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RESPONSE ANALYSIS OF SHIP STRUCTURES SUBJECTED TO A CLUSTER OF IMPULSIVE EXCITATIONS

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Abstract

In rough seas, the violent impact between the water and the hull, *i.e.*, slamming, may determine relevant local and global effects on the ship's structure, emphasized by the likely onset of hull vibrations, especially in innovative ship configurations characterized by lighter and flexible structures. Several models have been developed in last years both to analyze the single slamming event and its effect on the hull portion directly impacting the water surface (local effect) and on the whole ship (global effect), as well as probabilistic theories to study the occurrence of such slamming events. Both of these information, *i.e.*, amplitude and frequency of the load cycles, are needed to evaluate the fatigue life of the structure. Though the random wave loads in operative conditions follow a Gaussian distribution, the slamming impacts show much more complex statistical features such as, for instance, the presence of clusters. On the basis of seakeeping experimental results performed on a segmented elastically scaled model of a fast monohull as well as on theoretical modeling of the slamming load, the global and the local ship response to these repeated impacts is investigated.

1. INTRODUCTION

In recent years, an important trend in shipbuilding industry has been the increase in the length and speed of high-speed crafts, thus demanding for lighter structures. The resulting fast and flexible vehicles are more likely to be excited by impulsive loads, like slamming, due to the violent impact between the water and the hull when a ship sails in rough sea. Among global effects, the impulsive characteristic of the slamming load induces ship's vibrations at hull girder natural frequencies, commonly denoted as whipping. Regarding local effects, it is clear that slamming loads can produce high stresses in the hull plates directly impacting the water surface causing permanent deformations and possible failures. Besides the large amount of works devoted to the theoretical [1,2] or numerical [3] modelling of the single impact in terms of force or pressure distribution, a very important issue is the statistical characterization of the occurrence of the slamming events. The most popular theories [4], based on the hypothesis that each slamming event is uncorrelated from the others, assume that slamming impacts follow a Poisson process. However, it was demonstrated theoretically [5] and experimentally [6] that the narrow-band character of the relative motion process tends to concentrate the slamming impacts in distinct clusters; this fact violates the hypothesis of mutual independence

of the individual slamming events. It is thus interesting to analyse both the local and the global structural response to these repeated impulsive loads. The characterisation of the single impact can be done if at least the relative vertical displacement (and its time derivatives) between the wave and the hull are known. In the present work these quantities are derived from tests performed in irregular waves on a prototype high-speed vessel scaled elastically using the segmented model concept with supporting backbone. The model consists of six segments connected to an elastic beam whose sections are shaped in order to give the proper stiffness in bending. In this way, it was possible to scale the lowest hull bending natural modes in addition to the rigid-body features and to measure the elastic response of the whole ship in terms of vertical bending moment (VBM). The local response is provided solving the plate equation of motion forced by a pressure distribution obtained by an analytical expression derived by a two-dimensional linearised theory.
The performed analysis shows that cluster phenomena can be important when evaluating the global ship response. In fact, the time scale of the ship vibrations may be of the same order than the time interval between consecutive impacts causing an amplification of the oscillation amplitude due to a superposition of the effects induced by the slamming events. On the contrary, due to the completely different time scales between the occurrence of slamming impacts and the typical period of oscillation of hull panels, the local slamming effects can be studied as the response to single and uncorrelated impact events.

2. HYDROELASTIC SCALING AND TESTING

2.1 Reduction to the equivalent ship-beam

It is well established among naval architects that the ship bending behaviour can be carefully described through a beam model if the analysis does not concern the description of the transverse section deformation under loading. Such level of approximation is satisfactory for the determination of the distribution of the shear force and bending moments due to the wave and inertial loading (*i.e.*, longitudinal bending vibrations), excited by the presence of water impacts in rough sea. The 1D characteristics of such descriptive model, considering the natural 3D extension of the ship structure, have to be determined using as input data the finite element (FE) model of the structure. The reduction methods generally consist of a re-definition of a discretized dynamical system using only a sub-set of the state-space variables (master degrees of freedom, dofs) but keeping an acceptable level of accuracy. This approach is of relevant interest in the present case because the dynamic analysis involves a finite frequency band and, moreover, it is legitimate to consider that ship structural elements play different roles in responding to the applied loads. A first “natural” change of co-ordinates is given by the structural real eigenvectors: in this case, the mass and stiffness matrices of the reduced problem, accounting for the low-frequency mode of the structure (*e.g.,* heave, pitch and 2-node bending mode) can be ideally obtained from the generalized displacements of the FE eigenmodes evaluated at the elastic axis of the 3D structure. Nevertheless, this approach presents some practical difficulties. A similar but more efficient approach is based on the determination of the collapsed and equivalent distribution of mass, bending and shear stiffness affecting ship bending vibration in the vertical plane, leading to transform the 3D FE model into a Timoshenko 1D-beam model. The structure is discretized longitudinally defining new nodes along the elastic axis (20 nodes in the present case), at which the mass per unit length and the shear area are evaluated. The equivalent sectional moment of inertia is obtained instead as that one inducing the same static response of the 3D FE model.

