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ANALYTICAL AND NUMERICAL ANALYSIS OF NON – SYMMETRICAL ALL STEEL SANDWICH PANELS UNDER UNIFORM PRESSURE LOAD

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1. Introduction

The demand for bigger, faster and lighter moving vehicles, such as ships and trains has increased the demand for efficient structural arrangements. Sandwich panels form one type of efficient structures enabling the use of steel, aluminium or composites for the constructions. (Kujala et al., 1995). Development of laser welding techniques enables the fast production of high quality and dimensionally accurate all steel sandwich panels. Helsinki University of Technology/Ship Laboratory has developed during the last 7 years a number of various applications of laser welded steel sandwich panels. The work carried out includes e.g. development of design and production methods with extensive strength testing for various types of sandwich panels (as summarised by Romanoff and Kujala, 2002). Steel sandwich panels welded by laser can offer 30-50 % weight savings compared to the conventional steel structures.

In this paper analytical analysis of all steel sandwich panels, such as shown on Figure 1a, have been performed in order to determine the structural behaviour under uniform lateral pressure load, especially the shear stiffness transverse to the core stiffeners are under consideration. Sandwich construction has been analysed as a simply supported beam, Figure 1b. Specific formulations have been developed and closed form solution established.

When analysing all steel sandwich constructions under bending loads one has to take into account the influence of shear stiffness *S* on a construction and not only the bending stiffness *D*. We will define here the accurate shear stiffness relation. Another thing to consider when analysing sandwich beams is *thick faces effect* that we will explain in the text.

Numerical finite element models have been developed to verify the results of here presented analytical relations. Models have been created using FEM application *Abaqus*. Beam numerical models have been defined with varying face plate thicknesses.

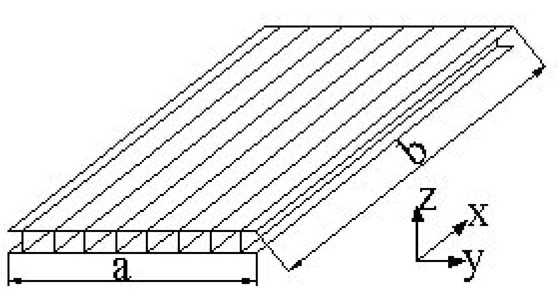

Figure 1a. I core all steel sandwich panel

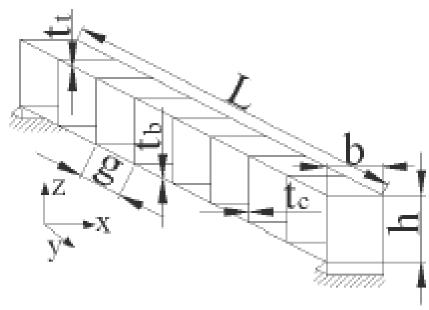

Figure 1b. Sandwich panel modeled as a beam

2. Background

Laser welded steel sandwich panels applications and design methods have been under active research in USA and Europe during the last decade. The US Navy has studied the applications of laser welded corrugated core sandwich since 1987 (Marsico et al., 1993).). These applications include bulkheads and decks in accommodation areas, deckhouses, deck edge elevator doors, hangar bay division doors and applications on SWATH structures. In Germany Meyer Werft has both designed and produced steel sandwich panels since 1994 (Roland & Metschkow, 1997). They have both produced and designed steel sandwich panels for example for ship applications such as staircase landings, decks, bulkheads, etc. They have also made several strength and fatigue tests together with both analytical and finite element analyses for steel panels. MacGregor has developed a new kind of steel sandwich panel where the core is made out of steel wires. They also sell this product as a Ro-Ro decks etc. (Macgregor, 2000). In Finland the research related to all steel sandwich panels was initiated in 1988 in the Ship Laboratory of Helsinki University of Technology. The research was initiated by studying the application of sandwich panels in the shell structures of an icebreaker. Since then there have been several research projects in Finland regarding the manufacturing, design and design optimisation of steel sandwich panels. The topics, which have been investigated and related to design or design optimisation of steel sandwich panels (Romanoff & Kujala, 2002), see Figure 2 for typical steel sandwich cross sections studied. At the moment there is also a European union funded project called Advanced Composite Sandwich Steel Structures (Sandwich, 2001) where an extensive test program, together with the development of design formulations and prototype designs are carried out.

