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Local buckling behaviour of steel plate elements supported by a plastic foam material

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Abstract. Sandwich panels comprising steel facings and a polystyrene foam core are increasingly used as roof and wall claddings in buildings in Australia. When they are subjected to loads causing bending and/or axial compression, the steel plate elements of their profiled facing are susceptible to local buckling. However, when compared to panels with no foam core, they demonstrate significantly improved local buckling behaviour because they are supported by foam. In order to quantify such improvements and to validate the use of available design buckling stress formulae, an investigation using finite element analyses and laboratory experiments was carried out on steel plates that are commonly used in Australia of varying yield stress and thickness supported by a polystyrene foam core. This paper presents the details of this investigation, the buckling results and their comparison with available design buckling formulae.

Key words: local buckling; steel plates; plastic foam core; sandwich panels.

1. Introduction

The sandwich panels investigated in this study comprise a thick, light-weight plastic foam core sandwiched between two relatively thin and strong facings which are commonly steel (see Fig. 1). They have a number of advantages including high strength to weight ratio, excellent thermal and sound insulation, long spanning capability, and ease of fabrication. All these make the sandwich panel a potential load-carrying component in buildings. The structural performance of the sandwich panel depends on the composite action of the two steel faces and the plastic foam core. One or both steel faces may be flat, lightly profiled or fully profiled as shown in Fig. 1. The core material is usually a rigid plastic foam material such as polyurethane, polyisocyanurate, polystyrene or phenolic resin.

From the early 1940's, sandwich panels have been used extensively in the field of aeronautics. Since the 1970's, these panels have commonly been used as roof and wall claddings not only in cold-storage buildings, but also in all other categories of buildings in Europe and the USA (Allen 1969). This is mainly due to the excellent thermal insulation properties of these panels. Sandwich panel construction in the USA and Europe often uses polyurethane foam and mild steel, which is, bonded together using the inherent bonding capability of injected foam. In contrast, in Australia, polystyrene foam and thinner (0.42 mm) and high strength steels (minimum yield stress of 550

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MPa with reduced ductility) are used which are bonded together using separate adhesives. Further, these panels are not commonly used except in cold-storage buildings due to the lack of design methods and data. Although extensive research has been carried out in Europe and the USA, and adequate design rules are available for mild steel and polyurethane foam (ECCS 1991, Davies 1987), their applicability to Australian sandwich panels requires validation because of the differences mentioned above. Therefore this research is aimed at validating some of the important aspects of sandwich panel behaviour and design rules so that this efficient building product can be used in Australia.

When the sandwich panels are used as roof or wall claddings subject to axial compression loads and/or bending, the thin plate elements of their profiled steel faces have to carry compression forces, and are therefore susceptible to elastic local buckling. The critical plate elements are often modelled as a thin plate supported by both the plastic foam core and the adjacent webs. This paper presents the details and results of experimental and finite element analyses of the local buckling behaviour of these critical steel plate elements and compares the results with the available design rules. Details in relation to the ultimate strength behaviour are given in Jeevaharan and Mahendran (1996).

2. Buckling stress of steel plate elements supported by a foam core

In cold-formed steel construction, thinner steels of higher yield stress are increasingly used. This means the width to thickness (b/t) ratios of the plate elements of these structures are usually large, for example, in sandwich panel construction, b/t ratios can be as large as 600. Therefore, elastic local buckling is one of the major design criteria of these thin-walled high strength steel plate elements. Local buckling involves out-of-plane flexural displacements of the plate elements, which appear in a wave format at the point of buckling which then increases in magnitude with increasing load. This mode of buckling does not constitute member failure as the plates often continue to carry increasing loads. The elastic critical buckling stress σ_{cr} of plate elements subjected to axial compression is given by the following formula:

$$\sigma_{cr} = K\pi^2 E/[12(1-\nu^2)(b/t)^2] \qquad (1)$$

where, b/t =width to thickness ratio of plate element

E =Modulus of elasticity of steel

ν =Poisson's ratio=0.3 for steel

K =Buckling coefficient (dependent upon support conditions)

As noted in the above equation, the buckling stress is independent of the yield stress of the material, and thus high strength steel will lead to a higher ultimate strength through the presence of a significant post-buckling strength. The buckling coefficient K is about 4 for plate elements, which are simply supported along their longitudinal edges. For example, the commonly used Australian sandwich panel with a profiled face has no foam within its ridge (see Fig. 1), and thus the K value for these configurations is 4.