2.2 Froude similarity and experimental constraints

The mass, bending and shear stiffness distributions, obtained as described above, were then used as reference data for the scaled model (λ = 1:30). The scaling is based on the Froude similarity that implies the Froude number to be preserved through scale reduction, defined as $Fr = V/(g\,L_{pp})^{1/2}$, where V is the forward speed, g is the acceleration of gravity and L_{pp} is the length between perpendiculars. It is interesting to note that, since Froude similarity implies directly $V^{(ship)}/V^{(model)} = \sqrt{\lambda}$, the same relationship holds between the ship and the model time scales, whereas the inverse of this, $f^{(ship)}/f^{(model)} = \sqrt{\lambda}$, is valid between the ship and model frequencies. Of course, these reference distributions can not be reproduced exactly with the scaled model design because the experimental set-up (for instance, the presence of the towing device) introduces specific mechanical elements that do not pertain to the original ship construction. Though the use of a fully elastic, self-propelled model should be the most accurate choice for representing the full-scale ship, it is not a cost-effective solution and, as long as the global response is mainly described in terms of the first (2-node) bending mode, the backbone-modelling technique, utilized for the construction of the segmented hull, is capable of satisfactory level of representation.

Thus, the vertical bending behaviour of the ship was reproduced by shaping properly the elastic beam that formed the backbone of the segmented model. The elastic beam was made of an aluminium alloy with 20 elements of constant length and variable transverse sections. The fiber-glass hull (see Figure 1) was divided into six segments, each one connected to the elastic beam with a vertical steel leg, whereas the gaps between adjacent segments are made water-tight by using rubber straps. The reference value of 2-node bending mode (11.3 *Hz*) was closely approximated in dry-vibration tests of the segmented model (10.8 *Hz*). This satisfactory result was confirmed also for the corresponding wet mode.

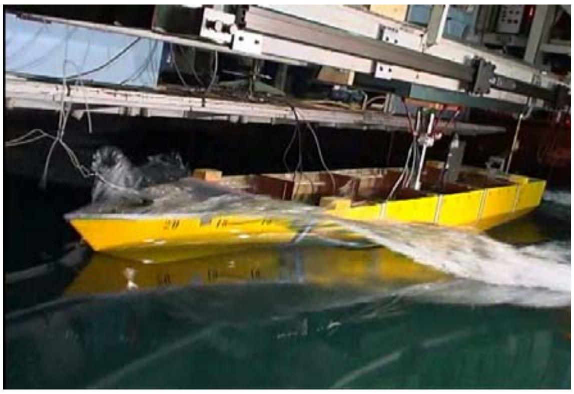

Figure 1. Segmented model.

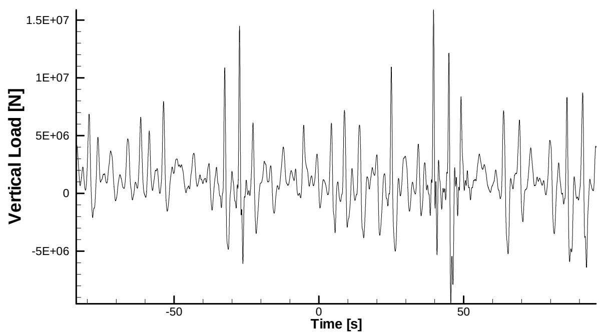

Figure 2. Residual force acting on segment 2.