The formulations presented in basic textbooks for sandwich structures (e.g. Allen 1969, Zenkert, 1995) concentrate mainly on the calculation of the elastic response of sandwich panels in general. For steel sandwich panels these books include little information about the special features of the steel sandwich panels. The steel sandwich panels have strongly orthotropic behaviour due to the having the core stiffeners only in one direction. This orthotropic behaviour and especially the shear stiffness transverse to the core stiffeners have crucial importance for development of the design methods for these panels. This topic has been studied earlier by e.g. Fung et al. (1994) and Lok et al. (2000) for symmetric steel sandwich panels. Here the analysis is extended to cover also the possibility of having non-symmetric cross section i.e. the top and bottom plate have different thicknesses. In addition, the effect of the thick face plating is included in the analysis as follows.

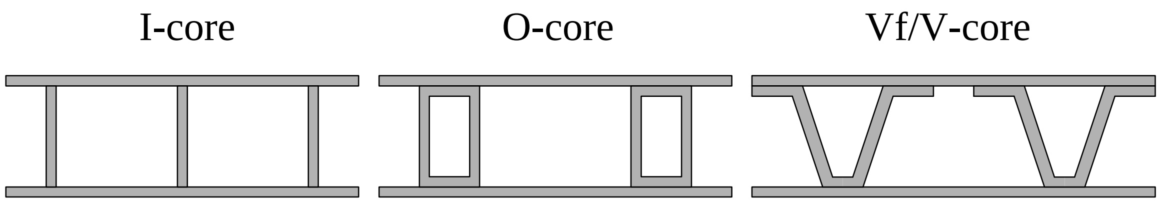

Figure 2. Typical steel sandwich panel cross sections

3. Defining shear stiffness S_y

All steel sandwich panels which have a core stiffeners in only one direction and are therefore strongly orthotropic in shear, initiate a question of determination of shear stiffness in the direction perpendicular to the core. We assume that the face plates are carrying the shear forces in y direction, alongside with the bending moments. Shear stiffness is defined by (1), from which emerges the problem of finding the shear strain g_{yz}. Shear force T_y is divided on the part carried by the top plate and on the part carried by the bottom plate. We also consider the influence of shear force when it is applied at a distance. This causes horizontal force $T_y\ p/h$ [Fung et al. 1994], where $p = g/2$.

$$S_y = \frac{T_y}{g_{yz}}, \tag{1}$$

Now, we can create the analytical model. Since it is symmetrical about stiffener we consider just half of the segment. Analytical model on Figure 3 is a frame, simply supported in points B and C to prevent rigid body motion. We predict small displacement in horizontal direction since the face plates are much more rigid in compression/tension then in bending for approximately same amplitude of force. Due to symmetry of the segment and position of the vertical forces we also predict small vertical displacements of the points B and C.

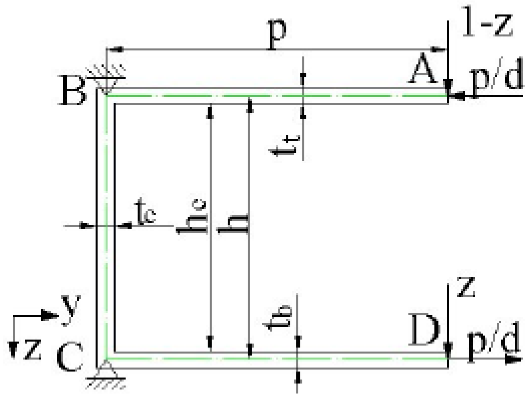

Figure 3. Half segment

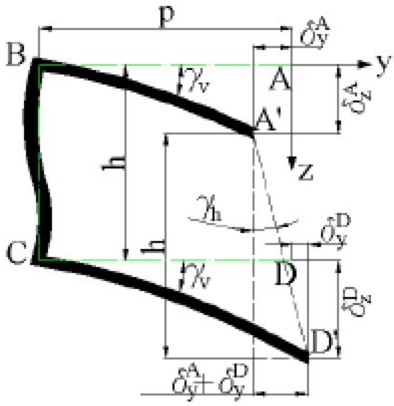

Figure 4. Deformed half segment

Let shear force T_y be of magnitude 1. In Figure 3 the shear force has been divided into two parts, $(1-z)$ and z to be carried by the top and bottom plate. The value of z is a function of face plate thicknesses and it will be defined later. To calculate the total deformation of the segment under internal forces the displacements of points A and D must be defined. Since the breadth of the frame model part b in direction x is larger than the length p, see Figure 1, frame part is in plane strain so Young's modulus takes the value $E/1-n^2$ [Fung et al. 1994]. When calculating the beam which breadth of the part is smaller than one quarter of its length we use the one dimensional Young's modulus E.

