INFLUENCE OF CURVATURE ON FOAM CORE SANDWICH PANELS

M. A. Battley [1], H. Enoksson[2]

[1] *Industrial Research Limited, PO Box 2225, Auckland, New Zealand*
[2] *Department of Aeronautics, Royal Institute of Technology, S 100 44, Stockholm, Sweden*

SUMMARY: This paper describes a study of the effects that curvature has on deformations and stresses within single curvature sandwich panels subjected to transverse pressure loads. Experimentally verified shell and 3D solid finite element models are used to characterise the influence of curvature and boundary conditions on core shear and transverse stresses, face stresses and deflections. Results are compared to relevant marine structural design codes. Curvature generally reduces panel deflections, skin stresses and core shear stresses, but leads to higher transverse core stresses. Hence for design purposes, a flat panel based analysis should provide conservative answers when the skin or core shear allowables are critical. However design codes based on flat panel stress analysis actually produce unconservative values, particularly for skin stresses.

KEYWORDS: Sandwich, Panel, Foam Core, Curved, Finite Element, Strain Measurement, Design, Marine

INTRODUCTION

Sandwich panels are widely used in the hull skins of marine vessels. Such structures provide excellent specific strength and stiffness as well as manufacturing and durability advantages. Despite the fact that such panels are often curved, most of the relevant design codes either treat the panels as flat or make only simple empirical corrections for curvature. Failures of such structures do occur, often by core shear fracture [1], meaning that it is very important to develop a better understanding of how they behave and should be designed.

The primary aim of this study was to determine the effects of curvature on deformations and stresses in single curvature foam core sandwich panels subjected to transverse pressure loads. Predictions from 3-D solid and sandwich shell finite element models were compared to experimental measurements to verify the modelling techniques used. Parametric finite element analysis (FEA) was then used to characterise the influence of curvature and boundary conditions on transverse deflections, skin in-plane stresses, and core transverse shear and transverse normal stresses. Predicted and measured results are compared to design codes from American Bureau of Shipping [2], Det Norske Veritas [3] and Australian Standard 4132.3 [4].

PROBLEM DEFINITION

Fig.1 shows the basic problem geometry which is representative of that used in the bilge region at approximately station three in a 13 metre yacht. The dimensions shown are for the experimental specimen which had a radius of curvature of 650 mm (c=50mm), and an aspect ratio (a/b) of 0.5. Within the parametric FEA study, curvatures were varied from flat to a radius/thickness (r/t) of 10. The shaded region with numbered edges in Fig. 1 is the quarter panel modelled in the finite element analyses. Uniform pressure of up to 66.5 kPa (calculated design head from ABS rule [2]) was applied to the convex side of the panel. Inner and outer skins, Fig. 2, were E-glass fibres (two 1135 g/m2 triaxial plies, one 350 kg/m3 biaxial outer ply) in West Z105/206 epoxy matrix at a fibre volume fraction of 0.50. The laminate was symmetric about the midplane of the core which was 80 kg/m3 PVC HEREX C70.75L-25 (25mm thick).