When the plate elements are supported by foam, their critical buckling stress will be significantly increased (higher K values). In sandwich panels, plane and lightly profiled faces subjected to axial compression behave like a wide compressed flat beam on an elastic foundation, and undergo a wrinkling failure (Davies *et al.* 1991) whereas plate elements in the profiled faces undergo elastic local buckling followed by significant post-buckling. This paper addresses only

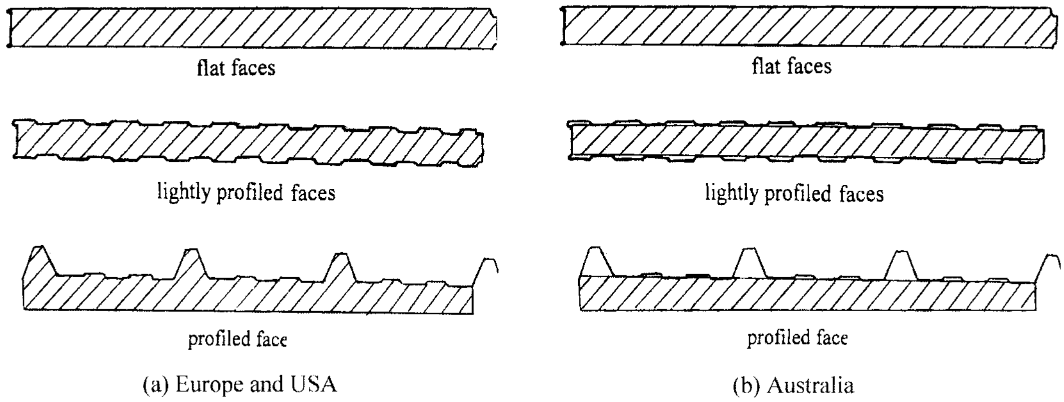


Fig. 1 Sandwich panels

the latter topic. It is to be noted that the buckling behaviour of lightly profiled faces depends on the interaction of local buckling of the plane parts and global buckling of the complete face (Davies *et al.* 1991), which is not investigated here.

For the purposes of design, the increase in local buckling stress (i.e., increase in K value) due to the presence of the foam core should be quantified. This has been well researched in the past (Davies *et al.* 1991). A linear buckling analysis using an energy method was used for this purpose. A plate element with simply supported longitudinal edges was analysed with the supporting foam core foundation modelled as an elastic half-space (infinite depth). In this method, the strain energy of the foam core is more problematical, for which purpose an exact and a simplified method were used. In the exact method, this supporting core foundation was modelled as elastic half-space within which the displacement functions in three directions (x, y and z) have been assumed for a homogeneous and isotropic foam material. In the simplified method, only the most important stress components were included and thus a single displacement function was used for the core deformation. Exact and simplified expressions for strain energy were included in the total energy equation and a buckling stress equation in the form of Eq. (1) was obtained, where K is given by the following.

