DOUBLY CURVED COMPOSITE SANDWICH PANELS FOR HYBRID COMPOSITE/METAL SHIP STRUCTURES

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Abstract

Doubly curved composite sandwich panels loaded by evenly distributed pressure were designed, analyzed, manufactured and tested. Quick and cost effective methods for making molds for vacuum infused doubly curved composites were studied and implemented. Several different manufacturing techniques for making doubly curved panels and doubly curved foam cores were investigated. Tests were performed using a hydrostatic water tank.

Keywords: doubly curved, glass fiber, foam core, composite sandwich panel, vacuum infusion, hydrostatic testing, joints

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1. Introduction

Since the beginning of time there has been a trend towards making stronger, more lightweight and more cost effective structures. These structures can be found almost everywhere, including in aerospace, naval, automotive and public transportation vehicles, bridges and other civil infrastructure, sporting equipment, etc. Composite sandwich structures are particularly well suited for marine vehicles because of high strength and stiffness to weight ratios, high corrosion and fatigue resistances, and the ability to be manufactured into complex shapes.

Many advanced structures have complex curved geometries that complicate accurate design and analysis. There is plenty of literature on doubly curved shells, investigating buckling, vibration, etc., but considerably less on doubly curved panels subjected to hydrostatic pressure. Librescu and Hause [1] did a survey of the developments in the modeling and behavior of advanced sandwich constructions, focusing in particular on post-buckling. Hohe and Librescu [2] investigated a nonlinear theory for doubly curved sandwich shells. Drake et. al. [3] did analytical approximations for a square panel with flat top skin and curved bottom skin loaded with a uniform pressure. Burton and Noor [4] compared nine different 2D modeling approaches for curved shells under thermal loadings. Skvortsoc et al. [5] assessed different models for simply supported beams with single curvature loaded under uniform pressure. O'Sullivan and Slocum [6] studied alternatives to honeycomb and corrugated core sandwich designs with double curvature. Russo and Zuccarello [7] investigated the failure modes of glass fiber reinforced plastic (GFRP) sandwich panels both experimentally and numerically with non linear Finite

Element (FE) models. MacDonald and Chen [8] analytically studied simply supported rectangular sandwich panels with small initial curvature under general loading. Thompson et al. [9] studied flat composite sandwich panels loaded with hydrostatic pressure both experimentally and numerically. Cunningham et. al. [10] studied the effect of curvature, material, fiber orientation, and boundary conditions using a FE model and did experimental free vibration tests on carbon fiber skin / honeycomb core sandwich panels.

Hydrostatic pressure is of particular importance for ship hull panels. Hydrostatically loaded doubly curved panels is the focus of the present paper. Panels were numerically analyzed, manufactured, and then tested under hydrostatic loading in a specially designed test tank. Results of the testing are compared to FE models.

There have been few papers that study inexpensive mold manufacturing methods for complex shaped vacuum infused parts. Kuppusamy [11] studied rapid tool manufacture for several different cases and listed different tool materials for a Resin Infusion between Double Flexible Tooling (RIDFT) process which is similar to Vacuum Assisted Resin Transfer Molding (VARTM). McCaffery et. al. [12] studied low cost mold development for Resin Transfer Molding (RTM) and concluded that the optimal mold should be plastic with wood stiffeners.

2. Geometry of Experimental Test Panels

The width of sandwich panels on high speed light craft ranging from smaller motorboats to ships well over 50 m in length is typically on the order of 0.5 or 1 m. For the present study, a panel size of 0.6m x 0.6m was chosen for the experimental testing. Apart from being of a relevant size, it also had the benefit of fitting in existing CNC machines including a 5-axis router and an abrasive waterjet cutter. The panels were doubly curved, made in a doubly curved mold, and consisted of an outer skin, a lightweight PVC foam core which was smaller than the panel, an inner skin, plus some additional layers to be explained shortly. A schematic of a panel is shown in Fig. 1. The core was smaller than the panel and tapered off such that the inner and outer skins joined to form a single skin flange at the edge of the panel. This single skin edge would be very thin and weak unless reinforced. Thus, some "thickening layers" of coarse fiber architecture were added between the skins at the edge. Further, the area around the tapered part of the foam core was covered with additional "reinforcing layers" to prevent failure; see Fig. 1. The single skin edge is of particular interest for steel / composite ship hulls, where the hull consists of a stainless steel frame to which lightweight sandwich panels are bonded; see for example Cao et al. [13,14], Maroun et al. [15], and Grenestedt [16].

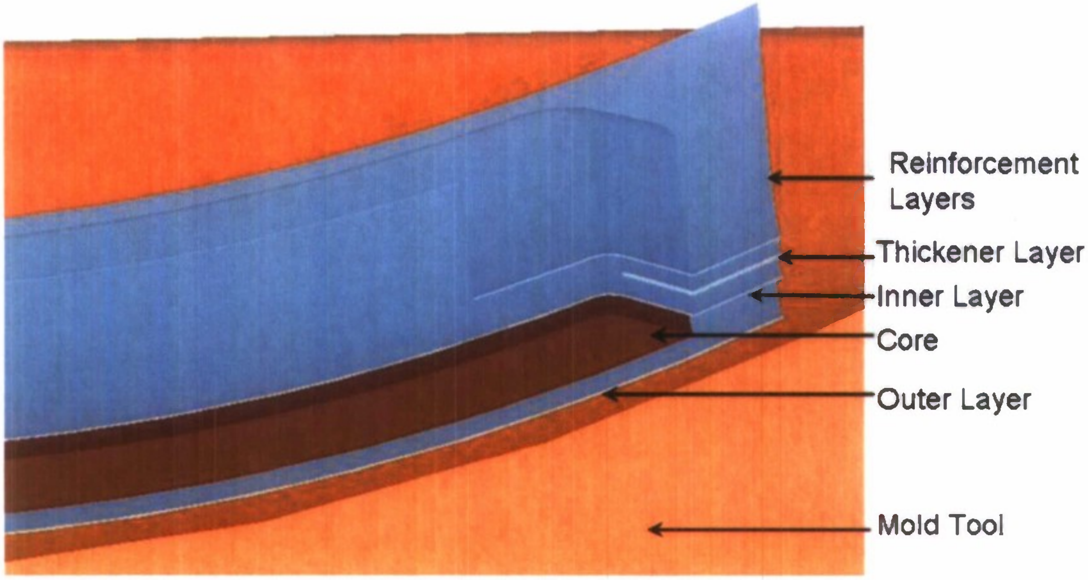


Fig. 1. Panel Geometry.

In order to test the panel under hydrostatic pressure it needs to be attached to a test fixture. After the panels were vacuum infused, they were trimmed and bonded to a steel test fixture interface that was bolted to a hydrostatic pressure tank. The fixture essentially consisted of a welded steel collar ending in a stainless steel "bonding plate" to which the double curvature sandwich panels were bonded, Fig. 2. The fixture mimics the panel attachment to steel bulkheads and longerons on a full-scale hybrid ship that is presently being built at Lehigh University.

Fig. 2. Test fixture.

2.1 Finite Element Analysis

Ansys Academic Teaching Advanced 11.0 was used for the modeling and Finite Element (FE) analysis. Due to symmetry only one quarter of the panel was modeled. The shape of the outer skin of the double curved panels was defined by

$$z = \frac{x^2}{2r_a} + \frac{y^2}{2r_b} \tag{1}$$

where x, y, and z are Cartesian coordinates, and r_a and r_b are parameters related to the radii of curvature of the panel about their respective axes[1]. The edges of the panel were at $x=+/-a/2$, and $y=+/-b/2$, where a is the length and b is the width of the panel.

The panel design can be broken up into four parts: inner (facing) skin, foam core, various reinforcement layers along the perimeter of the panel, and the outer (backing) skin. As already mentioned, this panel design shown in Fig. 1, where the foam cores tapers off to a single skin edge is similar to composite panels on some steel/composite hybrid ship hulls [13-16].

The thickness of the single skin flange was designed by limiting the average shear stress τ to 10 MPa, and thus

$$\tau = \frac{Pab}{2(a+b)t_{flange}} \leq 10MPa \tag{2}$$

where τ is the average shear stress, P is the pressure, t_{flange} is the total thickness of the flange (including inner and outer skins, thickener layers, and reinforcement layers), and a and b are the panel width and length, respectively. The fact that the perimeter length increases with curvature was ignored. Rearranging Eq. 2 yields:

$$t_{flange} = \frac{Pab}{2(a+b)\tau} \tag{3}$$

[1] For this study, r_a is always equal to r_b and r_a is used

Sandwich structures can be analyzed using either 2D shell elements, 3D solid elements, or a combination where the sandwich core is modeled with 3D solid elements while the skins are modeled with 2D shell elements. A comparative study between two different modeling approaches was made - using 3D brick elements for the core and 2D shell elements for the skins and using 2D shell elements for the complete sandwich (skins and core). The conclusion was that they produced similar results for the panels under investigation. Both approaches are used subsequently in this paper - 2D elements only for a larger parameter study, and 3D elements for the core and 2D for the skins for some select panels. The results from the comparative analysis can be seen in Fig. 3; where the x axis label corresponds to different locations where strain was measured and the y axis is the strain value. The data will be discussed in greater detail later.

