trials was undertaken by the US Navy on an INCAT 98m vessel modified for military operations [4]. The most severe dynamic responses were in head seas. Conditions for these trials were more severe, including steady cruising in waves of significant (i.e. not extreme) wave height up to 3.7m at 20 knots and 2.7m at speeds up to 35 knots. In these trials the vessel was operated at steady speed for half hour tests in an octagonal pattern so as to record responses in seas from all directions relative to the ship heading. Amin [5] analysed strain gauge records in these trials and found an extreme slam of 2200 tonnes in head seas. Figure 2 shows a strain gauge record of a gauge located on the port side demi-hull keel during these trials. The trials were so severe that slams were sometimes recorded on as many as six successive wave encounters in a wave group, the last generally being the most severe. Typically the duration of a slam event was approximately 0.4 seconds, although definition of duration is not straightforward as initial upward loading from under the bow is followed by a downward loading as the bow moves to leave the immersed condition.

The main problem with sea trials testing is that there is no systematic control of encountered wave conditions and that wave conditions are in any case somewhat random. Therefore it is not clear whether sea trials have shown the maximum hydrodynamic bow loadings which can occur. For that reason model testing has been adopted to seek to delineate maximum hydrodynamic loadings on the bow structure.

Model testing to delineate hydrodynamic slam loads

Initially drop testing of typical ship sections was carried out [6] to seek to identify slam loadings. However it was found that the loadings were unrealistically large when compared to full scale trials loadings. This was attributed to the two dimensional constraint of these tests. Similar high loadings have been found when computed by two dimensional analysis. However these two dimensional drop tests did demonstrate that slam loadings could be ameliorated to a limited extent by moving the top of the arch connecting the centre bow and the demi-hulls outboard as far as possible, this maximising the volume of the centre bow. It was concluded that three dimensional model testing in a towing tank at the correct forward speed Froude number was the only reliable way to proceed.

Figure 3. Hydro-elastic 2.5m model of INCAT Tasmania 112m Wave Piercing Catamaran, showing transverse beam, mounts for bow and segment joins covered by flexible latex tape.

It is found in full scale trials that slam durations are approximately 0.4 seconds. However it is also observed that the slam impulse excites the main longitudinal bending mode of the vessel, typically at a period of about 0.4 seconds [4,5]. Since these two time periods are almost identical it is necessary that a test model replicates the bending vibration of the hull correctly if transfer of impulse and energy from the transient hydrodynamic slam load into the structure is to be represented correctly at model scale. Complete elastic simulation of the entire ship structure is not practicable and so the approach adopted is to use a segmented 2.5m long test model, in this case broken down into three segments along the ship length as shown in Figure 3. These segments are connected by short rectangular elastic links machined from aluminium and adjusted in thickness so that the frequency of the first bending mode is 13.8 Hz, scaled in proportion to the square root of the dimensional scaling from the full scale vessel. The second mode of this three element system was at 30 Hz and this was sufficiently high that it did not become evident during testing. The ratio of damping to critical damping of the main bending mode at 13.8 Hz was found to be 0.19, fortuitously close to the full scale value of 0.2. The aluminium segment links are also fitted with strain gauges on top and bottom surfaces connected into a single strain gauge bridge so as to sense the hull bending moment in terms of the differential strain due to bending. The bow was constructed as a separate segment and mounted onto the forward hulls with axial pin mounts in the demi-hulls. Smaller elastic aluminium links incorporated into the two transverse support beams of the centre bow segment and were also fitted with top and bottom strain gauges were thus used to sense the total hydrodynamic vertical force on the centre bow. The centre bow segment extended to the position where it joined the outboard hull at a point of vertical tangency. The stiffness of these links was such that the frequency of vertical vibration for the bow on its mountings was in excess of 40Hz and so this vibration did not become evident during testing which was dominated by the main bending response at 13.8Hz.

Figure 4 shows the 2.5m model during tank testing. It can be seen that when the centre bow makes a deep entry into the encountered water surface it displaces water in the outward direction and this may pass over the top of the lower bows of the outboard demi-hulls on either side. A spray screen can be seen fitted to the model to protect instrumentation within the model but this was of very lightweight material and sustained no significant loads. Tests were carried out in regular waves up to equivalent full scale wave heights of 5.4 m.Even when the wave length and frequency corresponded to the maximum motion response displaced water did not come over the true vessel bow showing that the wave piercing design is inherently very sea worthy as it is virtually impossible to submerge the centre bow.