$$K = \left[\frac{1}{\phi} + n^2\phi\right]^2 + R\,\phi[1 + n^2\phi^2]^{1/2} \tag{2a}$$

where, ϕ=Half wave buckling length a/plate width b

n=Number of buckles across the width (n=1 for primary buckling mode)

$$\text{Exact } R = \frac{24(1-\nu^2)(1-\nu_c)E_c}{\pi^3(1+\nu_c)(3-4\nu_c)E}\left(\frac{b}{t}\right)^3 \tag{2b}$$

$$\text{Simplified } R = \frac{12(1-\nu^2)}{\pi^3}\,\frac{\sqrt{E_c G_c}}{E}\left(\frac{b}{t}\right)^3 \tag{2c}$$

where, E_c=Modulus of elasticity of the foam core material

G_c=Shear modulus of the foam core material

ν_c=Poisson's ratio of the foam core material

As seen above, the influence of composite action is modelled by the dimensionless stiffness parameter R. In both cases, the critical buckling stress was found by minimising the buckling coefficient K with respect to ϕ, which gives,

$$2n^4\phi - \frac{2}{\phi^3} + R(2n^2\phi^2+1)(n^2\phi^2+1)^{-1/2} = 0 \tag{3}$$

This equation was solved for ϕ using a numerical method such as the Newton Iteration method, from which the buckling coefficient K (using Eq. (2a)) and stress σ_{cr} (using Eq. (1)) were found. This complex procedure was used to obtain the buckling coefficient K for varying R values, based on which an approximate solution given by Eq. (4) was developed for K that can be used for practical design purposes (Davies *et al.* 1991). Although this equation was derived independently of the type of R (i.e., exact R given by Eq. (2b) or simplified R given by Eq. (2c)), Davies *et al.* (1991) recommended the use of simplified R with this equation.

$$K = [16 + 11.8R + 0.055R^2]^{1/2} \tag{4}$$

Davies *et al.* (1991) investigated only the range of R from 0 to 200, for which they found the primary buckling mode with n=1 to be always critical. Beyond this range it may be necessary to consider other buckling modes (n>1). In a similar manner, two other equations for K were also proposed which are shown next.

Based on a simplified foundation model

$$K = 4 - 0.474R + 0.985R^2 \tag{5}$$

where:

$$R = \frac{b}{t}\left[\frac{E_c G_c}{E^2}\right]^{1/6}$$

Based on the half-space assumption,

$$K = 4 - 0.415R + 0.703R^2 \tag{6}$$

where:

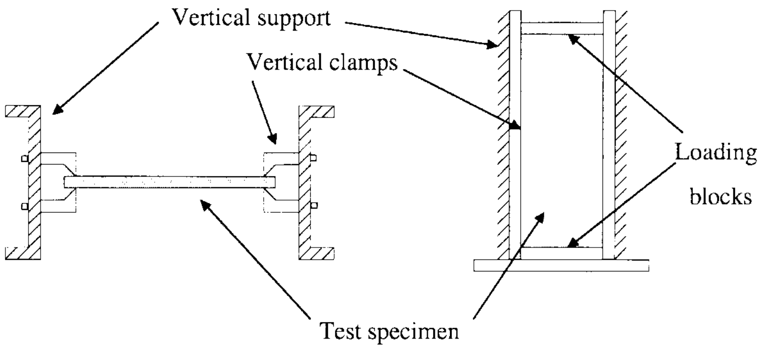
$$R = \frac{b}{t}\left[\frac{E_c}{E}\right]^{1/3}$$

Eq. (5) assumes that the Poisson's ratio for steel ν to be 0.3 whereas Eq. (6) assumes ν to be 0.3 and also Poisson's ratio of the foam core ν_c to be 0.25. The latter assumption of ν_c=0.25 therefore eliminates the use of Eq. (6) for all foam cores as the value ν_c varies significantly between foam materials. In this investigation the ν_c value of polystyrene foam was found to be only 0.08 and hence Eq. (6) is not applicable.