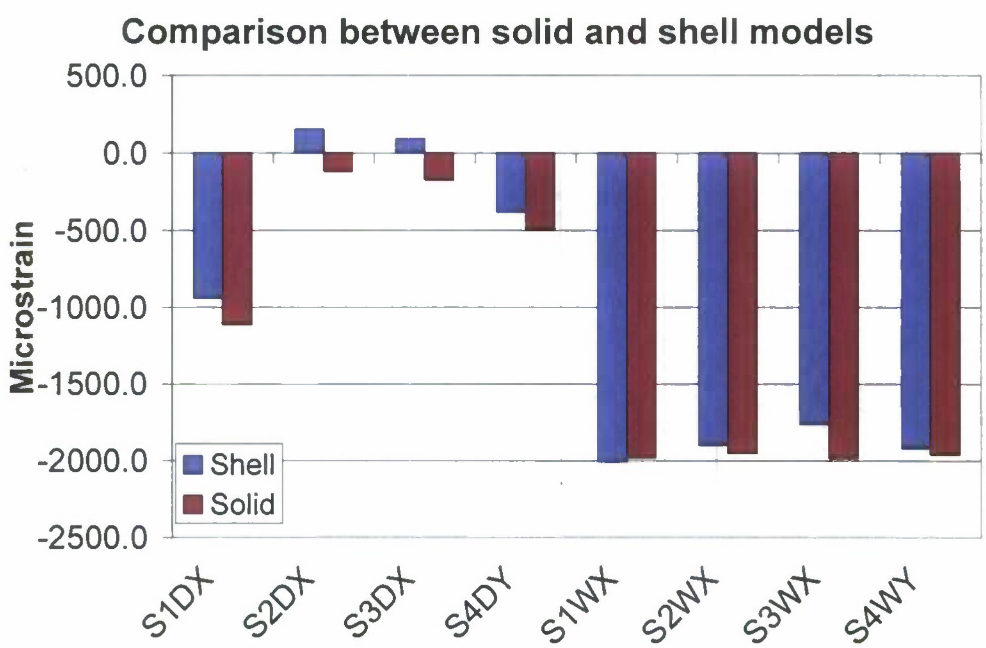


Fig. 3. Solid and Shell results for strain gage locations.

For the solid model (3D core and 2D skins), Solid95 elements were used to model the core and Shell91 elements were used to model the face sheets. Solid95 is a higher order version of the 3D 8-node solid element Solid45. It can tolerate irregular shapes without losing much of its accuracy. Solid95 elements have compatible displacement shapes and are well suited to model curved boundaries, [16].

For the 2D shell model, 8-node, Shell91 elements were used, with the 'sandwich logic' option turned on. The Shell91 elements are defined by layer thicknesses, material direction angles and orthotropic material properties. The total thickness of each element must be less than twice the radius of curvature and when using sandwich logic the core must be at least 5/6 the total thickness. Sandwich logic is specifically designed for sandwich construction with thin face sheets and a thick and relatively compliant core. The core is assumed to carry all of the transverse shear and the face sheets are assumed to carry all (or almost all) of the bending load [17]. The 45 degree taper of the foam core was modeled by modifying the 'real constants' (specifically the lay-up details) of the layered Shell91 elements and the nodes were located on the bottom surface of each element so the taper and flange were in the correct position. Using the sandwich logic option for all the elements that included the foam core meant the 45 degree taper was defined with the thicker half having the sandwich option turned on and the thinner half having the sandwich option turned off (since the thinner half of the tapered foam core would not be at least 5/6 of the total thickness). Unlike in the flat section, in a tapered section of a sandwich composite, shear forces are also resisted by the face sheets due to the angle of inclination of the taper with respect to the applied load [18,19]. There is a

coupling between the axial and flexural response that is inherently accounted for in a 3-Dimensional analysis. Vel et al. [20] describes the coupling in detail and provides formula for computation of the coupling coefficients in plate analysis of tapered sandwich composites.

2.2 Parameter study

Numerous numerical simulations were run with varying skin thickness and skin material, core thickness and core material, radii of curvature of the panel, and length-to-width aspect ratios to study their effect on panels with double curvature. For this parameter study, a simplified sandwich model was used with face sheets on either sides of a foam core. The previously mentioned tapered model will be discussed later. The simplified model used 2D shell elements with isotropic material properties. For the (fiberglass) skins, Young's modulus of E=30 GPa and Poisson's ratio of ν=0.3 were assumed. The material properties for the different PVC foam cores are listed in Table 1 (Poisson's ratio ν=0.32 was used for all foam densities).

Table 1. Core Material Properties [21].

Quality		H80	H100	H200	H250
Density	kg/m^3	80	100	200	250
Compressive Strength	MPa	1.4	2	5	6
Compressive Modulus	MPa	90	135	240	300
Tensile Strength	MPa	2.5	4	7	9
Tensile Modulus	MPa	95	130	250	320
Shear Strength	MPa	1.15	2	4	5
Shear Modulus	MPa	27	35	85	104
Shear Strain	%	30	40	40	40

The panels were modeled as a sandwich with the nodes located at the midplane and constrained in the z direction at the perimeter. A hydrostatic design pressure of 150 kPa was applied normal to the midplane of the panel. The panels were constrained at their edges. Symmetry boundary conditions were applied to the planes of symmetry at $x = 0$ and $y = 0$, respectively. In summary, the boundary conditions were u_z=0 at the edge of the panel; u_x=0 and θ_y=θ_z=0 at x=0; and u_y=0 and θ_x=θ_z=0 at y=0.

Various simulations were run looping over different variables to study their effect on doubly curved panels. Throughout the study, isotropic material properties, PVC foam cores, and a 150kPa pressure loading were used. The variations of parameters consisted of face sheet stiffness, face sheet thickness, foam core thickness, foam core strength, boundary conditions and length to width aspect ratio. Some of these parameters were studied in detail but for conciseness, the following are mainly discussed; The face sheet stiffnesses were either 30 or 100 GPa to represent GFRP or Carbon Fiber Reinforced Plastic (CFRP). The face sheet thicknesses were either 0.5mm or 2mm. The core thicknesses were either 12.7mm (thin) or 50.8mm (thick). The different foam core types used properties from DIAB Inc's Divinycell H-Grade PVC foam cores of either H80 or H250; table 1. Boundary conditions (BC) of clamped, hinged, or simply supported were used. The length to width aspect ratio varied between 1 and 2.

Reduction in the dimensionality of the model was achieved by using the following dimensionless parameters: a/r_a where a is panel length and r_a is radius of curvature, U/U_{flat}, where U is the max panel deflection in the z direction and U_{flat} is the max panel

deflection in the z direction of a flat panel with otherwise identical parameters, t/d where t is the skin thickness and d is the core thickness, and E_s/E_c where E_s is Young's modulus of the skins and E_c is Young's modulus of the core material.

Fig. 4 shows how the U/U_{flat} varies with curvature for different t/d and E_s/E_c values; the skin thicknesses, t, are held constant while the upper two curves have a 50.8mm thick foam core, the middle two curves have a 25.4mm thick foam core and the lower two curves have a 12.7mm foam core. Each of the three foam core thicknesses have a combination of either H80 foam core with CFRP face sheets or H250 foam core with GFRP face sheets; E_s/E_c of 1080 and 97 respectively. Generally speaking, panels with more curvature (smaller r_a) deflect less than flat panels, the core thickness affects the shape of the curve, and higher E_s/E_c values translate the graph down. For example, the deflection of a panel with a thin light foam core and stiff skins in comparison to a flat panel is much more affected by an increase in curvature than a panel with a thick, denser foam core and less stiff skins. However, it should be noted that for certain configurations curved panels deflect more than a corresponding flat panel, a phenomenon which will be discussed later in greater detail.

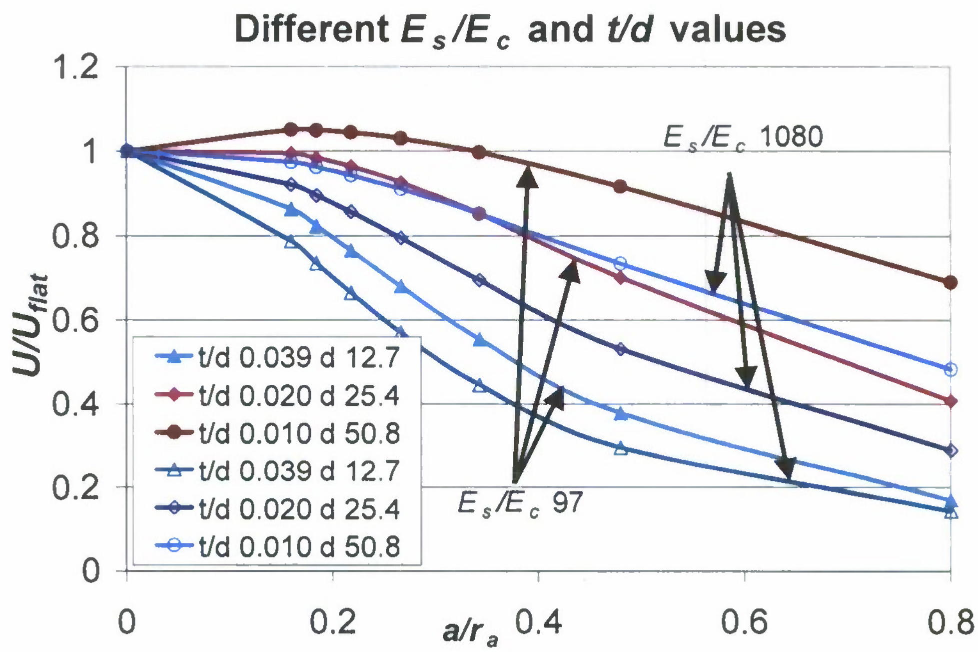


Fig. 4. Comparison between different *Es/Ec* and *t/d* values.