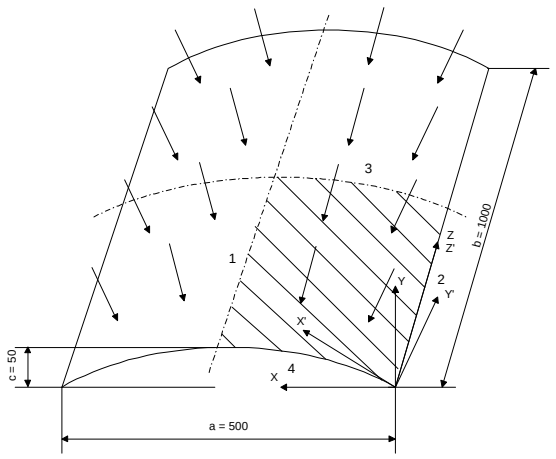

Fig. 1: Problem definition

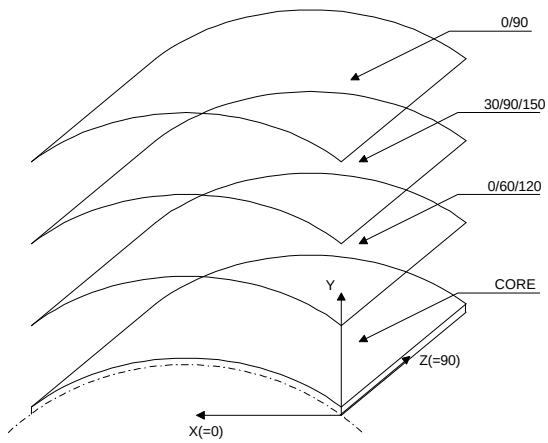

Fig. 2: Lamination sequence

FINITE ELEMENT MODELING

Because of aspect ratio limitations caused by the relative thickness of the skins compared to the core, solid element based meshes of sandwich composites require a significant number of elements and hence degrees of freedom. In practice many real sandwich structures are analysed with shell elements in order to keep the size of the numerical problem to an acceptable level. Most commercial FEA codes offer shell elements that are formulated for the analysis of sandwich structures. These typically assume that that the faces are in plane stress, and the core only contributes to the transverse shear stiffness of the sandwich. Stress results are therefore usually limited to in-plane stresses in the skins and transverse shear stress in the core. Depending on their formulation, such shell elements are often unable to calculate out-of-plane stresses such as transverse tension/compression that occurs in the core of curved panels. For this reason, 3D solid models (EMRC NISA V7.0, NKTP 4). were used for the main parametric study of curvature. Sandwich shell elements (NKTP 33) were used to predict deflections, skin stresses and core transverse shear stresses to determine how their predictions compared to the solid elements and to the experimental results.

A convergence study resulted in the quarter panel meshes shown in Fig. 3 for shell elements, and Fig. 4 for solid elements. The regions of high mesh density correspond to the areas of high stress gradients mid way along each side of the panel. Comparisons between linear and geometrically non-linear analyses confirmed that linear analyses were adequate for the

deformation levels relevant to this study. The experimental results also demonstrated a linear relationship between applied pressure and measured stresses and deformations.

Boundary conditions have very significant effects on the behaviour of this type of panel. In reality a hull panel is part of a continuous structure, supported by transverse and longitudinal frames at the panel edges. The actual degree of fixity provided by the supporting structure and adjoining panels is difficult to define accurately. Design codes for marine panels generally assume either fully fixed [2,3,4] or simply supported [3]. For a curved panel, simply supported conditions can be hard, with no displacements in the plane of the panel; or soft, displacements free in the plane of the panel. This study used either fully fixed or soft inplane simply supported conditions on edges 2 and 4 of the model (edge numbering is defined in Fig. 1) to represent the extreme cases. In the case of the solid model the simply supported boundary conditions were applied to the inner face of the sandwich to represent the experimental case. The experimental test system was also designed to provide close to soft simply supported conditions.

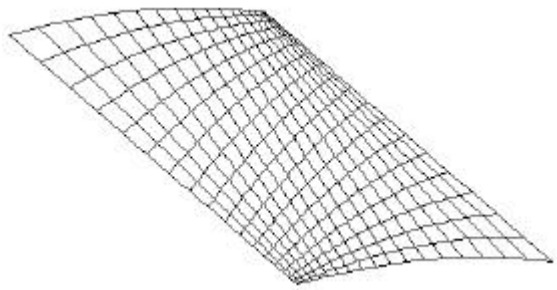

Fig. 3: Mesh for shell models

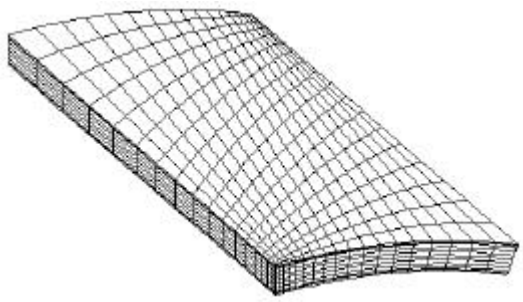

Fig. 4: Mesh for panel models

EXPERIMENTAL

A single curvature sandwich panel was subjected to uniform transverse pressure load of up to 65 kPa. The test system, shown in Fig. 5, consisted of two steel fixtures, the lower of which contained a water filled rubber bladder. The upper frame provided support to the panel edges through 14mm nylon rods and a wooden frame. Pressure was applied to the bladder by a static head of water.