2.3 Experimental set-up

The model experiments were carried out at the INSEAN towing tank basin. The basin (220 *m* long) is equipped with a single-flap wave-maker capable to generate regular and irregular wave patterns. As a rigid-body, the physical model was free to heave, to pitch and, partially, to surge. The measured physical quantities considered in the present investigation are the absolute wave height, the rigid-body dofs, the vertical force on each segment and the vertical bending moment on several beam sections. The incoming, absolute wave height was measured with a finger probes placed on the model left side corresponding to the mid-point position of the second segment. The heave, pitch and surge dofs were measured with the Krypton Rodymm DMM system. This is an optical system based on cameras placed on the carriage and on four LEDs glued on a plate carried onboard the model. The legs connecting the segments to the elastic beam embedded the cell loads to measure the vertical force. The bending moment acting upon the beam is measured in 12 points by using strain gauges glued on the top face of the beam. The calibration of the strain gauges was performed loading statically the beam and comparing the theoretical bending moments with the voltage values. The acquisition system based on a National Instruments SCXI module recorded globally 28 signals at a 500 *Hz* sampling rate. Experimental data were analised in particular for the second segment that is recognised to be the most significant for the slamming events.

3. GLOBAL RESPONSE

The experimental tests were performed in irregular (stochastic) head waves at a forward speed of 2.82 *m/s* corresponding to 30 *kn* at full scale. The irregular sea is represented as a stochastic (Gaussian) process expressed as the sum of harmonic components, with uncorrelated phases and amplitude and frequency of each obtained by the discretization of typical marine spectrum (JONSWAP spectrum in the present case).

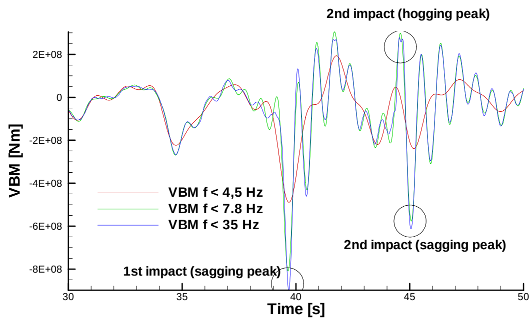

Figure 3. Different filtering of the recorded amidship VBM.

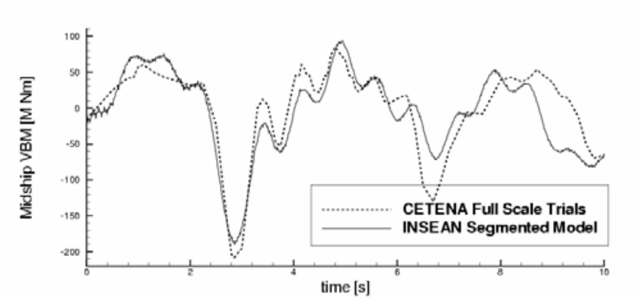

Figure 4. Comparison between full-scale and model-scale VBM measurements at midship.

The onset of a relevant whipping contribution to the VBM is highlighted considering a severe slamming test, with a significant wave height at full scale $H_{1/3} = 5$ *m*, a significant period $T_{1/3}$ =6 *s* and a full-scale forward speed of 30 *kn* (experimental results are transformed to full-scale). In these test conditions several impacts occur, often grouped in clusters lasting short time periods, as put in evidence in Figure 2 where several peaks in the time history of the hydrodynamic load appear, corresponding to slamming events. The VBM signal close to midship is plotted in Figure 3 considering three different low-pass filtering at 4.50 *Hz* (response to continuous wave loading), 7.8 *Hz* (inclusive of whipping VBM) and 35 *Hz* (thus including also the 3-node mode of the segmented hull). The response appears to be dominated by the whipping VBM when a slamming event occurs and it is only slightly affected by the higher order modes. In particular, sagging moment is relevantly affected by whipping oscillations. The presence of consecutive impacts allows to put in evidence that, if the second one happens when previous oscillations are not yet damped out, also the hogging moment preceding the impact may be emphasized. The closeness of the responses between the experimental tests and full-scale trials is shown in Figure 4 for a different condition, approximately correspondent at 30 *kn* and $H_{1/3}$= 3 *m* in full-scale. The test trials were performed by CETENA [7] during routine cruises of the ship monitoring several quantities, like motions, relative wave at bow and deformations of a bottom panel. Three long-base strain gauges (LBSG) were placed amidship, one on the central keel and two under the main deck. Each LBSG was constituted by a LVDT sensor linked to a 2.43 *m* truss. Though a perfect similarity in the input sea condition can not be stated (the correspondence of the seas is assessed only from a statistical point of view), the period and damping of the whipping VBM oscillations appear in good agreement to each other. Therefore, for this kind of ships, it is apparent that wave-induced stresses arise as a consequence of both external fluid loading - continuous waves and random slamming loads, - and induced, transient loads, - generated by excited hull vibrations.