The effect of curvature on panels with different boundary conditions was studied by looking at curved panels with the following boundary conditions. Simply supported, or u_z=0 along the edge $x=a/2$, and along the edge $y=b/2$ was the first boundary condition denoted BC1. The boundary conditions were changed from simply supported to hinged, BC2, by allowing the edges of the panel to rotate but not translate or $u_x=u_z=0$ along the edge $x=a/2$, and $u_y=u_z=0$ along the edge $y=b/2$. Changing the boundary conditions to clamped, BC3, or $u_x=u_y=u_z=0, \theta_x=\theta_y=\theta_z=0$ along the edge $x=a/2$, as well as along the edge $y=b/2$ is shown in Fig. 5. The boundary conditions have a large effect on the deflection of the panel. The 12.7mm thick core with BC1 is on top of the 25.4mm thick core with BC2 which is important because thinner cores are usually affected more by curvature than thicker cores for every other condition shown.

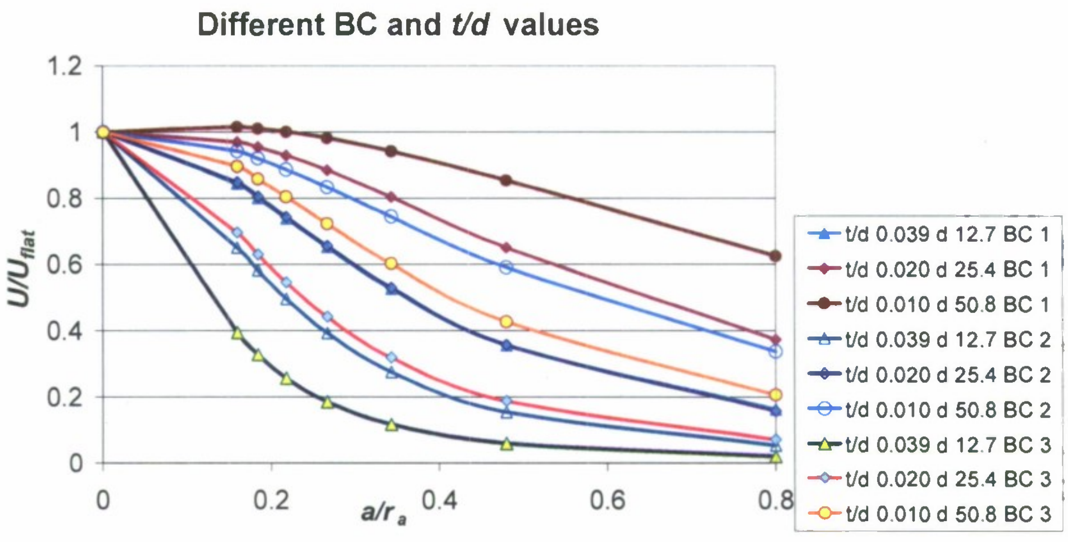


Fig. 5. Comparison between Boundary Conditions (BC) and *t/d.*

In Fig. 6 there is a comparison between different face sheet thicknesses over three different core thicknesses. It appears that a thicker face sheet has a bigger effect on thicker cores compared to thinner cores.

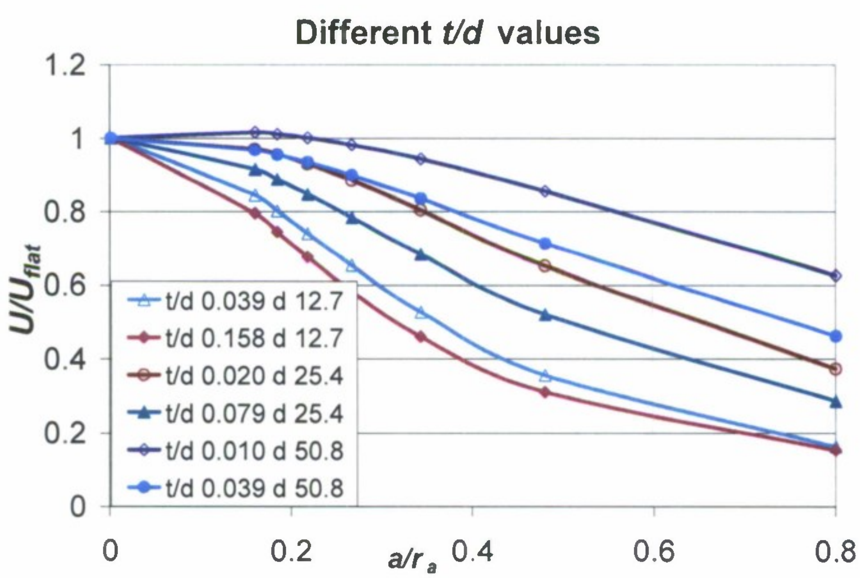


Fig. 6. Varying skin thicknesses for each core thickness.

Fig. 7 shows the effect of changing the aspect ratio from a square to a rectangular panel. The value of a is fixed at 0.6m and b changes. The ratio listed for each curve is the ratio of the panel width divided by its length, b/a. For example, the point [0.48,0.70] on the graph corresponds to curve t/d 0.01 H80 d 0.0508 ratio 2 has a face sheet thickness, t, of 0.5mm, an H80 foam core thickness, d of 50.8mm, a r_a=r_b of 1.25m, an a of 0.6m, b of 1.2m. For the case shown in fig. 7 with a being constant and r_a=r_b making the panel longer in one direction makes the curvature have a greater effect on the stiffness.

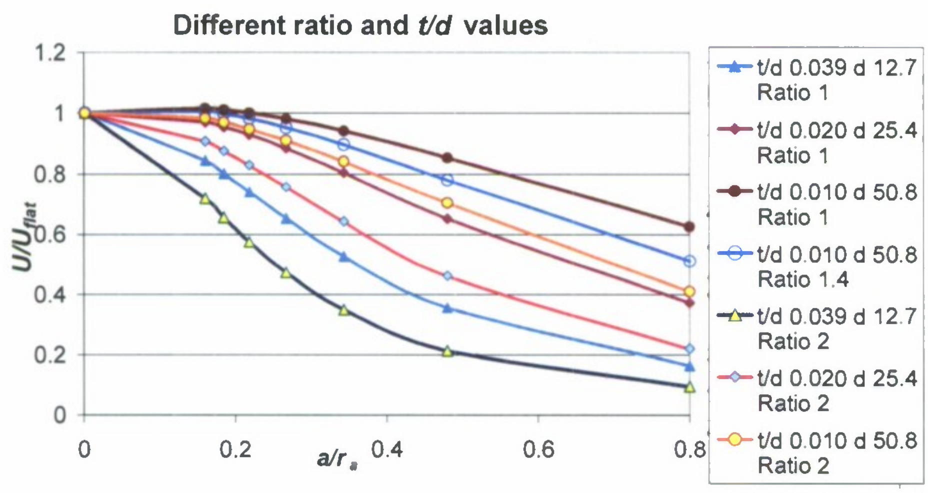


Fig. 7. Different length to width ratios.

Fig. 8 shows the percent deflection by taking the maximum deflection and dividing it by the shortest panel side for different face sheet and foam core types and foam core thicknesses. It can be seen how thinner cores are influenced much more by curvature than thicker ones.

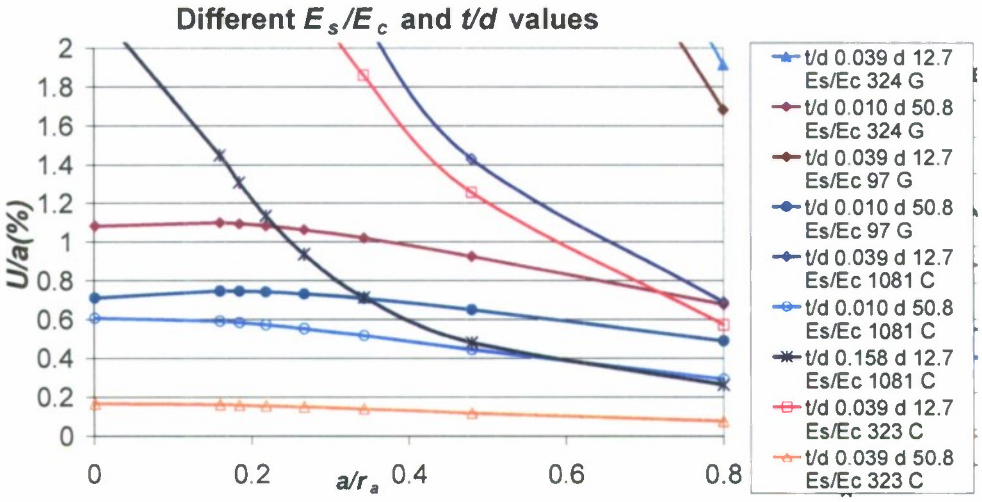


Fig. 8. *U/a* vs. a/r_a.

From the results of the parameter study, a better understanding of panel behavior was obtained, which led to further FEA analysis using a more detailed FEA model followed by the selection of the panel design to be tested experimentally. Design requirements of a hydrostatically pressure loaded panel were implemented to select a panel design. Once the panel design was selected, the testing fixture was analyzed. The panel and fixture design are discussed in the next two sections.

