PROGRESSIVE DAMAGE AND FAILURE OF CURVED SANDWICH STRUCTURES DUE TO WATER SLAMMING

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1 Introduction

The local water slamming refers to the impact of a part of a ship hull on water for a brief duration during which high peak pressure acting on the hull can cause significant local structural damage [1]. von Kármán's [2] work on water entry of a rigid v-shaped wedge with small deadrise angle β was generalized by Wagner [3] to include effects of water splash-up on the body. Zhao et al. [4] extended Wagner's solution to wedges of arbitrary deadrise angles, numerically solved problems by using a boundary integral equation method, ignored effects of the jet flow, and found that the variation of the hydrodynamic pressure on the rigid hull agreed well with that found experimentally implying that the jet flow does not significantly affect the pressure variation on a rigid wedge. Mei et al. [5] analytically (numerically) solved the general impact problems of cylinders and wedges of arbitrary deadrise angles by neglecting (considering) effects of the jet flow.

In practical slamming impact problems, the hull is curved as well as deformable, and its deformations affect the motion of the fluid and the hydroelastic pressure on the solid-fluid interface.

Sun and Faltinsen [6] have considered hydroelastic effects in analyzing deformations of circular steel and aluminum shells by studying deformations of the fluid by the boundary element method (BEM) and those of shells by the modal analysis. Qin and Batra [7] have analyzed the slamming problem by using the {3, 2}-order plate theory for a sandwich wedge and Wagner's theory modified to account for wedge's infinitesimal elastic deformations. The plate theory incorporates the transverse shear and the transverse normal deformations of the core, but not of the face sheets which were modeled as Kirchhoff plates. Here we analyze the local water slamming problem for a curved deformable sandwich hull by using the {3, 3} theory for the face sheets and the core. Structural deformations are analyzed by the finite element method (FEM) and those of the fluid by the BEM. The two are coupled by requiring the continuity of the pressure and the normal component of velocity at the water/hull interface.

2 Problem formulation and solution method

2.1 Problem formulation

The material of the face sheets is assumed to be linear elastic and transversely isotropic with the fiber axis as the axis of transverse isotropy, and the material of the core taken to be linear elastic and isotropic. Infinitesimal deformations of each face sheet and the core are studied by using the 3rd order shear and normal deformable plate/shell theory (HSNDT) of Batra and Vidoli [8]. In governing equations derived by Hamilton's principle, all inertia effects are considered. However, we study only delamination between the face sheets and the core with a criterion quadratic in transverse shear and transverse normal stresses at the interface. That is, the delamination occurs when

$$f_s(\tau_i) - 1 = 0$$
$$f_s(\tau_i) = \left(\frac{\sigma_z}{[\sigma_z]}\right)^2 + \left(\frac{\sigma_{xz}}{[\sigma_{xz}]}\right)^2, \quad \sigma_z \geq 0$$
$$f_s(\tau_i) = \left(\frac{\sigma_{xz}}{[\sigma_{xz}]}\right)^2, \quad \sigma_z < 0$$

Here σ_z and σ_{xz} are the normal and the tangential tractions at a point on the interface between the core and the face sheets, and $[\sigma_z]$ and $[\sigma_{xz}]$ are the corresponding strengths of the interface. It is simulated by including $w_{t0}(x,t)$ etc. in the expression for the deflection, i.e.,

$$w_t(x,z,t) = w_{c0}(x,t) + h_c l_z(x,t) + h_c^2 m_z(x,t) + {h_c}^3 n_z(x,t) + w_{t0}(x,t) + (z - h_c) l_{tz}(x,t) + (z^2 - h_c^2) m_{tz}(x,t) + (z^3 - h_c^3) n_{tz}(x,t),$$

$$w_c(x,z,t) = w_{c0}(x,t) + z l_z(x,t) + z^2 m_z(x,t) + z^3 n_z(x,t),$$

$$w_b(x,z,t) = w_{c0}(x,t) - h_c l_{cz}(x,t) + h_c^2 m_{cz}(x,t) - h_c^3 n_{cz}(x,t) + w_{b0}(x,t) + (z + h_c) l_{bz}(x,t) + (z^2 - h_c^2) m_{bz}(x,t) + (z^3 + h_c^3) n_{bz}(x,t)$$

Here w_t, w_b and w_c are displacements along the z-direction of top and the bottom face sheets and the core, respectively. Variables l_{cz}, m_{cz}, n_{cz}, l_{bz}, m_{bz}, n_{bz}, l_{tz}, m_{tz}, and n_{tz} are higher order generalized displacements along the z-direction. Similar expressions are assumed for the axial displacement, u. Quantities u_{b0} and w_{b0} represent the jump in displacements when delamination occurs at the interface between the core and the bottom face sheet.