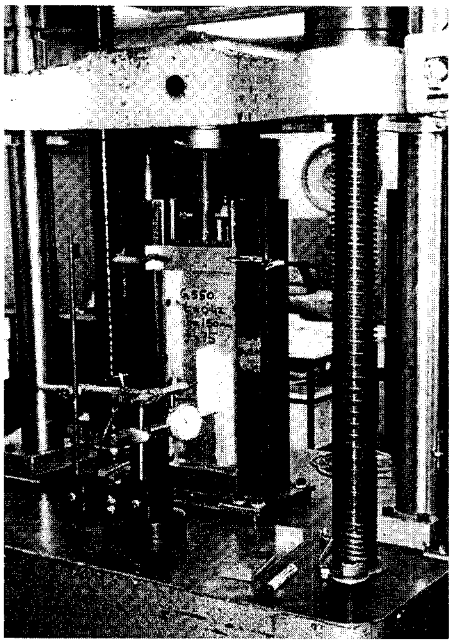
Davies *et al.* (1991) compared the above three alternatives (Eqs. (4) to (6)) with an exact solution and concluded that for practical purposes they are interchangeable. However, Davies and Hakmi (1991) recommended Eq. (4) for buckling stress calculations, and this has been adopted by European design codes (ECCS 1991). Their research and testing was mainly concerned with panels made of polyurethane foam and softer low strength steel which are bonded together using the inherent bonding capability of injected foam. Hence it was considered necessary to verify their findings, in particular the accuracy of Eqs. (4) and (5) above, for higher strength steels with reduced ductility and polystyrene foam, which are bonded together with separate adhesives. The following sections describe the methods used in this investigation and the results obtained for this purpose.

3. Experimental analysis

A large number of steel plates with varying grades of steel and *b*/*t* ratios with and without polystyrene foam were tested in the structures laboratory. A mild steel grade G250 and a high strength steel grade G550 with minimum guaranteed yield stresses of 250 and 550 MPa, respectively, were chosen. For each steel grade, four different nominal (specified) thicknesses of 0.4, 0.6, 0.8 and 1.0 were used, representing the typical steel panels used in sandwich panel construction. In order to model the required simply supported boundary conditions along the longitudinal edges, a special test rig shown in Fig. 2 was made to test plates of width 100, 150 and 200 mm. These widths therefore gave 12 different *b*/*t* ratios for each grade of steel and a wide range of *b*/*t* ratios from about 105 to 513 (see Table 1). It is to be noted that the special



(a) Schematic Diagram



(b) Experimental Set-up

Fig. 2 Experimental set-up for steel plate elements with and without foam core boundary condition along the longitudinal edge of the plate attempts to simulate the real conditions present on the plates of the profiled face supported by adjoining plates. The vertical supports allowed shortening of plates and rotation about the vertical edges to occur freely (see Fig. 2). The plate specimen lengths were approximately three times the width in all the experiments. Such a length enabled formation of a number of elastic buckles within the plate length, but was short enough to eliminate any global buckling of the plate specimen.

A series of preliminary experiments were conducted with different foam core thicknesses equal to the plate width *b*, *b*/2 and *b*/4 in order to study the effect of foam core thickness on the experimental results. Since the foam core thickness appeared to have little effect on the results, a convenient foam thickness of *b*/2 was chosen for all the following tests. Steel plates and polystyrene foam cut to the required sizes were joined using a Thermosetting Urethane glue, which were then cured for 48 hours in room temperature before testing.

Table 1 shows the details of all the tests except the preliminary tests with different foam thicknesses. As seen in this table, there were 48 different cases, and with two tests for flat steel plates and three tests for flat plates with foam on one side, a total of 120 tests (48 flat steel plates and 72 steel plates with foam on one side) were conducted in this investigation.