2.3 Design of Panel for Experimental Investigation

The design requirements of a pressure loaded ship hull panel includes stiffness and strength requirements. A typical stiffness requirement may be that the maximum deflection is less than, L/50, where L is the length of the shorter side of the panel. A strength requirements may be that the shear stress in the core is less than the allowable shear strength, that the tensile strains anywhere in the skins are less than the allowable

tensile failure strain of the face skins, and that the compression strains in the skins are less than both the allowable compression failure strain and the wrinkling strain. At present the 2% deflection requirement was used. The shear strengths of the cores were taken from the manufacturer (Diab Inc.) and are given in Table 1. The failure strains of the skins were assumed to be 1.3 % in tension and 1.3 % in compression. The wrinkling strain may be estimated by the Hoff and Mautner [22] wrinkling formula:

$$
\begin{aligned}
&\sigma_{wr} \approx 0.5(E_S E_C G_C)^{1/3} \\
&\varepsilon_{wr} \approx \frac{\sigma_{wr}}{E_S} \approx 0.5\left(\frac{E_S E_C G_C}{E_S^3}\right)^{1/3} \\
&\approx 0.5\left(\frac{E_C^2}{E_S^2 * 2(1+\upsilon_C)}\right)^{1/3}
\end{aligned}
\tag{4}
$$

Where σ_{wr} is the wrinkling stress, ε_{wr} is the wrinkling strain, and υ_C is the Poisson's Ratio of the core. The criterion varies with foam core density but the two that were considered, Divinycell H80 and H100, give wrinkling strains of 0.9% and 1.1%, respectively.

There may be further requirements on impact, in particular on the outer skin of a ship hull. This typically leads to thicker skins on the outside than on the inside of the sandwich panels, resulting in an unsymmetric response with respect to the mid-surface where axial and flexural responses are coupled. At present, explicit impact requirements were not included but the outer skin was forced to be 50% thicker than the inner skin. This is reasonable for many applications.

Panels were initially modeled as described in the parameter study above, i.e., modeling only the panel (no steel fixture or other complex support), assuming simply supported edges, and using 2D shell elements. The experimental design panel model had additional detail. The tapered foam core where the inner and outer skins came together to a single skin was added to the model along with reinforcement layers and orthotropic GFRP face sheet properties, Table 2.

Table 2. 3-D Orthotropic material properties used in Ansys.

Ex	22	GPa
Ey	22	GPa
Ez	5.5	GPa
Gxy	4	GPa
Gxz	2	GPa
Gyz	2	GPa
PRxy	0.275	-
PRxz	0.275	-
PRyz	0.275	-
Dens	1800	kg/m^3

The lightest panel configuration that fulfilled the design requirements on stiffness, core strength and skin strength had the following parameters: curvature r_a=r_b =0.75m, 0.75 mm outer skin, 0.5 mm inner skin,18 mm H80 foam core, and 2.25 mm thick flange where the foam core tapers to a single skin. This panel configuration was chosen as the final panel to manufacture and experimentally test under hydrostatic pressure, Table 3. The properties of this panel were used to design the steel fixture for testing which is discussed in the next section.

Table 3. Properties of panel chosen to be manufactured.

Density (kg/m^3)	Outer skin (mm)	Inner Skin (mm)	Core (mm)	R	Max z-Deflection (mm)	Mass (kg)
80	0.75	0.5	18	0.75	11.0	1.283

2.4 Fixture Analysis

The previously mentioned fixture to attach to the panel for testing was designed using Solidworks 2006 educational version and analyzed using Cosmosworks 2006 educational version. A quarter of the fixture and a homogenized sandwich panel that was modeled using shell elements. The bolt holes were constrained to have no displacement, and symmetry boundary conditions were applied to the planes of symmetry at $x = 0$ and $y = 0$, respectively. In summary, the boundary conditions were $u_x=u_y=u_y=0$ at the bolt holes; $u_x=0$ and $\theta_y=\theta_z=0$ along the edge $x=0$; and $u_y=0$ and $\theta_x=\theta_z=0$ along the edge $y=0$. The "bond plate" to which the sandwich panels were adhesively bonded was given the properties of AL-6XN stainless steel whereas the rest of the test fixture was given the properties of mild steel. There were no sandwich elements available for the analysis using Cosmosworks, instead, the sandwich panel was modeled as a homogeneous material with a modulus and thickness such that it had the same in-plane stiffness and bending stiffness as the previously mentioned sandwich panel designed to be experimentally tested. The following two approximate formulas were used to calculate the material properties and thickness of the homogenized panel:

$$E_{\text{hom}} * h_{\text{hom}} = E\ (t_1 + t_2)$$

$$E_{\text{hom}} * \frac{h_{\text{hom}}^3}{12} = (t_1 h_1^2 + t_2 h_2^2) E \qquad (5, 6)$$

E_{hom} and h_{hom} are Young's modulus and the thickness of the homogenized material, E is Young's modulus of the fiberglass, t_1 and t_2 are outer and inner skin thicknesses, h_1 and h_2 are the outer and inner distances from the midline of the outer and inner skins to the neutral axis. The calculated E_{hom} and h_{hom} of the homogeneous material equivalent to the designed sandwiched panel were 0.87 GPa and 32 mm respectively. These equivalent material properties were used in the analysis of the test fixture. A summary of material properties used in the fixture analysis is given in Table 4.

Table 4. Fixture analysis properties.

Part	**Thickness (mm)**	**Material**	**Ex (GPa)**	**PRxy**	**Density (kg/m3)**
Bottom Plate	9.5	Carbon Steel	210	0.28	7800
Wall Plates	4.8	"	"	"	"
Bond Plate	2	AL6XN	195	0.28	8060
GFRP Flange	2.25	Fiberglass	22	0.275	1800
Homogeneous Material	32	Homogeneous	0.87	0.3	1000

The test fixture Finite Element analyses in COSMOSWorks used a pressure of 300kPa, twice the design load, applied normal to the surfaces of the panel, the fiberglass flange, and outer plates. The COSMOSWorks FFEPlus solver was used and results verified that the fixture would not reach the material's yield stress. In order to make the fixture lighter

and easier to transport, the actual fixture had a 4.8 mm thick bottom plate with a 12.7 mm thick bolt plate instead of the 9.5 mm thick bottom plate as used in the analysis. Making two plates instead of one was done so the fixture was lighter and easier to transport and the thicker bolt plate reduces the deformation of the fixture under higher pressures. The side plates were MIG welded to the bottom plate. The bond plate was formed to the curvature of the sandwich panel and welded. It should be noted that the bond plate overhung the inner wall plate by 5 mm to improve weldablity. The overhang was not ground flush with the inner wall plate in an attempt to slightly reduce a stress concentration at the interface and create a small stiffness gradient. By making the stainless steel bond plate overhang a small amount, there is the high stiffness box beam transitioning to a thin, unsupported piece of stainless steel, to a single fiberglass skin. The welded fixture is shown in Fig. 2.

The designed panel for testing was manufactured, followed by the steel test fixture. They were then joined and tested under hydrostatic water pressure which is discussed in the following sections.

3. Manufacturing of doubly curved sandwich panels

In order to fabricate the panel with properties listed in Table 3, a doubly curved mold with a radius of curvature of 0.75m needed to be made. Several methods for making molds were studied. The requirements were that the mold material should be inexpensive and easy to transport, the mold should be easy and inexpensive to machine, the geometric

tolerances should be good, the mold should be non-metallic, and it should be compatible with vinyl ester such that panels could be vacuum infused directly in the mold. Considering the requirements and facilities at hand, Renicell E320 polycarbamate foam from Diab Inc. was chosen for the mold material. This is an inexpensive, easy to machine material with a density of 320 kg/m^3. It can be obtained in blocks thick enough for the present panels. The mold was designed using Solidworks to have the same curvature as the doubly curved panel, and large enough to lay up materials, infusion and vacuum hoses, etc. and fit a vacuum bag. The Renicell was machined in-house using a 5-axis CNC router, Fig. 9.

Fig. 9. CNC machining of mold.

The foam cores for the sandwich panels needed to be formed to have close to the same curvature as the desired finished panels for testing (r_a=r_b=0.75m, eq. [1]). This was due

to the significant curvature and test trials indicated a flat foam core could not be successfully vacuumed down to the mold, infused, and maintain the desired geometry. In principle, one could machine the foam core from a 125mm thick block of foam, but it would be very expensive and quite cumbersome. Another option was to machine the foam core to the correct flat size, soak it in acetone until it became soft, vacuum the softened foam core to the curved mold surface, and hold under vacuum until the acetone evaporated from the core. This method shapes the foam core very nicely but the effects of acetone on the foam core properties are not fully known [23]. Rather, the foam cores were formed by first machining the flat foam cores, then applying heat until they softened, and then vacuuming them down to the mold and let cool. The 18mm thick foam cores were machined with 45° beveled edges with 55mm corner radii, 15mm radius fillets on top of the bevels, and 2x2mm infusion grooves spaced 25mm apart machined on the top and bottom. These machined cores were thermoformed by heating with IR heaters, placed into the Renicell mold, and vacuumed until cooled. After several trials to fine tune the process, the foam core conformed very nicely to the mold with very little springback. The thermoforming presumably leads to no noticeable change in core/face sheet adhesion and only a small change in structural properties [24].