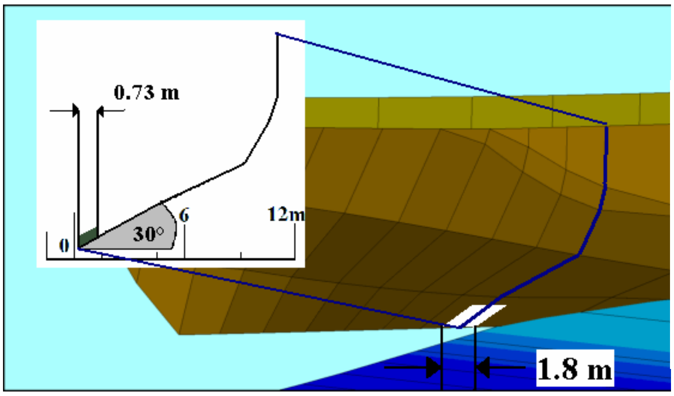

Figure 5. Bottom panel position and section characteristics.

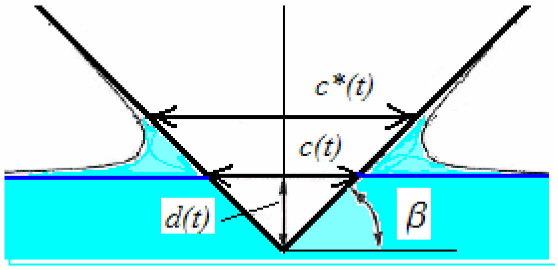

Figure 6. Wedge parameters.

4. LOCAL RESPONSE

4.1 Slamming load modelling

In this section, the description of the pressure field loading the bottom panel is carried out, with particular attention to slamming that is the most severe phenomenon for the panel response. In Figure 5, the position of the panel is shown (filled in white) with respect to the water surface before a slamming event (the picture refers to a virtual reconstruction of a wave test on the basis of measured experimental data). Since the shape of the ship section close to the bottom is well fitted by a wedge (Figure 6), the pressure field $p(y,t)$, within a 2D approach [1], can be calculated using the following expression:

$$p(y;t) \ = \ \pi\,\rho \left[\frac{c_*(t)\dot{c}_*(t)}{\sqrt{c_*^2(t)-y^2}}\dot{\zeta}_r(t) + \ddot{\zeta}_r(t)\sqrt{c_*^2(t)-y^2} \right] \tag{1}$$

where c_* is the wetted length, $\dot{\zeta}_r, \ddot{\zeta}_r$ are the entry velocity and acceleration, ρ_f is the water density. The wetted length is obtained basically from the draught $d(t)$,

$$c_*(t) = \gamma_*\, d(t)/\tan\beta$$

where β is the deadrise angle, times a coefficient γ_* that accounts for the increase in the wetted length due to the water up-rise. The above expression is herein discussed for the case of water entry for sake of conciseness, though it remains valid also during the exit phase if the first term is dropped and other conditions are applied to the value of the coefficients [8]. The splash-up coefficient γ_* can be considered constant during the entry phase if $\dot{\zeta}_r = \text{const}$; hence, since the worst loading of the panel occurs when the entry velocity of the section does not vary, this approximation is surely satisfactory. The entry velocity and acceleration are obtained by taking numerically the substantial derivatives of the relative wave elevation (the hull speed V is involved), defined as:

$$\zeta_r(t) = h(t) - w_G(t) - x\cdot\vartheta(t)$$

where $h(t)$ is the undisturbed wave elevation at the coordinate x, $w_G(t)$ and $\vartheta(t)$ are the displacement of the centre of gravity (heave) and the rotation around it (pitch), respectively. A delicate aspect concerns the determination of the wave elevation at x, because it is measured in a different but generally close point using the finger probe. In the case of regular (sinusoidal) waves, since the wave is ideally monochromatic, the wave elevation at a considered section is simply equal to the wave measurement at the time $t\text{-}\Delta t$, where the time interval Δt depends on the wave celerity and the covered distance from the probe to the section. In the case of irregular waves, made of multiple wave harmonics with amplitudes accordingly a typical sea spectrum and uncorrelated phases, the wave pattern is assumed to be frozen and to travel with a constant wave celerity corresponding to the largest amplitude harmonic.