The axial compressive load was applied to the steel plates via the specially made top and bottom loading blocks using a Tinius Olsen Testing machine (see Fig. 2). The load was always applied to the steel plate element whether it was foam-supported or not. The axial compression load, axial shortening and out-of-plane deflection were measured and were continuously recorded by a calibrated Labtek computer data acquisition system. The load at which local buckling of

Table 1 Test sections

Test Series	Plate Width *b* (mm)	G550 Steel Plates					G250 Steel Plates				
		Thickness (mm)		Measured		*b*/*t* Ratio	Thickness (mm)		Measured		*b*/*t* Ratio
				fy	*E*				*fy*	*E*	
		Spec.	bmt	(MPa)	(GPa)		Spec.	bmt	(MPa)	(GPa)	
1	100	0.42	0.42	726	239	238	0.40	0.39	368	220	256
2	100	0.60	0.60	682	235	167	0.60	0.54	360	218	185
3	100	0.80	0.80	656	230	125	0.80	0.73	345	217	137
4	100	0.95	0.95	637	226	105	1.00	0.93	326	216	108
5	150	0.42	0.42	726	239	357	0.40	0.39	368	220	385
6	150	0.60	0.60	682	235	250	0.60	0.54	360	218	278
7	150	0.80	0.80	656	230	188	0.80	0.73	345	217	205
8	150	0.95	0.95	637	226	158	1.00	0.93	326	216	161
9	200	0.42	0.42	726	239	476	0.40	0.39	368	220	513
10	200	0.60	0.60	682	235	333	0.60	0.54	360	218	370
11	200	0.80	0.80	656	230	250	0.80	0.73	345	217	274
12	200	0.95	0.95	637	226	211	1.00	0.93	326	216	215

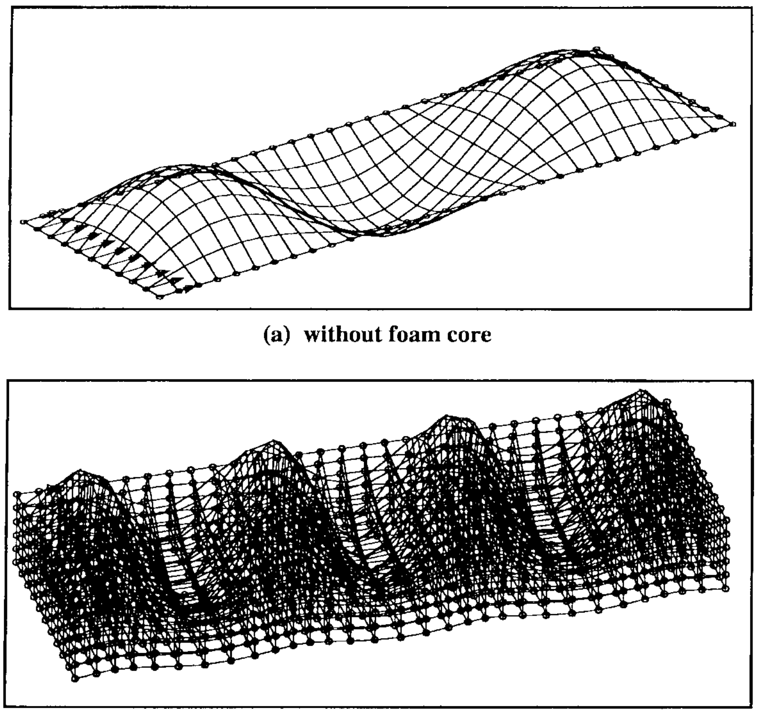
Note: Spec. Specified thickness
bmt – estimated base metal thickness based on measured total coated thickness
fy – measured yield stress of steel
E – measured Young's modulus
b/*t* ratio – Plate width b/bmt

plates initiated was observed for each test using out-of-plane deflection measurements. It was then determined accurately using load versus deflection curves. It was found that plates with foam core buckled at a higher load with a smaller half-wave buckling length. For flat plate elements, there were no more than three half-waves within the specimen length implying that the half wave buckling length was approximately equal to the width b, whereas for those with a foam core a maximum of 12 half-waves were observed (half-wave length=$3b/12$). These observations were similar to those from the finite element analyses shown later in Fig. 3.