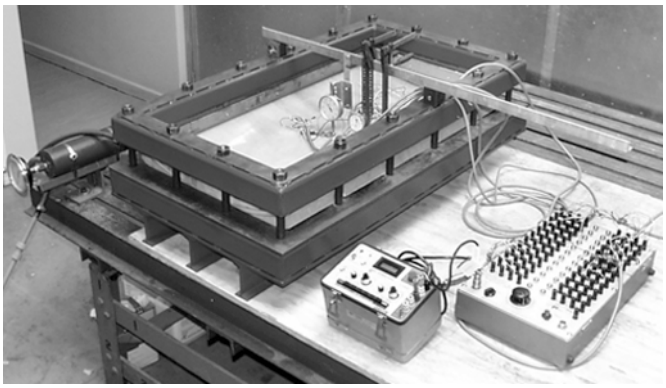

Fig. 5: Panel test rig

Fig. 6: Experimental boundary conditions

Strains were measured on the outer surface of the inner skin and within the foam core at the 7 positions shown in Fig. 7. Displacements were measured at the same points as the strains, and at all corresponding symmetry points. The strain gauges on the skin were uniaxial MicroMeasurements CEA-00-187UW-350 and those embedded within the core were CEA-00-125-UR-350 rosettes.

The embedded gauges were bonded between two blocks of foam which were then machined into a 25 mm diameter cylinder of the same length as the thickness of the foam core. These “strain plugs”, as shown in Fig. 8, were then bonded at the correct orientation into holes that were drilled through the full thickness of the foam core. Two gauges of the rosette were used, enabling measurement of the through thickness strain and the strain at 45 degrees, i.e. one component of the transverse shear strain.

Two rectangular beams were also manufactured with two shear plugs in each. These were loaded in four point bending to obtain a calibration coefficient between the measured strain and the average applied shear stress. These beams were then cut into sections and loaded through their thickness to obtain a calibration coefficient for the transverse normal stress. There were significant differences in the calibration coefficients obtained from each of the strain plugs in the beams ($^{+}/_{-}$10% for shear stress, $^{+}/_{-}$20% for normal stress), meaning that there were also likely to be variations in the calibration of the strain plugs in the test panel. This variability is believed to be due to differences in glue line thickness in the region of the embedded strain gauges affecting the strain field in the region of the gauge. Errors in the orientation of the strain gauge and the strain plug within the panel would also affect the accuracy of the measurements.

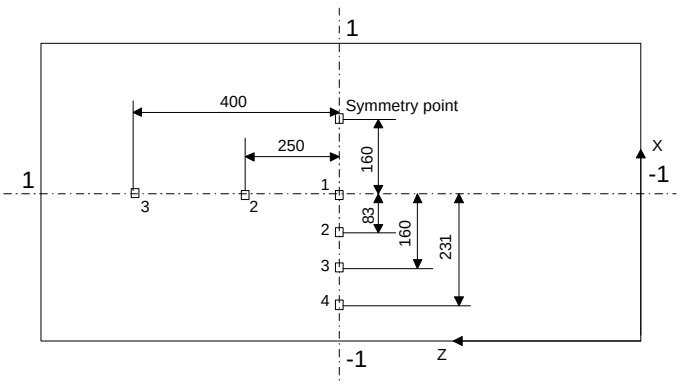

Fig. 7: Strain measurement points

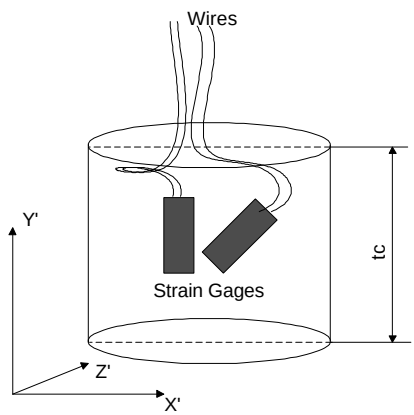

Fig. 8: Embedded strain gauges

The test specimen was constructed in stages on a convex curved wooden mould using wet lay-up and vacuum bagging techniques representative of those used within the marine industry. Prior to skin lamination, the foam was heated up to 120 °C for 15 min and shaped into the correct curvature on the mould. The inner skin was then laminated onto the mould and cured at approximately 50 °C. The pre-curved foam was impregnated with epoxy and micro balloons then bonded under vacuum to the skin. Holes were drilled in the core and the shear plugs bonded into the foam. The foam face was then preimpregnated with epoxy and microballoons and the outer face laminated onto the panel and cured.