Let us state two conditions that have to be followed when defining the shear strain:

a) during the deformation angles at points B and C stay constant, with the value of 90°,
b) vertical displacements of points A, d_z^A, and D, d_z^D, are equal, that way keeping the thickness of the panel constant.

Shear strain g_{yz} is separated into vertical and horizontal part, g_v and g_h each defined through values of displacement of point A and D [Lok 2000]. From Figure 4 we can write:

$$g_v = \frac{d_z^A}{p} \text{ and } g_h = \frac{d_y^A + d_y^D}{h}. \tag{2}$$

Vertical and horizontal displacements of points A and D are defined for unit shear force load and substituted into equations (2):

$$g_v = \frac{4\cdot(1-n^2)}{E}\left[\frac{(1-z)p^2}{t_t^3} + \frac{p\cdot h(2-3z)}{t_c^3}\right] \text{ and } g_h = \frac{p^2(1-n^2)}{h^2E}\left(\frac{1}{t_t}+\frac{1}{t_b}\right) \tag{3}$$

The value of factor z can be found from condition (b):

$$z = \frac{\left(pt_c^3 + 3ht_t^3\right)\cdot t_b^3}{6ht_t^3t_b^3 + pt_c^3\left(t_t^3 + t_b^3\right)} \tag{4}$$

Shear strain for the plate S_y can now be determined with equation (5) and for beam S_b with breadth of b with equation (6):

$$S_y = \frac{1}{g_v + g_h} \qquad (5) \qquad\qquad S_b = S_y \cdot b \qquad (6)$$

4. Closed form solution for beam bending

Let us consider now half of a beam in Figrure 1b under the uniform line load, see Figure 5. Its shear stiffness S_b is small compared to its bending stiffness D_b so its structural response will be largely influenced by shear stiffness. This is the reason why we use beam analysis to determine the accuracy of the shear stiffness model. Beam is analysed with the effect of thick faces to obtain more accurate results in structural response.

Effect of thick face plates of the sandwich panel assumes that the shear deformations cause not only the local core deformations but also the local deformations of the face plates [Allen 1969]. These deformations can be defined as bending of the faces about their own neutral axes. When the sandwich bends as a whole the faces deform locally and globally. In addition local bending stiffness described by this effect influences shear deformations of the core. Sandwich beams with thinner face plates are less influenced while the ones with thicker are more. Their face plates have significant local bending stiffness D_f and no longer behave as flexible membranes. Bending moments are:

$$D_b = E\left[\frac{b}{12}\left(t_t^3 + t_b^3\right) + A_{tp} \cdot e_{tp}^2 + A_{bp} \cdot e_{bp}^2\right], \; D_f = \frac{b}{12}\left(t_t^3 + t_b^3\right), \qquad (7)$$

where A's are cross section areas of face plates and e's are distance between the face plates and a neutral line. Full derivation of the closed form solution is omitted here [Allen 1969], just final expression is given in (8).

$$w = \frac{qx^2L^2}{48D}\left(3 - \frac{2x^2}{L^2}\right) + \frac{q}{S_b}\left(1 - \frac{D_f}{D}\right)^2\left[\frac{x^2}{2} + \frac{bL^2}{4q}\left(1 - \cosh ax\right)\right], \qquad (8)$$

$$\text{where: } a^2 = \frac{S_b}{D_f\left(1 - \frac{D_f}{D}\right)}, \; q = \frac{a \cdot L}{2}, \; b = \frac{1}{q \cosh q}. \qquad (9)$$

The closed form solution with and without the thick plate effect (for the case 1 in Table 2) is shown in Figure 6 together with the FEM results as described in further detail in the next section. As can be seen, including the thick face effect decreases remarkably the deflections of the beam and gives results close to the FEM calculations.