The fluid is assumed to be incompressible and inviscid, and its deformations to be irrotational. Effects of gravity forces are neglected and a plane strain problem is studied.

2.2 Solution method

The Laplace equation for the velocity potential of the fluid is solved by using the BEM. Deformations of the sandwich structure are analyzed by the finite element method (FEM), and are coupled to those of the fluid by enforcing the continuity of the hydrodynamic pressure and the normal velocity of the contacting fluid and solid particles at the fluid/solid interface.

3. Results and discussion

In order to verify the FE code, we have compared in Table 1 computed results for three values of S (= mean radius, R/thickness) with the exact solution of the shell with an outward pointing uniformly distributed pressure applied to it. Results of the {3, 3} shell theory are very close to those obtained from the elasticity solution thereby verifying the FE code. In Fig. 2 we have plotted time histories of the deflection of the centroid of the hull for different values of R. It is clear that with an increase in the value of R, the centroidal deflection approaches that of a flat hull. Also, at a fixed time, the deflection increases with an increase in R which could be due to the dependence of the wetted length and the pressure distribution upon R.

We have exhibited in Fig. 3 the variation of the hydroelastic pressure on a 1-m long circular ship hull of initial deadrise angle 5° impacting water at 10 m/s and using material properties of the core and the face sheets listed in [7]. It is clear that the curvature of the hull noticeably affects the magnitude of the peak pressure and the pressure distribution on the hull. At t = 6.02 ms and four values of R, the variation of the strain energy density in the core and the face sheets along the hull span is plotted in Fig. 4. It is clear that at a point on the hull the strain energy density in the core and the face-sheet decreases with a decrease in the value of R.

The delamination has been studied for a flat hull with values of material properties, the deadrise angle and of the downward velocity the same as those assigned in [9]. From results exhibited in Figs. 5 and 6, we conclude that the centroidal deflection increases and the energy absorbed in the core decrease dramatically when the delamination is considered.

Table 1: Comparison of computed and analytical values of the non-dimensional deflection $\underline{w}$, the axial stress $\underline{\sigma_x}$, and the transverse shear stress $\underline{\sigma_{xz}}$.

S	Exact			{3,3} Shell theory		
	$\underline{w}$ $\left(0,\frac{\emptyset}{2}\right)$	$\underline{\sigma_x}$ $\left(-\frac{h}{2},\frac{\emptyset}{2}\right)$	$\underline{\sigma_{xz}}$ (0,0)	$\underline{w}$ $\left(0,\frac{\emptyset}{2}\right)$	$\underline{\sigma_x}$ $\left(-\frac{h}{2},\frac{\emptyset}{2}\right)$	$\underline{\sigma_{xz}}$ (0,0)
10	0.144	-0.995	0.525	0.144	-0.992	0.525
50	0.0808	-0.798	0.526	0.0801	-0.791	0.521
100	0.0787	-0.786	0.523	0.0774	-0.775	0.516

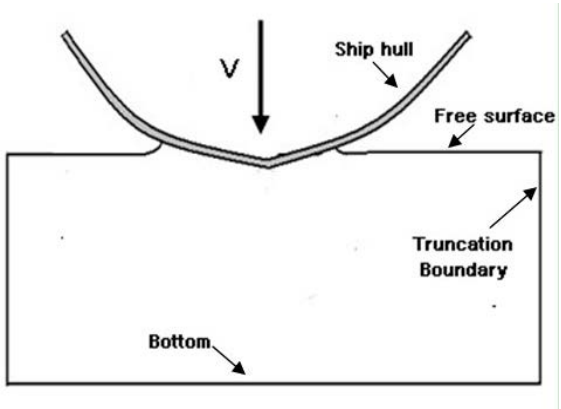

Fig. 1: Schematic sketch of the problem for curved shaped hull.

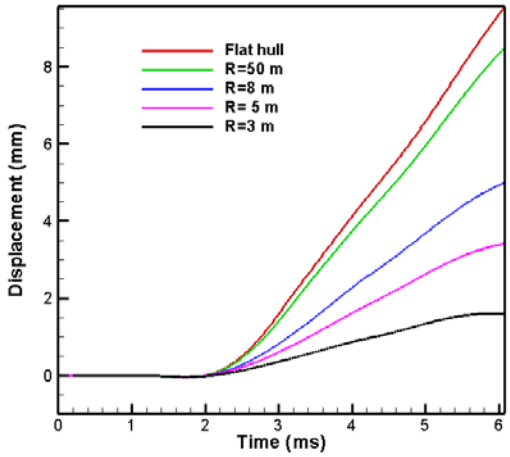

Fig. 2: Time histories of the deflection of the hull centroid for different values of the mean radius of the hull.