RESULTS

Stresses and Deflections

Stresses and deflections from the solid and shell element FEA models are compared to the experimental results in Figs. 10 to 13. The abscissa of each graph is the position along the curved centreline of the panel (edge 3 in Fig. 1). The position is normalised with respect to half of the width of the panel (distance a in Fig. 1), meaning that 0 represents the centre of the panel and 1 the edge. Experimental data points are averages of 7 repeated tests. The two coordinate systems used for the stress and deflection results are shown in Fig. 9.

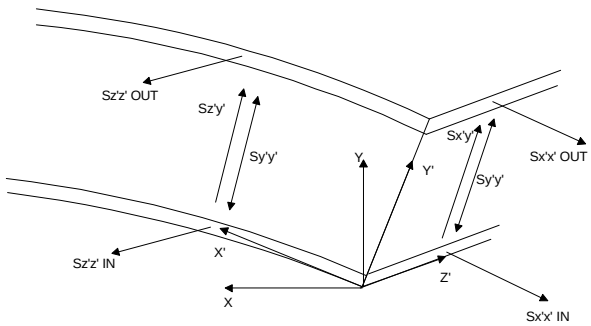

Fig. 9: Coordinate systems and stress components

The XYZ system is used for the displacements and is a Cartesian system with the X axis across the curved direction of the panel and the Y axis out of plane.

The X'Y'Z' system is a corresponding cylindrical system for the stresses in the plane of the panel.

Figs. 10 to 13 show that there is generally good correlation between the shell and solid element FEA predictions, and between the simply supported FEA models and the experimental measurements. However in both the simply supported and clamped cases the solid element models predict edge effects for Sx'x' (Fig. 11) and Sx'y' (Fig. 12) that do not appear in the shell element results. It is likely that these edge effects are due to the idealised boundary conditions of the solid elements. These edge effects do not appear in the experimental Sx'x' data.

The difference in predicted deflections between the shell and solid element models for the simply supported case, Fig. 10, is partly due to transverse compression of the core in the solid element model. It is also possible that the differences are due to difficulties that the NISA NKTP 33 sandwich shell element has when calculating the transverse shear stiffness of the sandwich. Test cases using beam and panel models demonstrated that these elements appear to incorrectly calculate the transverse shear stiffness of the sandwich resulting in virtually no transverse shear deformation or core shear stress unless the transverse shear stiffness of the skins is set to a very small value. The results presented here are for a skin transverse shear stiffness of virtually zero (1E-4 MPa), so correlation could be improved by increasing this value. The experimental deflections were measured relative to the upper steel fixture and hence include any deformation of the wooden frame and nylon rods at the edge of the panels. Measurements indicated that this was of the order of 0.5 mm which would significantly improve correlation with the solid model's predictions.

The experimental measurements of transverse shear stresses shown in Fig. 12 demonstrate similar trends as the predictions, but are of greater magnitude. This difference may be due to the difficulties encountered in obtaining accurate calibration coefficients for the strain plugs. Although the transverse normal stress is of very small magnitude, as shown in Fig. 13, there is reasonable correlation between the experimental measurements and the solid simply supported predictions. As in the Sx'x' and Sx'y' results, the solid models predict edge effects that do not

appear in the experimental data. No shell results are presented because the shell models do not calculate this stress component.

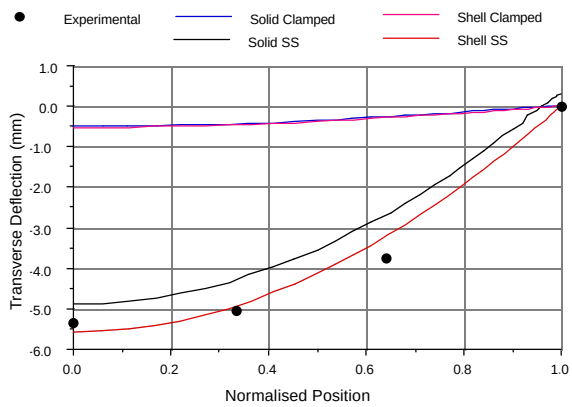

Fig. 10: Transverse deflection (Uy)