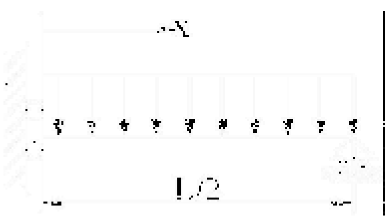

Figure 5. Half of a sandwich beam

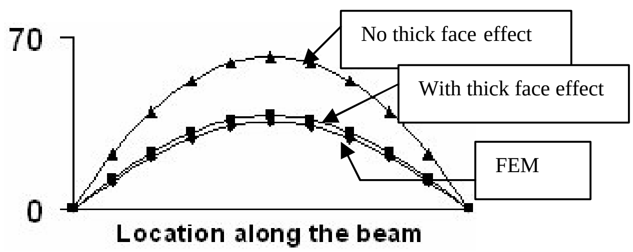

Figure 6. Beam displacement in mm

5. Numerical models and comparison of results

The numerical models are created to enable the determination of the accuracy of the analytical model of shear stiffness and also to verify the relation (8). Using the symmetry of boundary conditions of a beam only one half has been modelled on which meshing of 832 eight – nodal quadratic shell elements has been applied, see Figure 7.

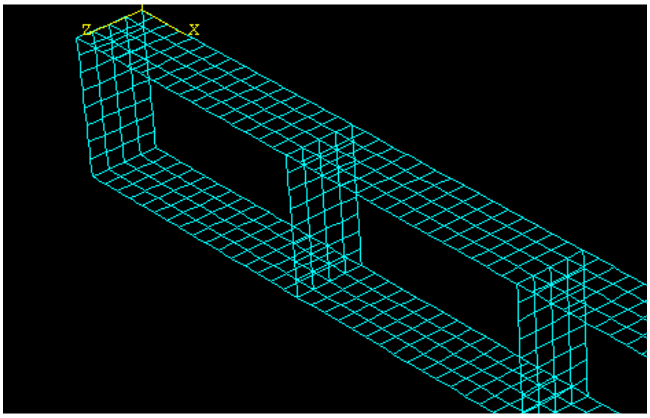

Figure 7. Element meshing applied to describe the beam

We calculate the maximal displacement for the beam with the characteristics given in Table. Uniform pressure load q = 1000 Pa has been applied over top plate surface. For the analytical beam calculations it has been transformed into a line load of q_b = 50 N/m. We present here 5 typical cases to compare the analytical and numerical analysis.

Table 1. Characteristics of beam numerical models

Material prop.	Beam prop.	Thicknesses
E = 206.8 Gpa	l = 2000 mm	t_t = 1÷5
n = 0.29	b = 50 mm	t_c = 2÷4
	h = 100 mm	t_b = 2÷3
	p = 100 mm	
	g = 200 mm	

Table 2 shows the lists of maximal deflections acquired by the analytical and FEM calculations and the shear stiffness calculated with the analytical formula for the analysed models. Also the margins of error of analytical model to the FEM are presented. As can be seen the maximum error is only 2.32 %.

Table 2. Maximal displacements of the calculated beams

Thick[mm]			Stiff.[Nm]	Max. deflection [mm]		Error
t_t	t_c	t_b	S	Analytic	FEM	[%]
1	2	2	1175,43	21,139	20,740	1,89
3	2	2	2842,78	8,692	8,490	2,32
3	4	2	6969,71	3,642	3,724	-2,24
3	3	3	7736,31	3,262	3,287	-0,77
5	3	3	10000,65	2,489	2,438	2,05

6. Conclusions

Analytical calculations shown in the paper result in small margin of error, so we can be sure that they have been developed correctly to satisfy the need of sandwich panel design. If we know that the bending stiffnesses D_b and D_f are, in manner we have described it, considered as an accurate values,

we can conclude that the analytical model for the determination of shear stiffness is accurate since the beam deflection is dependent only on these three variables. Shear stiffness S_b was derived for the beam and the plate with the same assumptions and in the same general manner, so then in future analysis of sandwich plates relations presented here could be considered correct.
We can also confirm that the closed form solution for determination of deflections is accurate and that the assumptions made when analysing thick faces effect are correct.
In future these formulations will be applied to optimise the sandwich panel configurations for various applications onboard ships such as movable car decks.

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