4.2 Plate response

The hull bottom panel considered for the local structural analysis is located, in longitudinal direction, at 88.32 *m* from the forward perpendicular between two main transversal frames and, in transversal direction, between the keel line and the first secondary reinforce. The plate material is aluminium (Young modulus E=69 *MPa*, density ρ=2700 kg/m^3, Poisson modulus ν=0.334) and its dimensions are: 1.8x0.84x0.006 *m*. The plate displacement w can be obtained solving the following equation of motion:

$$\nabla^4 w + \overline{m}\ddot{w} = p(x, y, t) \tag{2}$$

where $\overline{m}$ denotes the mass per unit surface given in general by the sum of the structural mass and of the hydrodynamic added mass.

According to the two-dimensional model adopted for the description of the hydrodynamic load it is assumed that pressure does not vary along the x axis. However, from the observation of expression (1), it can be noted that the two-dimensional linearised theory produce a singularity for the pressure distribution in correspondence of the water front. To avoid this problem, starting from the discretization of the plate width in elements of length dl, at each time step, the pressure distribution was integrated along the wetted part of each element and the results were then divided by this wetted length (less or equal than the panel length dl). Moreover, though the added water mass should be included in the evaluation of the structural response, the analysis of experimental data [9] demonstrated that the structure oscillates accordingly to its first natural dry period. This fact was explained by Faltinsen et al. [9] with cavitation and ventilation phenomena occouring because of the large initial structural deflection. Hence, as a first approximation, added water mass was not considered in the present analysis and the comparison performed with full scale experimental data seem to confirm the validity of this assumption. On the other side, damping was taken into account considering, as usual, a complex natural frequency $\tilde{\omega}_{mn} = \omega_{mn}(1 + i\eta)$, where η is the loss factor sum of the structural and of the added water damping. Structural damping, usually very low, plays an important role only in the first instants of the impact while the water added damping dominates the response when the panel is wetted. From the analysis of the full scale experimental data a constant value equal to 0.05 was assumed for the damping coefficient. Equation (2) was solved by using a modal expansion and a Runge-Kutta time integration method. Clamped edge conditions are considered and plate eigenfrequencies and eigenvectors were obtained by a FEM analysis. The numerical procedure was validated by comparison with full scale trials. In particular in Figure 7 a pressure slamming event, measured on the panel at y=0.168 *m* from the keel line, is shown together with the induced panel deformation in x direction, measured by the strain gauges positioned at the centre of the plate. Although the exact sea state conditions during the full scale trials were not replicated in the model scale experiments, it is reasonable to consider for comparison a slamming event that, at the same location, has a similar shape and a close maximum value. In Figure 8 the slamming pressure and the plate deformation obtained numerically are displayed showing a good agreement with the experimental data. In particular, the amplitude (peak to peak) of the first structural oscillation period is of the same order of magnitude than that obtained experimentally and the plate oscillations are damped in the same time interval.