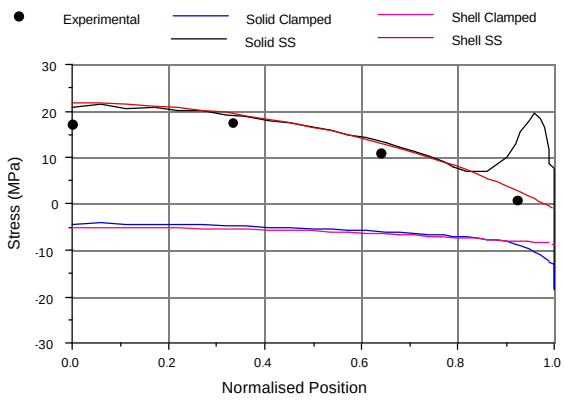

Fig. 11: Skin stress (Sx'x')

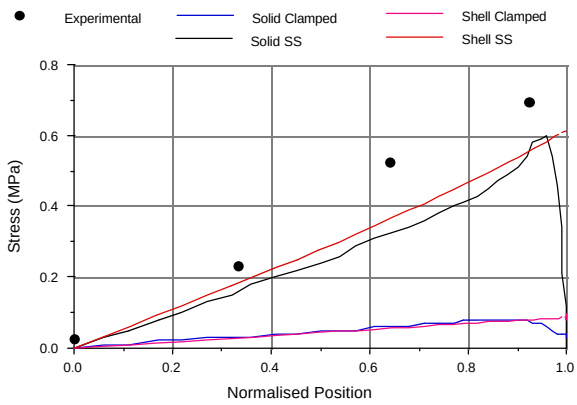

Fig. 12: Core transverse shear stress (Sx'y')

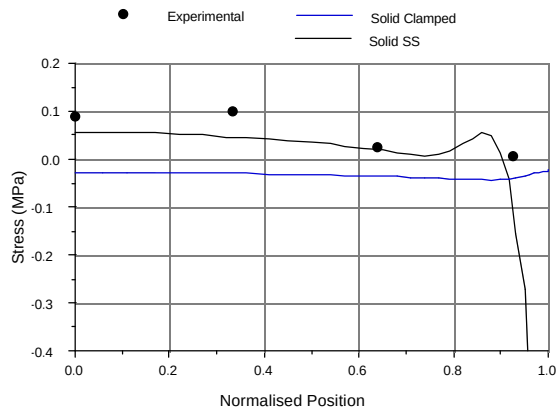

Fig. 13. Core transverse normal stress (Sy'y')

Effect of Curvature

Figs. 14 to 17 show the maximum deflection and stresses for panels with different curvatures as calculated by the parametric solid element FEA models. The stresses were characterised along both symmetry axes of the panel (edge 1 and 3 as shown in Fig. 1), however the maximum stresses all occurred along side 3 of the panel. Edge effects due to the idealised boundary conditions were excluded from the results. When the maximum stress appeared closest to the edge, such as for the core transverse shear stress, the value at position 0.8 was used (0 is the centre and 1 is the edge). The skin stresses were extracted on the outer and inner face and core stresses were taken at the midpoint of the core. The stresses and deflections were normalised by dividing by the corresponding value for a flat panel. Normalised curvature is defined as the highest point of the curved panel arc divided by the width of the panel (c/a in Fig. 1).

The results shown in Fig. 14 clearly demonstrate the significant reduction in deflection caused by increasing curvature. For a clamped panel the maximum deflection of a flat plate is reduced by 80% for a normalised curvature of only 0.10. Fig. 15 shows that for the simply supported panel the inner face stress decreased gradually by approximately 60% with increasing curvature. For the clamped panel a small curvature (0.05) doubled the inner skin stresses

compared to a flat panel, but the stresses decreased quite rapidly as the curvature increased further. The effects of the curvature on the outer face stress were similar but not as significant. Maximum values appeared in the centre of the panel except for the clamped inner face where the maximum value was close to the edge. Fig. 11 shows that the high stress is a consequence of the idealised clamped boundary condition. To avoid these edge effects the stress was measured at a normalised distance of 0.8 from the centre.