The face sheets consisted of three different types of glass fiber reinforcements; Hexcel 7725 which is a 2/2 twill with a surface weight of 298 g/m^2, Owens Corning Knytex WR24-5x4 woven roving at 815 g/m^2, and Owens Corning M-8610 continuous filament mat at 450 g/m^2. The Hexcel was used as the inner and outer skins as well as reinforcement layers around the tapered portion of the foam core. The woven roving and

the continuous filament mat were used to thicken the flange in order to improve the strength close to where the panel was bonded to the test fixture. The continuous filament mat was also used as resin flow medium. The woven roving fabric had a fabric weight of 815 g/m^2, with 440 g/m^2 in the 0° direction and 375 g/m^2 in the 90° direction. The material properties are shown in Table 5 [25]. The matrix was Ashland Derakane 8084 vinyl ester epoxy resin, mixed with Cobalt Naphthenate-6% (CoNap), Dimethylaniline (DMA), Methylethylketone peroxide (MEKP), and 2, 4-Pentanedione (2, 4-P). The CoNap and DMA promote the reaction, MEKP is the hardener, and 2,4-P is an inhibitor used to increase the gel-time. The weight percentages added of each chemical recommended by the manufacturer for 80 °F are as follows: 1.5% MEKP, 0.025% DMA, and 0.15% CoNap [26]. When 2,4-P is added, more CoNap is recommended and therefore 0.2% CoNap was used. The amount of 2,4-P generally varies between 0.13% and 0.5% depending on the desired gel time. At present, 0.14% of 2,4-P was used to give roughly a 3-4 hour gel time.

Table 5. WR24-5x4 Knytex glass fiber typical material properties [4].

Material Properties of Laminate based on 50% glass content by weight		
Tensile Strength	MPa	289
Tensile Modulus	GPa	14.3
Compression Strength	MPa	230
Compression Modulus	GPa	15.7
Flexural Strength	MPa	385
Flexural Modulus	GPa	15.2

3.1 Panel lay-up

In order to successfully manufacture the doubly curved foam core and infused sandwich panel to the determined design, two panels were made to shakedown the manufacturing process and testing. The doubly curved panels were made by first laying the dry glass fiber into the previously mentioned Renicell high density foam mold. However, vinyl ester adheres to Renicell foam. A protective surface was made by covering the mold with vacuum bag and evacuating the air. The vacuum bag was challenging to make to conform to the doubly curved mold surface with no wrinkles. When wrinkles did form, they were pushed to the edges of the mold to leave a smooth mold surface. Three layers of Hexcel 7725 bi-directional fabric were laid in the vacuum bagged mold to make the outer skin, and then the thermoformed foam core was positioned on top of the outer skin layers, followed by the inner two layers of Hexcel 7725, Figs 10-12.

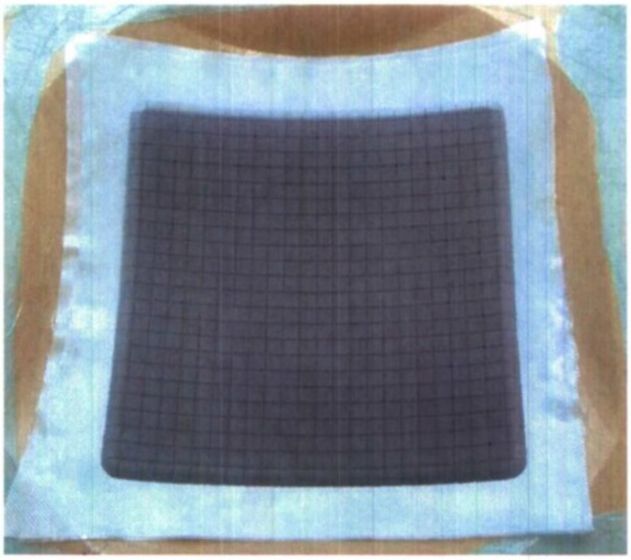


Fig.10. Thermoformed foam core in mold.

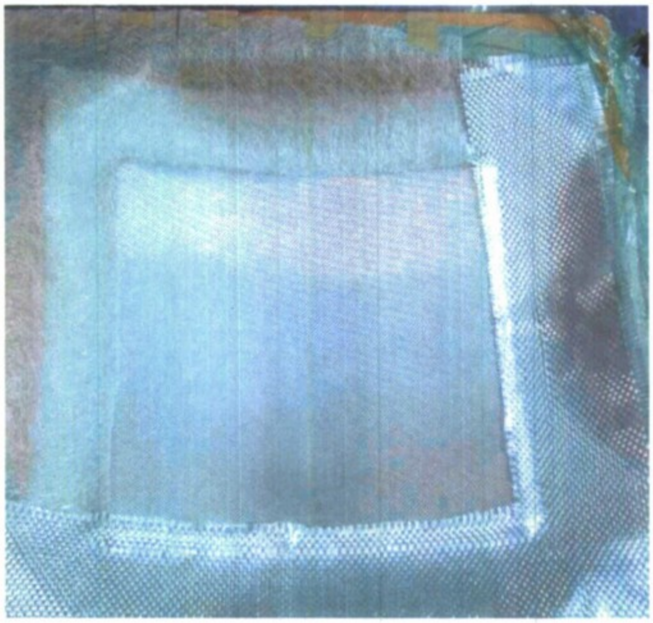


Fig.11. Thickening layers of WR-24 and CFM.

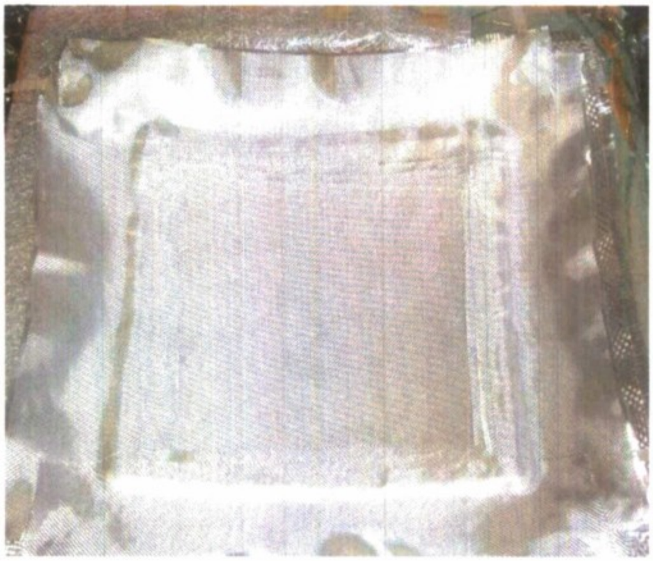


Fig. 12. Laying up the reinforcement layers.

In order to make the single skin flange thick enough to withstand the loading, a thickener layer was added. This thickener layer consisted of one layer of continuous filament mat

on the side closest to where the resin was introduced and two layers of WR-24 on the opposite side, Fig. 11. Reinforcement layers of Hexcel 7725 were then laid over the thickener layers to transfer the load up the beveled edges. The three reinforcement layers were staggered by 15mm, starting at 45mm from the edge of the top of the bevel, Fig. 12.

After the fiber reinforcements of the inner skin had been laid down, the complete panel was covered with peel ply and breather. The former was used so the breather can be removed from the part after cure, and the latter was used to entrap air bubbles from any leaks during infusion as well as to promote saturation of the fiber reinforcements. Resin distribution medium was used on top of the breather from the infusion tubes to the bevel along two edges of the panel. The lay-up was then covered with a vacuum bag and evacuated of air, Fig. 13. In order to reduce the risk of air leaks further, the bagged part was covered with breather and another vacuum bag and evacuated of air (not shown).

Fig. 13. Vacuum bagging.

The vinyl ester resin was mixed for 5 minutes, degassed for 15 minutes and then the resin was infused through the dry fibers by vacuum. It took approximately 35 minutes to infuse a panel. After the infusion had completed the resin line was then closed off and the pressure under the vacuum bag was allowed to equalize. The vacuum pressure was then slowly reduced using a vacuum regulator, from essentially pure vacuum to 25kPa absolute pressure, to reduce the chances of the vinyl ester boiling. The part was left under vacuum for 24 hours and then demolded and trimmed to the correct size using an abrasive waterjet cutter, Figs. 14 and 15. Each panel was then instrumented and the surface was prepared for bonding as described below.

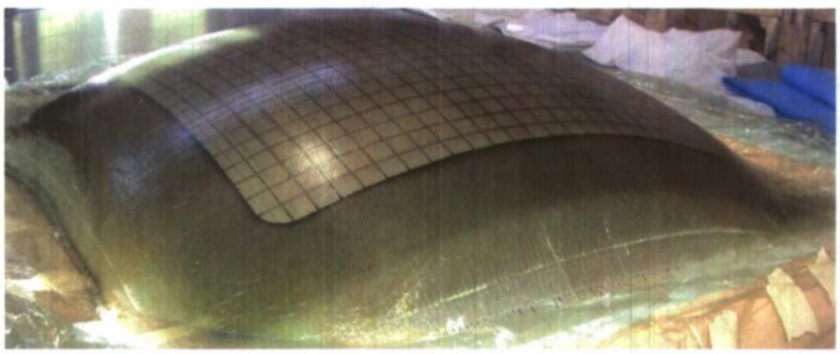


Fig. 14. Demolded part.

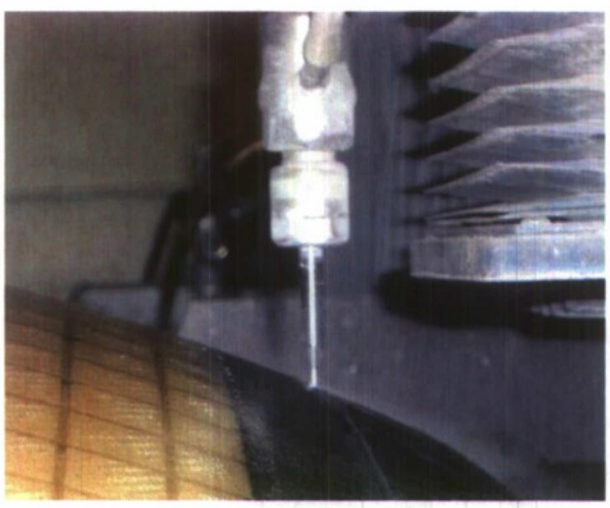


Fig. 15. Waterjetting to size.