Figure 4. Tank testing of hydro-elastic 2.5m model of INCAT Tasmania 112m Wave Piercing Catamaran at 20 knots in 4m full scale waves showing displacement of water by centre bow over the top of the outboard demi-hull bows (lightweight splash screens have been fitted to protect instrumentation)

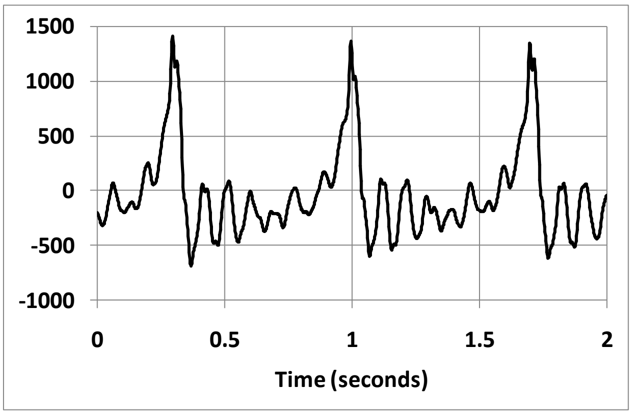


Figure 5.Record of forward port side demi-hull elastic link strain gauge of2.5m long hydro-elastic model during tank testing in 120mm wave height (5.4m full scale) at 1.5m/s (20 knot full scale), encounter frequency f_e= 1.4Hz, $\omega_e^* = 4.6 = 2\pi f_e\sqrt{L/g}$ =dimensionless encounter frequency, L=length of hull.

Figure 5 shows the response of the forward elastic link in the port side demi-hull, this essentially being a bending moment record. Comparison of this response with the full scale record shown in figure 2 (which also corresponds to a hull longitudinal vertical bending moment time record) indicates the success of the hydro-elastic model in simulating the slam response of the full scale vessel, the initial rapid rise of bending moment and subsequent whipping vibration both being well simulated. It can also be seen in figure 5 that the response on each encountered wave is effectively the same, the model reaching a regular slamming cycle in step with the encountered wave train. Subsequent analysis of the slam response could therefore be reliably based on any one of the measured slam responses during a run along the towing tank. Only during the first two or three encountered waves of the incident wave train was the motion found to be larger, sometimes to the extent that the model motion was restrained during encounter with the first few encountered waves to prevent damage due to initially large motions.

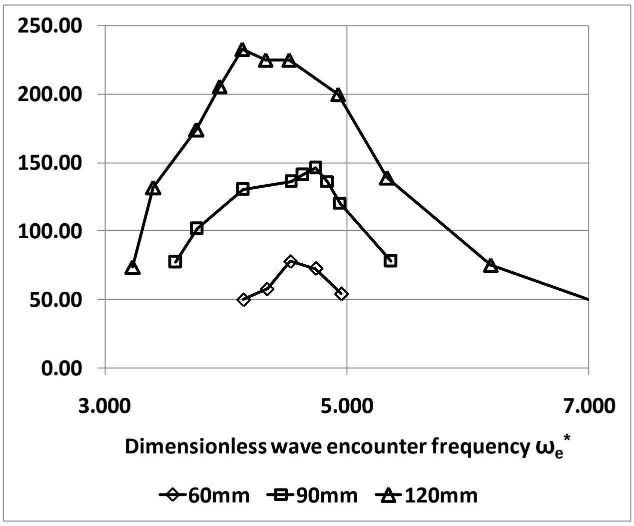
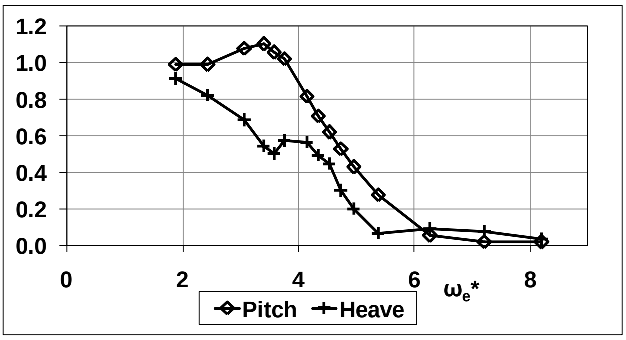
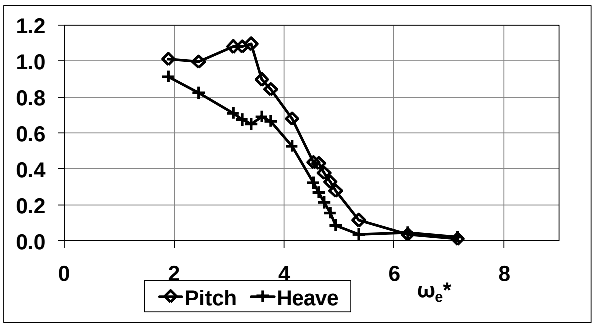


Figure 6. Peak upward slam force (N) on bow of 2.5m long hydroelastic model for various wave heights at 1.5m/s (20 knot full scale).



(a) Regular wave height 60 mm(2.7m full scale)



(a) Regular wave height 90 mm (4.0m full scale)

Figure 7. Motion responses of hydro-elastic 2.5m model of INCAT Tasmania 112m Wave Piercing Catamaran at 20 knots. Ordinate: Response amplitude operators (ratio of heave to wave height and pitch to wave slope)

Figure 6 shows the variation of maximum upward force on the centre bow during regular wave encounter in the towing tank. The maximum slam force of 240N at model scale corresponds to a force of 21.6MN at full scale, or 2200 tonnes weight. This corresponds broadly with the magnitude of the largest slams measured in sea trials. However, as figure 6 clearly shows the peak slam force increases quite rapidly with wave height. Peak slam forces equivalent to 3100 tonnes have been measured in irregular wave model testing at 4m significant wave height. It is thus apparent that because irregular wave systems occasionally contain much larger waves than the significant wave height significantly larger slam forces that those shown for a given regular wave height in figure 6 can occur in irregular wave conditions of the same nominal wave height. We see that the peak slam forces can significantly exceed the total vessel weight (i.e. its displacement).