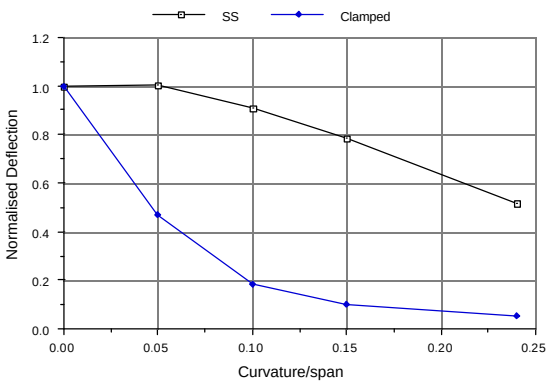

Fig. 14. Effect of curvature on transverse deflection (Uy)

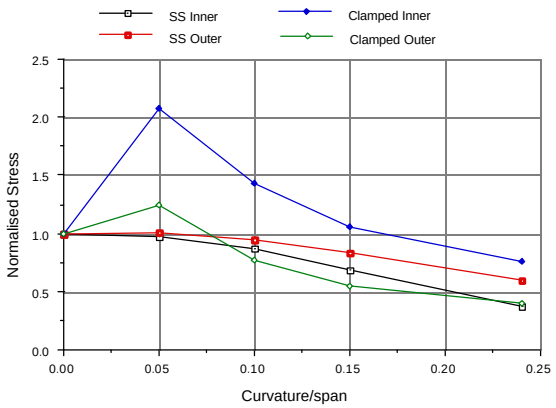

Fig. 15. Effect of curvature on skin stress (Sx'x')

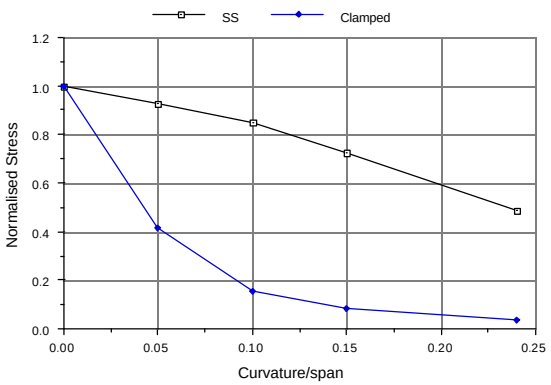

Fig. 16: Effect of curvature on core transverse shear stress (Sx'y')

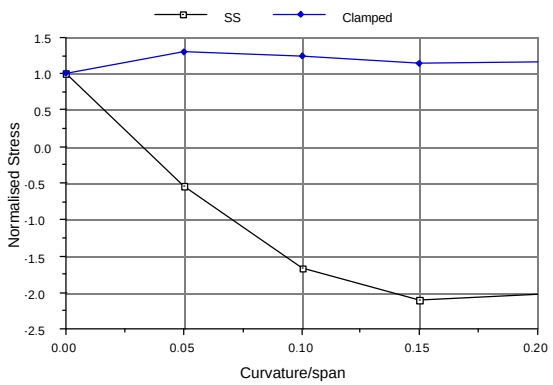

Fig. 17: Effect of curvature on core transverse normal stress (Sy'y')

As shown in Fig. 16, increasing curvature dramatically reduced the maximum core transverse shear stress, particularly for clamped boundary conditions. The maximum transverse shear stress in the core of a clamped panel with normalised curvature of 0.10 is only 17% of that in a flat panel. In the corresponding simply supported panel, as used in the experimental test, the shear stress would be about 20 % lower than in a flat panel.

Fig. 17 shows that in the simply supported panel the core transverse normal stress changed sign as the effects of the deformation on the stress exceeded the direct compression from the applied pressure. The transverse normal stress also increased to a maximum of approximately twice the stress in a flat panel at a curvature/span of 0.15. For the clamped case the stress was only slightly greater for a curved panel, presumably because the total deflection was smaller. The maximum core transverse normal stress appeared in the centre of the simply supported panel. However for the clamped case it appeared close to the edge and to avoid end effects the stress was measured at a normalised distance 0.8 from the centre.