3.2 Panel Preparation

The final doubly curved test panel was instrumented with eight Vishay CEA-06-500UW-350 strain gages. The shakedown panels were instrumented with one strain gage at the center of each panel on both the inner and outer skins. The strain gages were bonded to the final panel in eight locations, four on dry (inner) skin and four on wet (outer) skin. Vishay's recommended surface preparation and bonding techniques were followed [27]. The locations of the strain gages and their labels are shown in Fig. 16. The instrumented panels were then prepared for bonding to the test fixture.

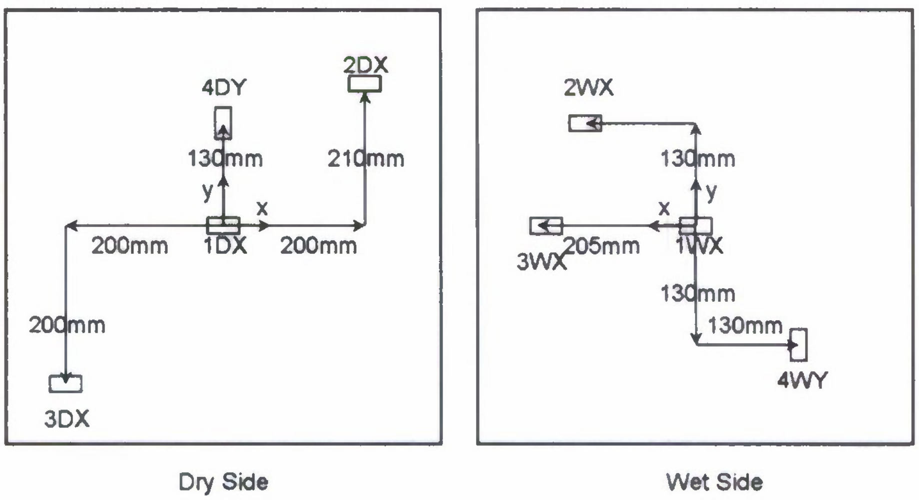


Fig. 16. Strain gage location and label.

Surface preparation is extremely important for the panel's performance and care was taken to promote a good panel/fixture bond [28]. The fixture's stainless steel bonding surface was prepared by grit blasting and thoroughly cleaned with trichloroethylene. The panel bonding surface was carefully sanded using 80 grit sand paper and then thoroughly

cleaned with trichloroethylene. An epoxy paste adhesive, SIA E2119 A/B, was used to bond the panel to the fixture. E2119 is a 1:1 two-part toughened epoxy adhesive that will achieve handling strength in less than 8 hours and full cure in 72 hours at room temperature [29].

The SIA epoxy adhesive was applied, using a pneumatic gun with a static mixing nozzle, to the fixture and panel and evenly spread over the bonding surfaces. An extra bead of epoxy was applied down the middle of the bonding surface to assure a sufficient bond line thickness and to make sure excess epoxy forced out any air when the panel was mounted to the fixture. The panel was placed on the steel fixture and fixed with duct tape, then turned over and placed in a CNC machined bonding jig. This jig had the doubly curved shape of the panel, but touched the panel only by the bonding surface. The jig was made of relatively soft Styrofoam which allowed for an even clamping pressure, Fig. 17. The steel jig was then weighted down with lead and left to cure. The adhesive cured for 14 hours while wrapped in an electric blanket, elevating the temperature to about 35C, then post-cured for 1.5 hours at 70C.

Fig. 17. Panel epoxied to fixture in Styrofoam jig.

After the epoxy was sufficiently cured between the panel and fixture, wire leads were attached to the previously installed strain gages and secured to the panel using silicon. The panel and fixture was then attached to the testing tank. To prevent leaking, a rubber gasket was placed in between the test tank and fixture and another gasket in between the fixture and bolt plate. Loctite 567 was applied to all bolts.

4. Testing

The curved panels were tested under hydrostatic loading at the Hybrid Structures Lab at the University of Maine. The instrumented panels, bonded to the test fixture which was bolted to the test tank, were repeatedly loaded and unloaded under increasing pressure until final failure.

4.1 Test Tank Design

The test tank was designed and manufactured at the University of Maine. The steel test fixture was designed to withstand 300 kPa water pressure, or twice the panel design load

of 150 kPa, without yielding. The test tank was designed for 450 kPa. The fixture, test tank and all connections were watertight. The tank consisted of MIG welded 835x240x25mm steel plates making up the walls, 835x76x12.7mm steel plates making up the top, and a 1090x1090x12.7mm steel plate for the bottom. In order to provide adequate stiffness to the sides and the bottom flanges, 240x101x9.5mm web stiffeners were welded onto the sides on 209mm centers. The tank was bolted to a stiffened 50mm thick steel plate to provide stiffness for the tank bottom. The top flange was drilled and tapped matching the bolt pattern of the test fixture. The overall dimensions of the tank can be seen in Fig. 18.

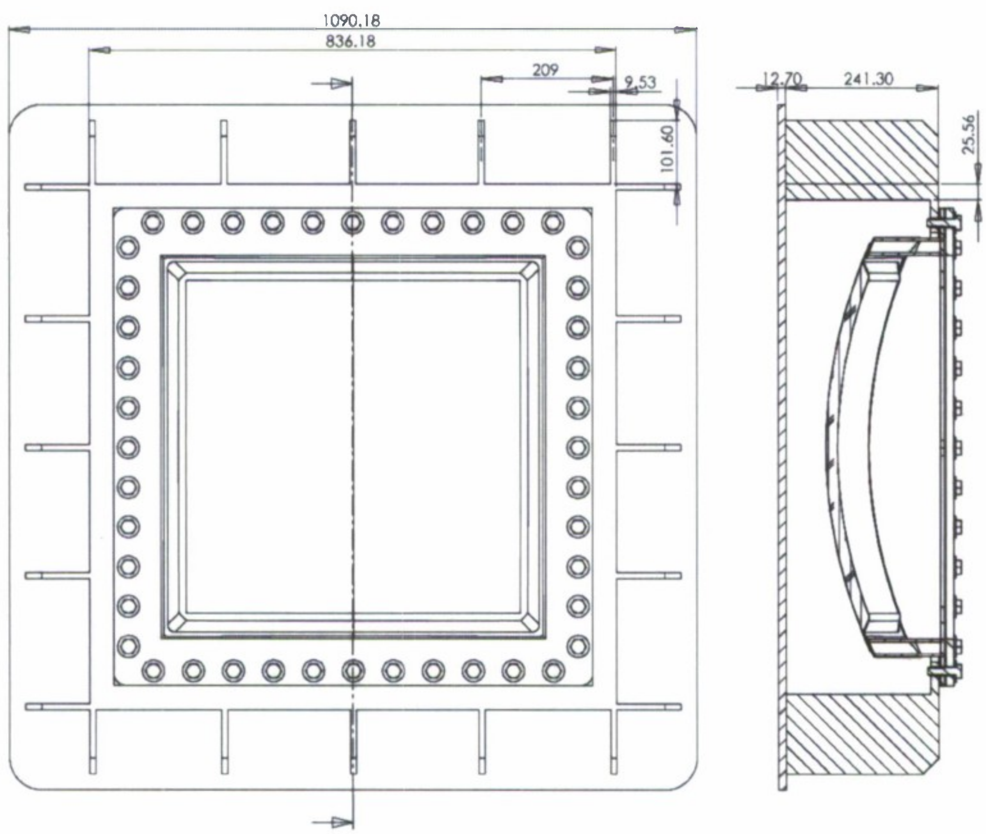


Fig. 18. Test tank schematic.

4.2 Instrumentation

The panels were instrumented with one linear variable displacement transducer (LVDT) at the center of each panel and eight metal foil strain gages. The applied pressure was measured using an Omega PX303 pressure transducer. Silicone was used to protect the solder joint from straining during panel installation and from the water pressure. The wire leads were soldered to cables connected to the data acquisition system and heat shrink was used to protect the solder joint during testing.

Data acquisition was carried out using a Pentium 4 computer with an IOTECH Daq-board 2000 card, and Vishay 2120 multi-channel strain signal conditioner. The system had 16 bit analog-to-digital conversion resolution and was capable of reading a total of 48 channels at a rate of 1 kHz, which was more then adequate for the present test. The data acquisition process was controlled using the DAQFI_D5 software, written at the University of Maine.

4.3 Testing Method

The doubly curved panel was tested at University of Maine's Hybrid Structures Laboratory, located in the Advanced Manufacturing Center using the previously described hydrostatic tank. An air-over-water method was used to load the panel due to its simplicity, safety and relatively low cost. It also allowed use of the laboratory's existing 827 kPa air supply. A 984-L pressure vessel, filled with water was the interface between the test tank and the compressed air. In order to insure that no initial hydrostatic

pressure was developed the vessel was filled to a height equal to the top of the test tank. A Control Air, Inc 700 precision, manual regulating valve was used to achieve the desired pressure level, by manually dialing in each pressure step, Fig. 19. A picture of the test setup is shown in Fig. 20.