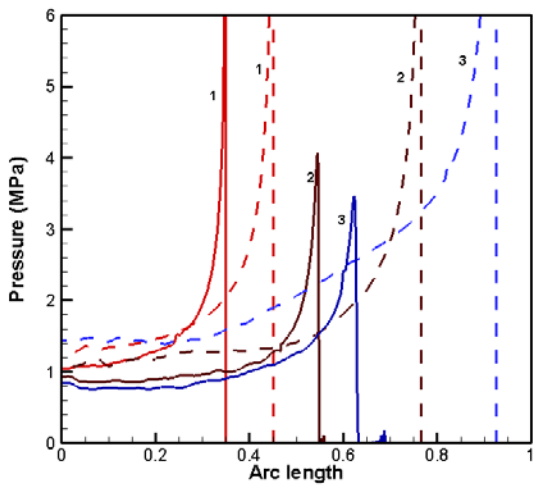

Fig. 3: At t = 2.72, 4.79 and 5.75 ms, respectively, curves 1, 2 and 3 represent distribution of the hydroelastic pressure on the hull/water interface; R = 5 m, solid lines for the curved hull, dotted lines for a straight v-shaped hull.

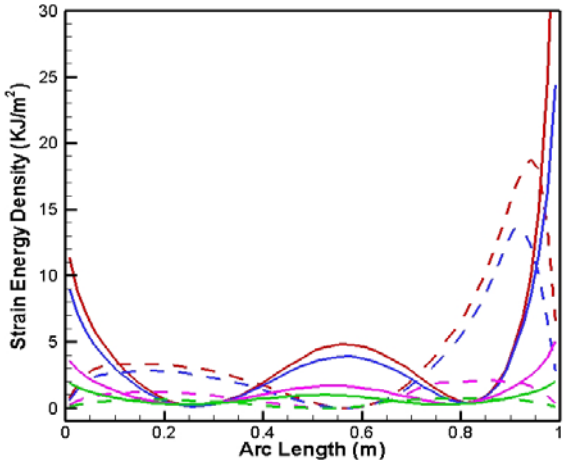

Fig. 4: At t = 6.02 ms variation of the strain energy density along the hull span; solid and dotted curves represent, respectively, energy density in the core and the face sheets. The red, blue, pink, and green curves are for R = ∞, 50, 8, 5 m, respectively.

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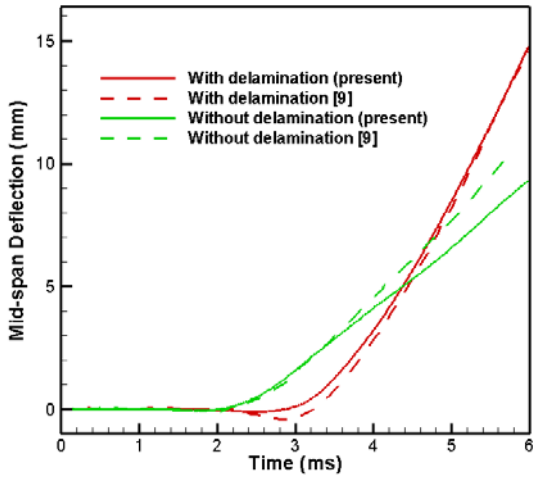


Fig. 5: Time history of the deflection of the flat hull centroid with and without the consideration of delamination.

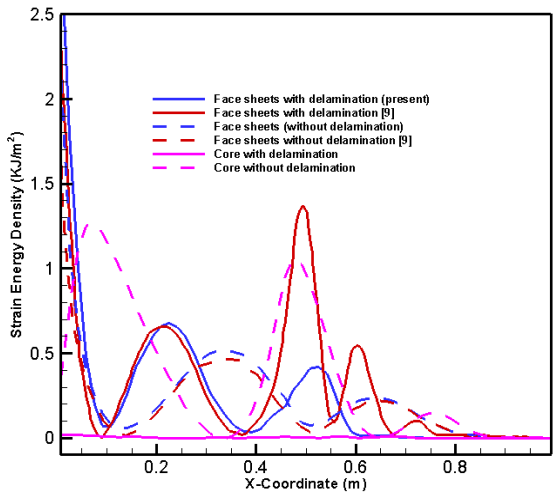
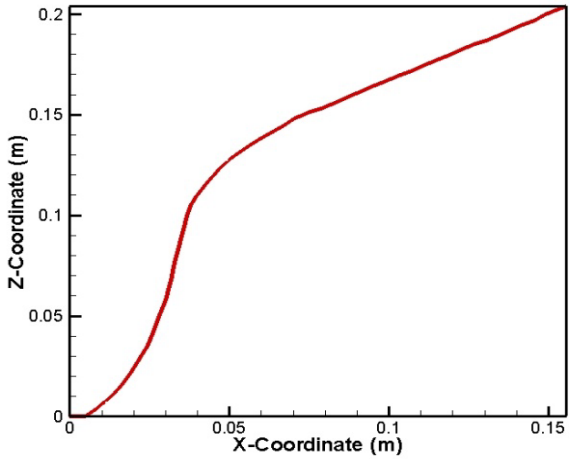
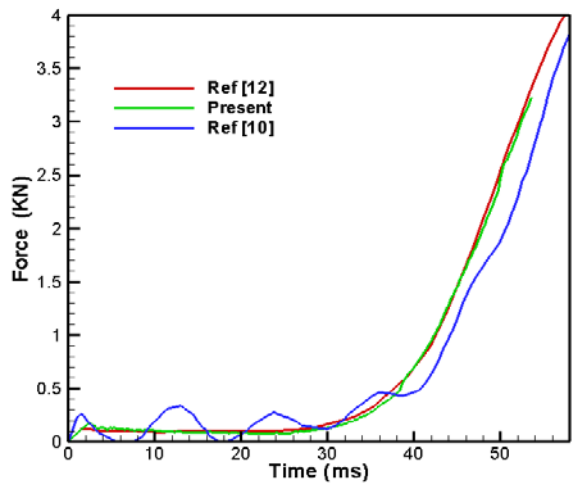