Tables 2 and 3 present the average buckling stress results obtained from these experiments, and the corresponding K value using Eq. (1). The K value was calculated using the measured E and b/t values given in Table 1 and an assumed Poisson's ratio of steel of 0.3. The difficulty in obtaining the half wave buckling lengths meant only the approximate values were obtained and are therefore not reported in Tables 2 and 3. None of the test specimens showed any delamination between the plate and foam and thus confirmed the adequacy of glue used in sandwich panel construction. This therefore validates the finite element model of the composite steel plate and foam core used in this investigation.

4. Finite element analysis

In order to study the local buckling behaviour of plate elements with and without a foam core, a linear elastic buckling analysis was also conducted. The same plates considered in the



(a) without foam core

(b) with foam core

Fig. 3 Finite element analysis of steel plate elements with and without foam core

Table 2 Buckling stress results for G550 steel plates

Test Series	b/t Ratio	Without Foam				With Foam				R	a/b
		Buckling Stress (MPa)		Buckling Coeff. K		Buckling Stress (MPa)		Buckling Coeff. K			
		Expt.	FEA	Expt.	FEA	Expt.	FEA	Expt.	FEA		
1	238	15.7	15.0	4.1	4.0	115.5	105.5	30.3	27.7	51.4	0.34
2	167	31.8	30.1	4.2	4.0	113.3	114.9	14.8	15.1	17.9	0.47
3	125	54.4	52.4	4.1	4.0	137.5	128.9	10.3	9.7	7.7	0.61
4	105	76.8	72.5	4.2	4.0	158.5	142.8	8.6	7.7	4.7	0.70
5	357	7.1	6.7	4.2	4.0	103.2	99.8	60.9	59.1	173.6	0.23
6	250	13.6	13.4	4.0	4.0	105.6	103.9	31.1	30.6	60.6	0.32
7	188	23.8	23.3	4.0	4.0	113.3	109.8	19.2	18.7	26.1	0.42
8	158	33.7	32.2	4.1	4.0	130.5	115.1	15.9	14.1	15.9	0.49
9	476	3.6	3.7	3.8	4.0	109.5	98.0	115.0	102.8	411.5	0.17
10	333	7.3	7.5	3.8	4.0	99.2	99.9	51.9	52.2	143.6	0.24
11	250	13.4	13.1	4.0	4.0	98.8	103.0	29.7	31.0	61.9	0.32
12	211	18.4	18.1	4.0	4.0	113.2	105.9	24.6	23.1	37.6	0.37

Table 3 Buckling stress results for G250 steel plates

Test Series	b/t Ratio	Without Foam				With Foam				R	a/b
		Buckling Stress (MPa)		Buckling Coeff. K		Buckling Stress (MPa)		Buckling Coeff. K			
		Expt.	FEA	Expt.	FEA	Expt.	FEA	Expt.	FEA		
1	256	12.3	11.9	4.1	4.0	102.6	100.8	33.9	33.3	69.8	0.30
2	185	23.1	22.6	4.0	4.0	108.3	107.7	18.9	18.7	26.5	0.42
3	137	43.8	41.1	4.2	4.0	123.3	119.6	11.8	11.5	10.8	0.56
4	108	67.7	66.4	4.0	4.0	161.3	137.0	9.6	8.2	5.2	0.68
5	385	5.6	5.3	4.2	4.0	94.9	96.3	70.6	71.8	235.6	0.21
6	278	10.5	10.0	4.1	4.0	109.3	99.2	42.8	38.9	89.6	0.28
7	205	19.6	18.3	4.2	4.0	102.3	104.7	22.0	22.4	36.4	0.37
8	161	31.5	29.5	4.2	4.0	132.6	111.9	17.7	14.9	17.7	0.47
9	513	3.2	3.0	4.2	4.0	108.3	94.9	143.3	125.6	558.3	0.16
10	370	6.0	5.7	4.2	4.0	91.2	96.3	63.5	66.9	212.3	0.22
11	274	11.0	10.3	4.2	4.0	102.7	99.3	39.3	38.0	86.3	0.29
12	215	17.2	16.6	4.1	4.0	112.9	103.4	26.7	24.5	41.9	0.36

experimental analysis (see Table 1) were analysed using a finite element program NASTRAN (MSC 1991). The full length of the plate with and without foam was modelled using four-noded quadrilateral shell elements (CQUAD4) for the steel plate and eight-noded solid brick elements (CHEXA) for the foam core as shown in Fig. 3.