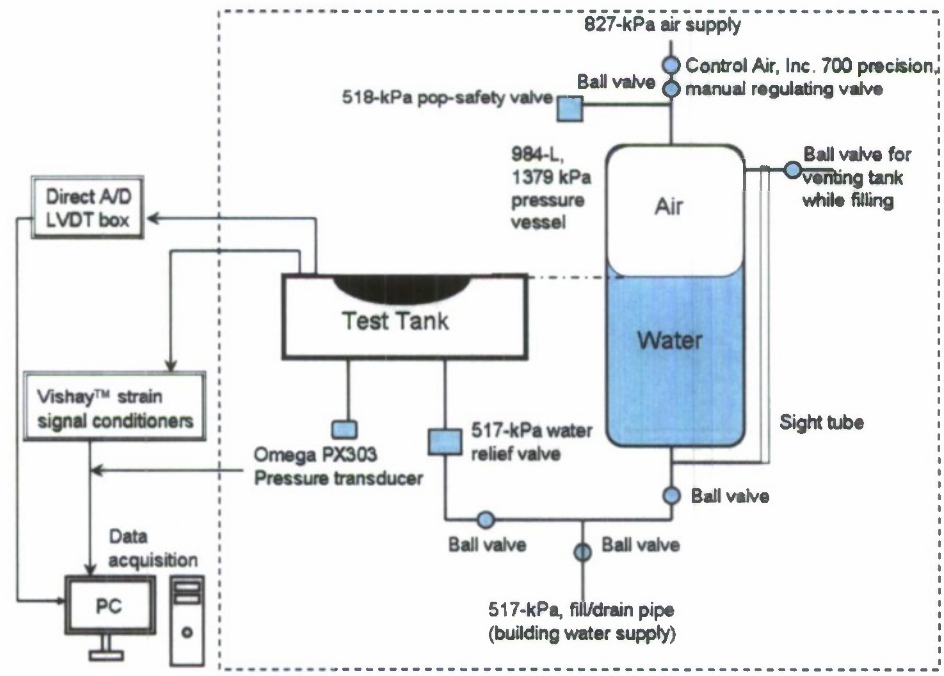


Fig. 19. Schematic of test setup.

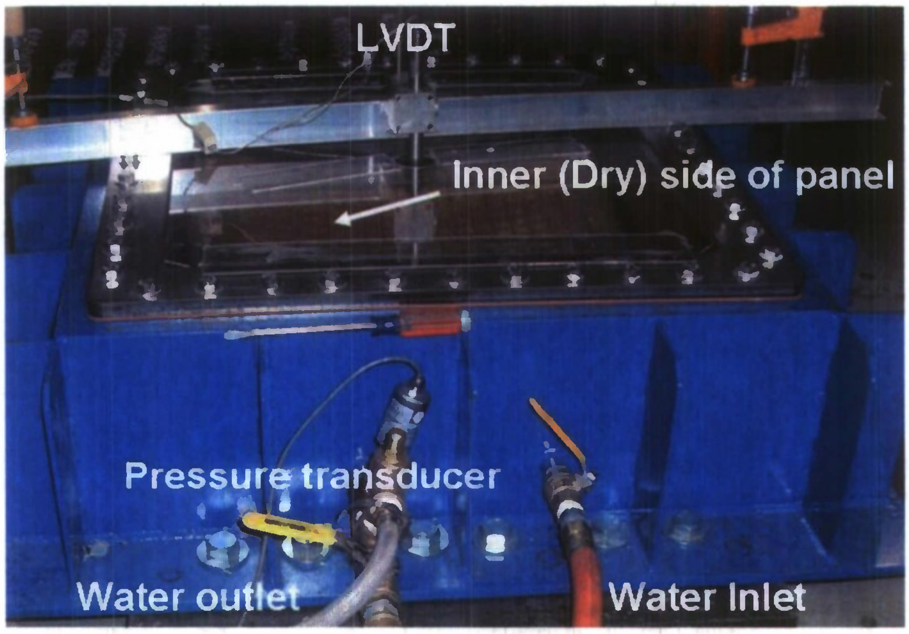


Fig. 20. Testing setup.

Two shakedown tests were preformed to test the data collection, instrumentation, and connections between the panel, fixture, and test tank. After the two successful shakedown tests, testing of the final panel assembly was conducted in a cyclic fashion. Cyclic testing was used to study the degradation of the structural system due to repetitive loading cycles and to assess the load level at which the onset of damage occurred. The test was composed of a total of five cyclic increment sets as shown in fig. 21. Each cycle set was comprised of two equal load cycles. The pressure was increased by equal increments of 40kpa until the design load was reached. After the design load was reached the pressure was increased by smaller increments up to 130% of the design load. The panel assembly was then tested to failure at 175% of design load. Load, displacement and strain data were recorded throughout the test.

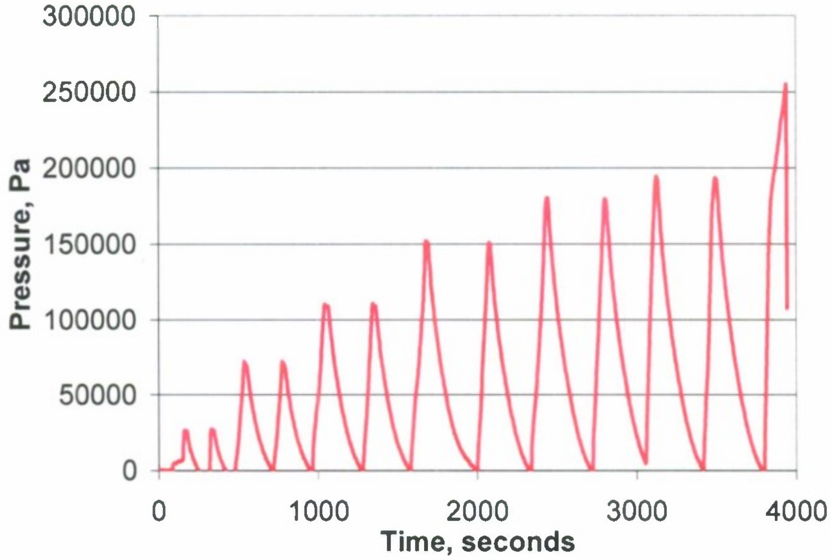


Fig. 21. Pressure History Plot.

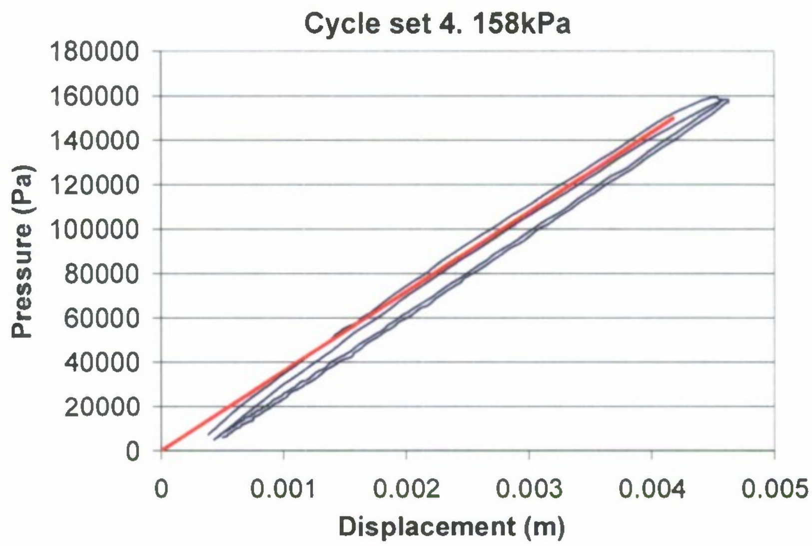
4.4 Results

The results of the hydrostatic pressure test are summarized in Table 6 for the design pressure of 150kPa. Central deflection and strains at various locations are provided.

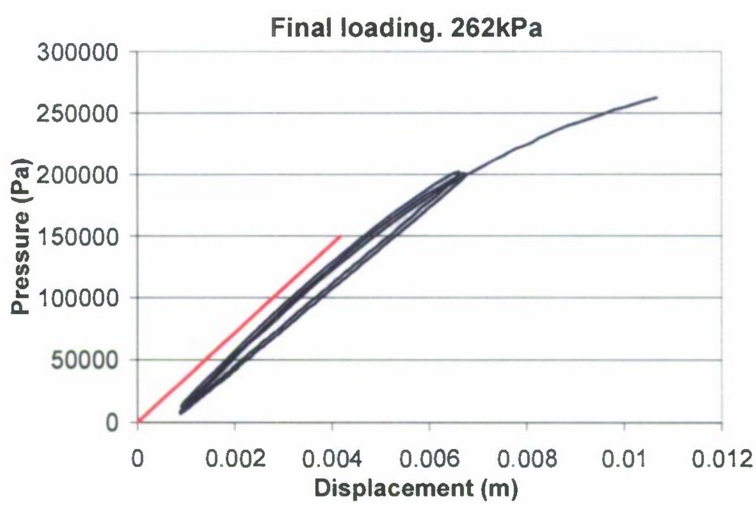
Table 6. Percent difference between Ansys and test results.

	Ansys	Test	Difference (%)
Center Deflection (mm)	4.18	4.13	1.2
Strain gages (microstrain)			
S1DX	-1362	-1351	0.8
S2DX	228	No data	-
S3DX	164	No data	-
S4DY	-616	-726	16.4
S1WX	-2318	-2260	2.5
S2WX	-2192	-2058	6.3
S3WX	-2143	-1979	8.0
S4WY	-2232	-2606	15.5

Table 6 shows the values the LVDT and strain gages recorded during the test, the Ansys solid model predicted results, and the percent difference between them. The Ansys predicted results are from the model with the entire fixture included in the analysis. Representative load verses displacement graphs are shown in Figs. 22a-b.



a)



b)

Fig. 22a-b. Load verses displacement curves for design and final loading.