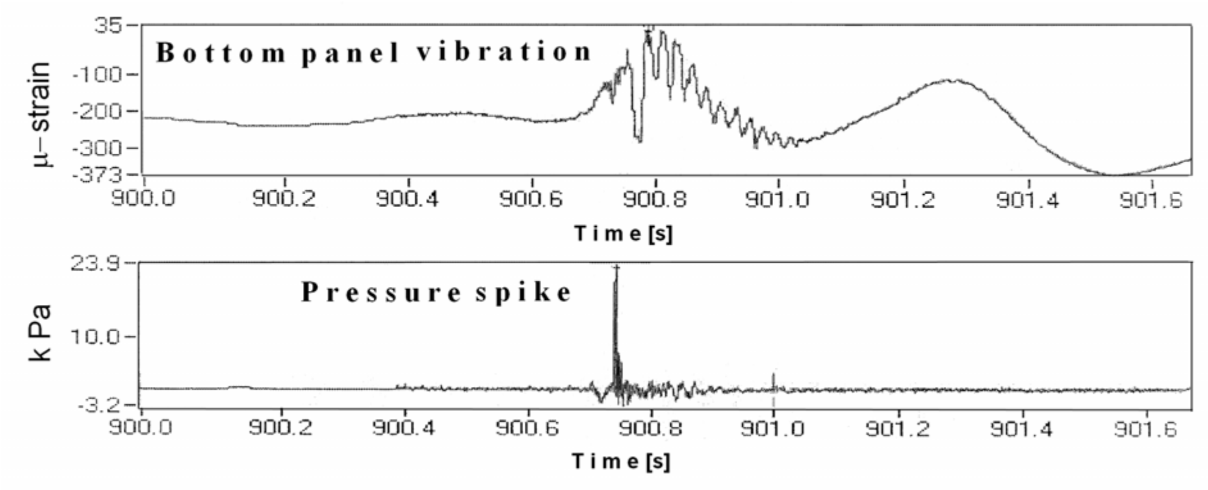

Figure 7. Full scale experimental pressure spike and induced panel deformation

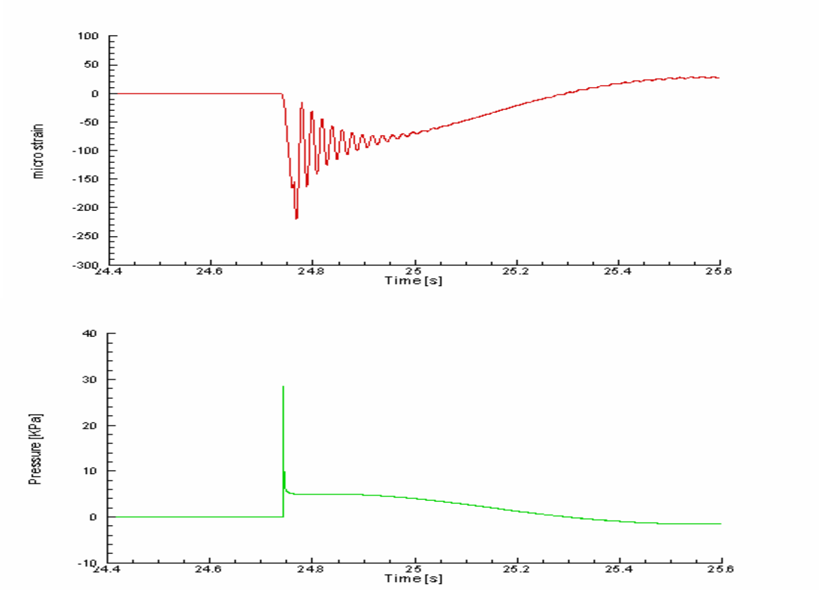

Figure 8. Full scale numerical pressure spike and induced panel deformation.

Let us now analysed the panel response obtained for the test already considered in section 3 for the global response. In Figure 9 a time history of the pressure for a point located close to the centre of the plate is shown. It can be noted that slamming peaks appear in a form of cluster confirming that they can not be considered as uncorrelated events. On the contrary structural response plotted in Figure 10 in terms of Von Mises stresses does not show any correlation. In fact the time scale of the structural vibrations is sensibly lower than the time interval between two slamming events; hence, it can be concluded that, from a local structural point of view, each impact can be treated as a single event.

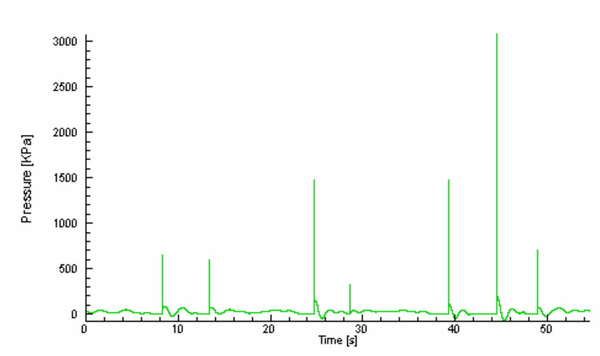

Figure 9. Cluster of slamming impacts.

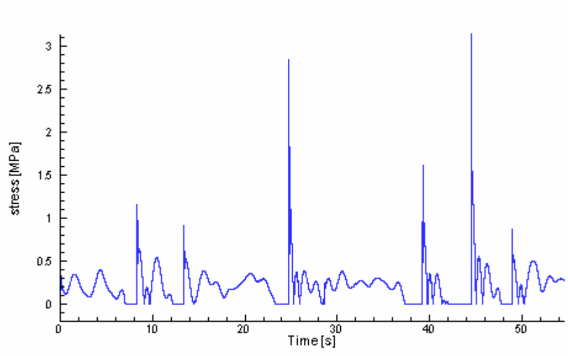

Figure 10. Von Mises stress induced by the cluster of impacts.

5. CONCLUSIONS

In this paper the local and global full scale respose of ship structures subjected to a cluster of slamming impacts is analysed. It is pointed out that a statistical representation of the slamming events is fundamental when dealing with the elastic response of the whole ship. In the case of local analysis, the hull bottom panel response can be considered affected only by separate and uncorrelated impacts.

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