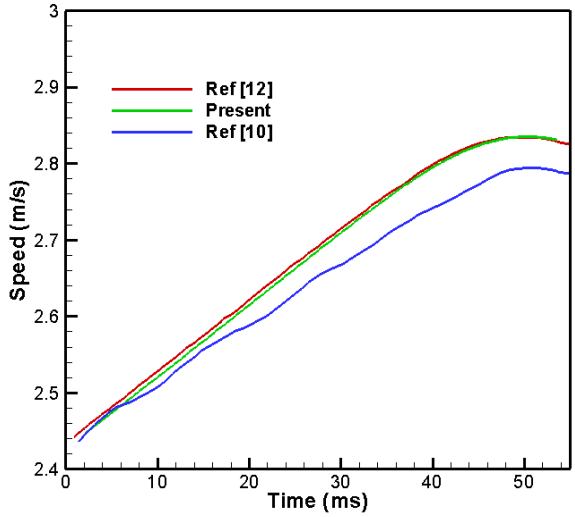
Fig. 6: At t = 2.735 ms, variation of the strain energy density in the core and face sheets along the hull span.



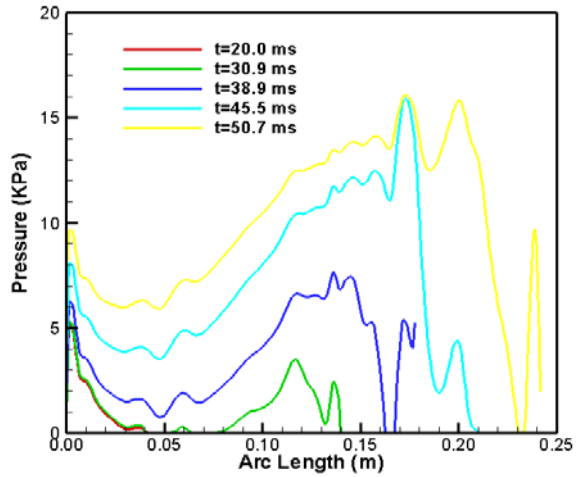
(a)



(b)



(c)



(d)

Fig. 7: (a) Cross-section of the ship bow section; time history of the (b) upward axial force acting on the ship bow section and (c) the axial velocity; and (d) variation of the pressure along the ship bow section at t = 20.0, 30.9, 38.9, 45.5 and 50.7 ms.

We have simulated the drop test of a ship bow section studied experimentally in [10]. The bow section of length 1 m shown in Fig. 7a and having a total weight of 241 kg is dropped freely into calm water with an initial vertical velocity of 2.43 m/s . The bow section profile is approximated by cubic splines. Computed time histories of the resultant axial force and the axial velocity are compared with

their corresponding experimental values in Figs. 6b and 6c; the two sets of results are found to be close to each other. Small differences between the computed and the test results could be due to the neglect of the fluid viscosity and the 3-dimensional effects. As depicted in Fig. 7d, the spatial variation of the pressure on the ship bow section is different from that on a straight wedge section. On a bow section the high pressure acts on a larger length than that on a straight wedge; this has also been reported in [11].

4. Conclusions

We have studied the hydroelastic interaction between a curved deformable sandwich hull and initially calm water. It is found that the curvature of the shell has a noticeable effect on the pressure distribution, and hence on deformations of the hull. For a flat hull, the strain energy absorbed by face sheets and the core decreases with an increase in the delamination length.

5. Acknowledgements

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