The graphs correspond to center point deflection of the panel measured by the LVDT verses the pressure recorded by the pressure transducer. For clarity, only the design load (fourth) and the final load steps are shown. The fourth load step was to the design load of 150 kPa. Each graph shows close to linear behavior. There is some hysteresis which may be due to mechanical connections (rubber gaskets, bolts, etc moving slightly) and/or microcracks forming in the composite skins upon loading. The maximum displacements at peak load were 4.1mm and 10.7mm, respectively, for the fourth and the final load steps. The Ansys model predicted a centerpoint deflection of 4.2mm for the design load of 150kPa. 'Pings' typical of damage in composites were heard during the 5^{th} and 6^{th} load steps for the first time, and several more times before panel failure. During the testing, no leaks or visual damage were noticed in the panel, fixture, or test tank until final panel failure when water came rushing through the panel. The panel went from showing no sign of damage, except that several pings were heard, to complete failure so quickly it was difficult to determine the exact mode of failure or failure progression. The top and bottom of the panel after failure are shown in figs. 23 and 24.

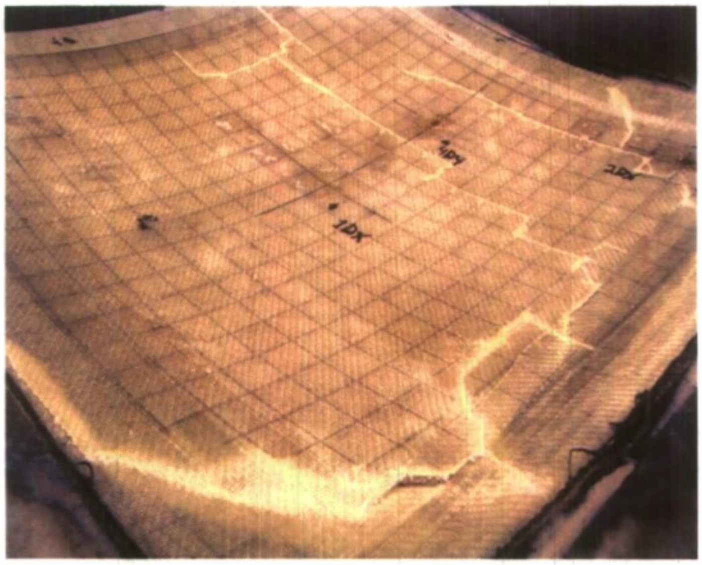


Fig. 23. Damaged Dry side.

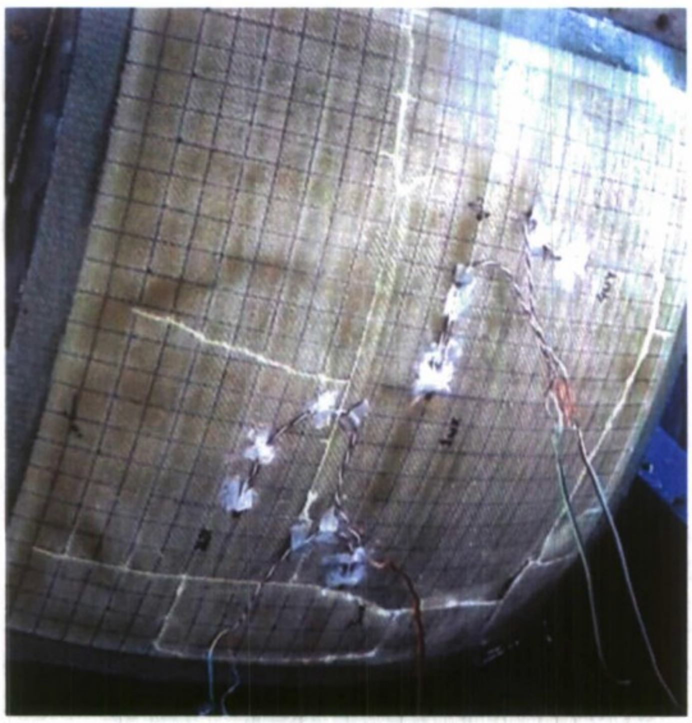
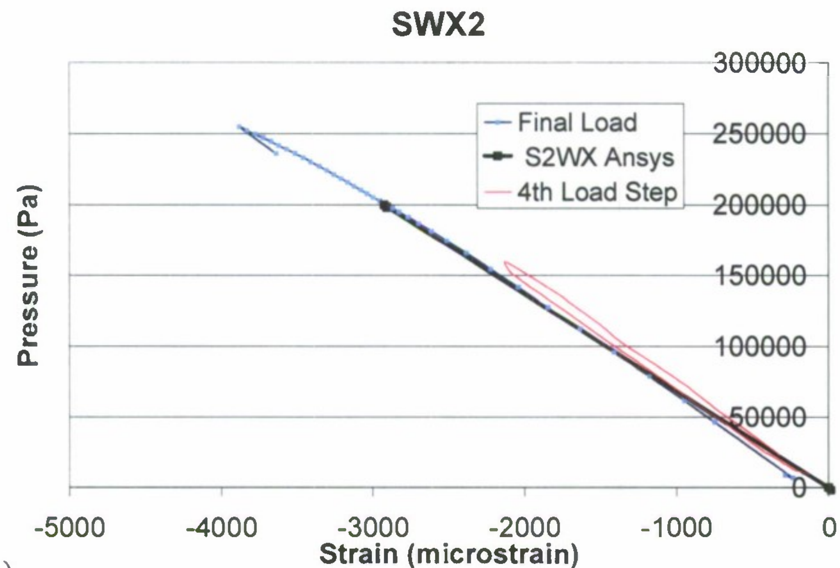


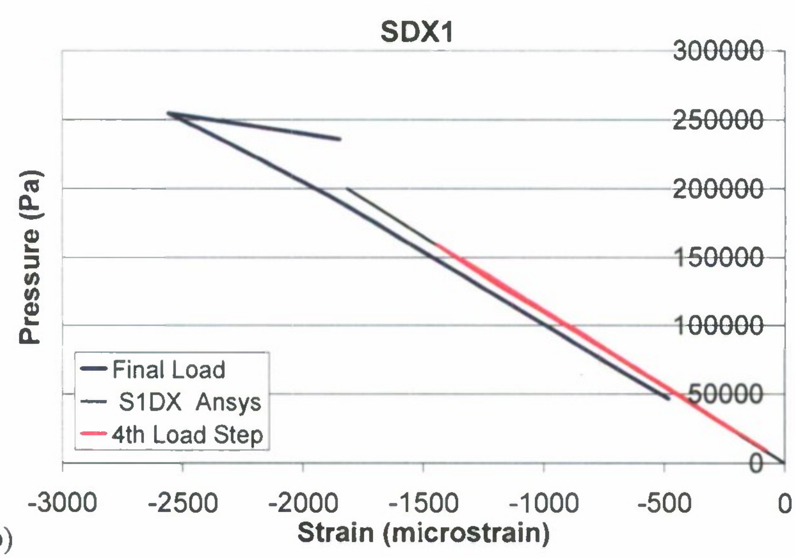
Fig. 24. Damaged Wet side.

Representative load verses strain curves are shown in figs 25a-b for the strain gages SWX2, located on the outer (wet) side 130mm in the x and y direction from the center of

the panel and SDX1 located on the inner(dry) side in the center of the panel. The recorded strain gage data showed good agreement with Ansys Finite Element results which are also plotted in Figs. 25 a-b.



a)



b)

Fig. 25a-b. Load verses strain for strain gage SWX2 and SDX1.

Based upon material coupon tests, a failure strain of 13,000 microstrain in tension and 9,200 microstrain in compression (wrinkling) was estimated for the composite panels. A maximum strain of 4,200 microstrain in compression was recorded by the strain gage at the bottom of the outer skin designated, WX1, which was considerably smaller than the predicted failure strain. This is believed to be due to the inherent waviness and thickness variation of a woven fiber reinforcement. The thickness variation reduces bending stiffness to a much larger extent than it reduces in-plane stiffness [30]. An appropriate wrinkling formula would use bending stiffnesses rather than in-plane stiffnesses, as was used in eq. (4). The low failure strain may also be due to the fact that draping the fabric on a doubly curved surface causes the fibers to be misaligned.

5. Conclusions

Doubly curved sandwich panels were studied numerically and experimentally. The numerical analyses confirmed that there may be substantial benefits in using curved sandwich panels, but that not all curved panels are superior to flat counterparts. Molds were efficiently made by CNC routing low cost foam. Curved sandwich panels were made by covering the foam molds with a thin film and vacuum infusing the panels directly in these foam molds. Test panels were adhesively bonded to a steel fixture and tested under hydrostatic water pressure. The stiffness predictions from finite element analyses were good, whereas the strength predictions showed some discrepency. The discrepancy is believed to be mainly due to using a very simple wrinkling formula.

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14. ABSTRACT

Doubly curved composite sandwich panels loaded by evenly distributed pressure were designed, analyzed, manufactured and tested. Quick and cost effective methods for making molds for vacuum infused doubly curved composites were studied and implemented. Several different manufacturing techniques for making doubly curved panels and doubly curved foam cores were investigated. Tests were performed using a hydrostatic water tank.

15. SUBJECT TERMS

doubly curved, glass fiber, foam core, composite sandwich panel, vacuum infusion, hydrostatic testing, joints

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