Comparisons to Design Rules

Table 1 compares the measured maximum values of skin membrane stress, core transverse shear stress and transverse deflection to predictions for the same panel size and materials from solid element based FEA models and three commonly used marine design codes; American Bureau of Shipping, Guide for Building and Classing Offshore Yachts – 1994 (ABS, [2]), Det Norske Veritas, High Speed and Light Craft (DNV, [3]), and Australian Standard 4132.3 – 1993, Boat and Ship Design and Construction (AS, [4]). The safety and pressure reduction factors for different locations and applications have been removed from the design code calculations to allow comparisons to be made of the panel analysis methods only. It should be noted that the DNV rule used is intended for power craft, the ABS rule for yachts and the AS rule is applied to both yachts and motor-yachts. This means that for certain panel sizes and positions the rules apply different safety factors and head reduction factors. Another source of differences is that the ABS rule does not consider shear in the deflection calculations while AS and DNV do. This is significant for the case studied, as the panel is of relatively small planar dimensions relative to its thickness. Only the AS and ABS codes include curvature effects and only the DNV code considers simply supported (SS) boundary conditions.

Table 1. Comparison between design rules, FEA and test panel results.

Method	Boundary Condition	Skin Membrane Stress (MPa)		Core Transverse Shear Stress (MPa)		Transverse Deflection (mm)	
		Curved	Flat	Curved	Flat	Curved	Flat
DNV	Clamped	--	8.0	--	0.54	--	3.1
AS	Clamped	6.9	8.5	0.59	0.59	2.1	2.8
ABS	Clamped	6.7	8.4	0.58	0.58	0.5	0.7
FEA	Clamped	8.1	9.6	0.08	0.49	0.5	2.7
DNV	SS	--	9.8	--	0.54	--	5.1
FEA	SS	21.0	24.5	0.60	0.67	4.9	5.4
Test	SS	18.0	--	0.70	--	5.4	--

The different design rules all gave similar maximum skin stress results for clamped edges, both with and without curvature. The FEA model predicted higher stresses than any of the design codes. For the simply supported panel both the FEA and measured maximum stresses were approximately twice the stresses calculated with the DNV design rule.

For the maximum core transverse shear stresses the different design rules again gave similar results, however none recognised the very significant reduction (about 83%) in core shear predicted by the FEA for a curved clamped panel. The DNV code did not predict any difference between the clamped and simply supported panel, while the difference was approximately 35% in the FEA models. Overall the DNV rule predicted smaller differences in stresses between clamped and simply supported cases and curved and flat panels than predicted by the FEA model. This was particularly true for the core transverse shear stresses.

The maximum transverse deflection predicted by the FEA and ABS rule correlated very well with curvature, but not without. Conversely, the AS rule worked well for flat panels, but not

for curved. Both the AS and ABS design rules predicted a much smaller difference in deflection between a curved and flat panel than the FEA. In the DNV rule the difference between the clamped and simply supported case was similar to the corresponding difference in the FEA.

The maximum stresses calculated with the FEA and measured experimentally were almost always higher than those predicted by the design rules. However in practice the codes' safety factors should make their predictions more conservative than is apparent from these results.

CONCLUSIONS

Deformations and skin and core stresses can be accurately predicted for sandwich composite panels. However to achieve reliable predictions it is necessary to fully understand the behaviour of the analysis methods used, and to model the panel boundary conditions as realistically as possible.

The curvature of a sandwich panel has a major impact on its stresses and deflections, especially for clamped boundary conditions. Curvature generally reduces panel deflections, skin stresses and core transverse shear stresses, but leads to higher core transverse normal stresses. Hence for design purposes, a flat panel based analysis should provide conservative answers when the skin or core shear allowables are critical.

Relevant marine design codes do not appear to accurately recognise the effects of curvature for sandwich panels, and without their "safety factors" yield generally unconservative predictions of stresses and deflections.

ACKNOWLEDGEMENTS

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REFERENCES

1. Hayman, B., "Response of Sandwich Structures to Slamming and Impact Loads", *Composite Materials in Maritime Structures, Volume 2: Practical Considerations*, Shenoi, R.A. and Wellicome, J.F., Ed., Cambridge University Press, 1993, pp. 161-177.

2. American Bureau of Shipping, "*Guide for Building and Classing Offshore Racing Yachts*", 1994.

3. Det Norske Veritas Classification AS, "*Tentative Rules for Classification of High Speed and Light Craft*", 1995.

4. Australian Standard 4132.3 – 1993. "*Boat and Ship Design and Construction*", 1993.