The vessel motion responses are shown in figure 7 for two wave heights. It is evident that the most severe slamming as shown in figure 6 takes place at the wave encounter frequency for which the pitch motions are greatest, indicating that severe slamming is primarily related to the vessel vertical bow motion rather than to bow encounter with incident wave vertical motion. It is also evident when we compare Figure 7(a) and 7(b) that in larger waves the centre bow has had little effect on the overall pitch motion.

Computational fluid dynamics solution for slamming

As mentioned there is clear potential for computational fluid dynamics (CFD) to be applied to the solution of a slam event. A scale model of the 112m INCAT Tasmania has therefore been set up using the CD-Adapco STAR-CCM software (www.cd-adapco.com). Figure 8 shows a single frame from the transient solution thus obtained and Figure 9 shows the transient upward force on the centre bow. This software has been applied here without using capability to simultaneously solve the structural response by finite element analysis and so there is of course no evidence of whipping vibration of the hull. The solution here is for relatively long regular waves compared to the wavelength which gives rise to maximum slamming. Therefore the computed peak force at model scale of approximately 60 N, whilst being rather small compared to the maximum value shown in figure 6, is approximately consistent with the tank test data. Bearing in mind the small wave height and low encounter frequency of the CFD solution shown in figure 9 we see that the experimentally measured slam force for the conditions of Figure 9 would be approximately 60N, although this value was not actually tested for the physical model and extrapolation from the data of Figure 6 is difficult owing to the rapid reduction of slam force as encounter frequency is lowered. It is also evident in figure 9 that the slam event determined by CFD has a double peak and that the second peak is very sharp. At this stage only preliminary CFD work has been carried out but it is clear that CFD has the potential to give reliable estimates of slamming forces for the INCAT Tasmania Wave Piercer design. Our future tasks are to extend the CFD solution to wave encounter frequencies where there are stronger slam events, to investigate the precise form of the CFD predicted slam event and to incorporate hull flexibility by simultaneous CFD and FEA solutions of the slam event.

Conclusions

Full scale and model testing to delineate the severity of hydrodynamic bow slamming on designs of the INCAT Tasmania Wave Piercer type have shown broad consistency. Since the time scale of slam loading and the period of hull bending vibration are similar it is important that test models replicate the dynamic hydro-elastic response of the ship structure. Peak forces due to slamming are large and can exceed the vessel weight although they are of short duration and do not greatly influence the overall pitching motion of the vessel. However that is not to say that the centre bow of these designs does not play an important role in preventing immersion of the bow areas which it clearly does although on a time scale greater that the slamming time scale. Computational fluid dynamics solutions are clearly quite suitable for solving transient slam events, but more investigation is needed to determine whether CFD has the capability to reliably predict the most severe slamming events.

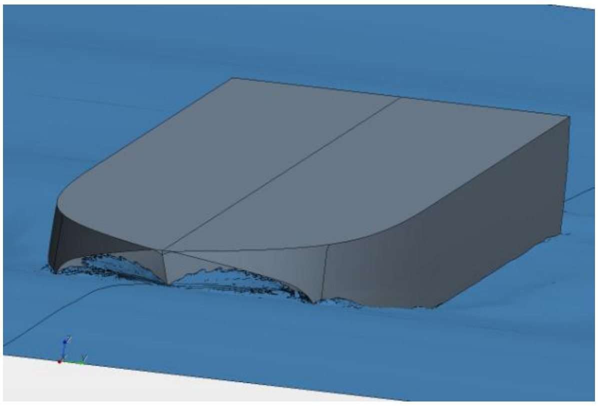


Figure 8.CFD solution during wave slamming on 2.5m model. Regular wave height= 90mm (4m at full scale), speed =1.53m/s (20 knots at full scale), wave encounter frequency=1.1Hz, ω_e*=3.5.

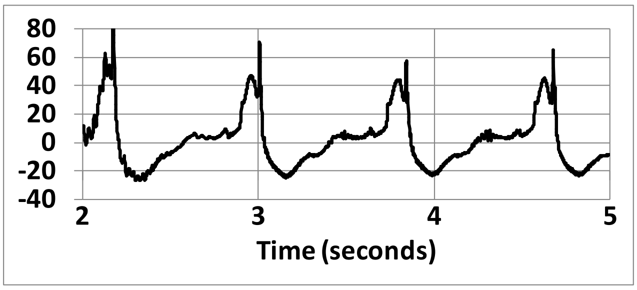


Figure 9.CFD solution for upward bow force (N) during wave slamming on 2.5m model. Regular wave height= 90mm (4m full scale), speed =1.53m/s (20 knots full scale), encounter frequency=1.1Hz, ω_e*=3.5.

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