In the analyses, measured material properties of steel and polystyrene foam were used. Table 1 presents the measured values of yield stress and modulus of elasticity of steel (E) for varying grades and thicknesses. The measured E values for the thinner and high strength steels were found to be greater than 200~210 GPa that is usually assumed for steel. This unexpected observation has been reported by a number of researchers in the past (Mahendran 1996). The effect of cold-working on the thinner, high strength steels and that of coating are considered possible reasons for this observation. The measured average modulus of elasticity and shear modulus values of polystyrene foam were 3.80 and 1.76 MPa which correspond to a Poisson's ratio of 0.08.

In order to determine the effect of foam thickness on the buckling stress and the required finite element mesh density, a series of preliminary analyses were conducted for full length plates with a b/t ratio of 167 (E=200 GPa, b=100 mm and t=0.6 mm) and varying foam thickness (b, 0.8 b, 0.5 b, 0.25 b) and mesh density. It was found that the buckling load was not very sensitive to the foam thickness (load variation was less than 0.2%). This observation is similar to other preliminary experiments and hence confirmed the use of a foam thickness of $b/2$ in both experiments and finite element analyses.

The analysis of the full length plate as shown in Fig. 3 produced also the half-wave buckling length 'a'. Using this information and the geometry and loading symmetry conditions, a smaller plate model with a width of $b/2$ and a length of $a/2$ was created with appropriate boundary conditions (see Fig. 4). The required mesh density was determined by conducting a convergence study of the buckling analysis results for this smaller model. Based on these results, a 10×10 mm mesh density was chosen for the analysis. The foam core had five layers. It would be advantageous to use this smaller model, but the half-wave buckling length 'a' was not known for plates with foam cores and hence a full length model was analysed first to determine an approximate value of 'a'. The smaller model with width $b/2$ and length $a/2$ and a foam thickness of $b/2$ was then analysed in each case to obtain the critical buckling stress. Since the value of 'a' obtained from the full plate model was approximate, the length of the smaller model ($a/2$) was varied until the minimum buckling stress was obtained. This produced the exact half wave buckling length 'a' corresponding to the minimum buckling stress. A typical smaller model used for this purpose is shown in Fig. 4.

The buckling stress thus obtained and the corresponding half-wave buckling length 'a' as a ratio of plate width b are presented in Tables 2 and 3. These results are compared with the corresponding experimental results in these tables that includes the non-dimensional stiffness parameter R. The simplified R parameter was calculated using the measured material properties of polystyrene foam and steel for each b/t ratio using Eq. (2c). In addition to the mesh details, Fig. 3 also shows the buckled shape of the plate elements with and without foam. As seen in Fig. 3 and Tables 2 and 3, the half-wave buckling length 'a' has been reduced from b to a fraction of b due to the presence of foam core, but the buckling stress has increased considerably.

5. Comparison of results and discussion

In this section, local buckling results obtained from experimental and finite element analyses are compared with the design buckling formulae given by Eq. (4).

As seen in Tables 2 and 3, results from the finite element analysis and experiments agreed reasonably well. This provided confidence in reviewing the suitability of the design buckling equations for high strength plate elements with a polystyrene foam core. The results showed that the buckling coefficient K of all flat plates of varying b/t ratio and steel grade was about 4.0. This is an expected result, but the main interest of this project was the effect of polystyrene foam on the local buckling of plates with varying b/t ratio and thus R. Both analytical and experimental

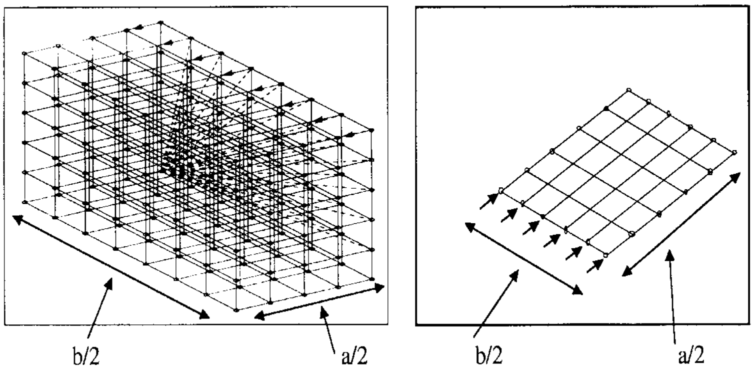


Fig. 4 Smaller symmetric model used in finite element analysis

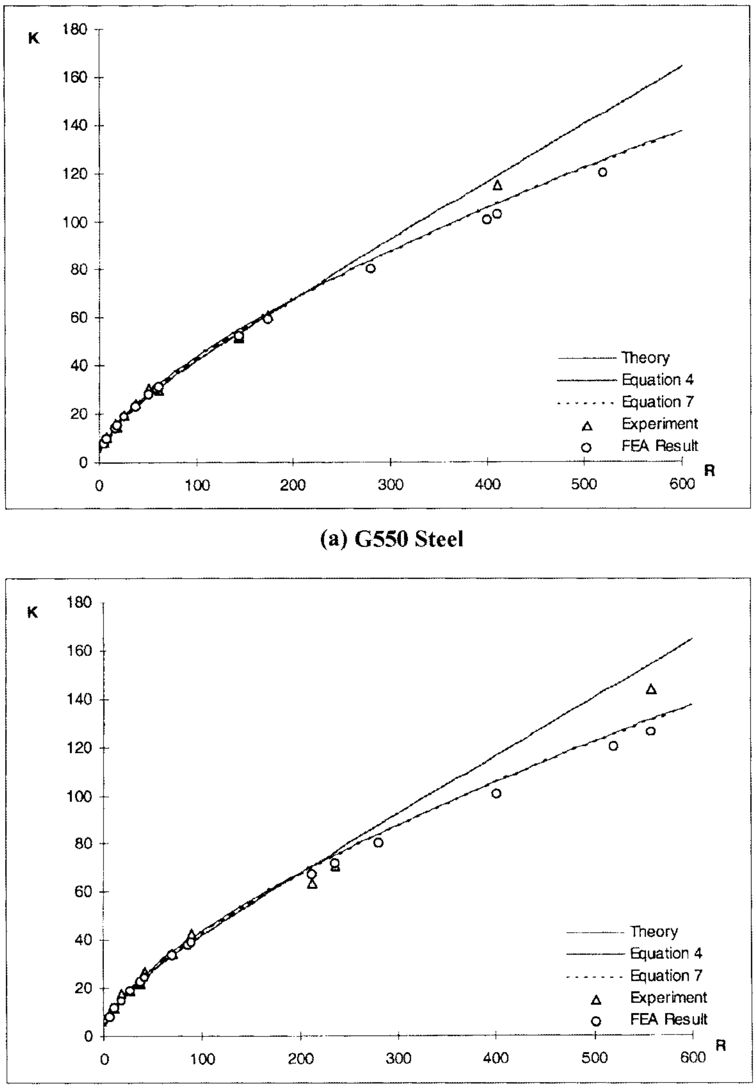
results clearly showed that the presence of polystyrene foam improved the local buckling of thin steel plates. The support provided by the foam core reduced the half wave buckling length from b to 0.15 b, but increased the K value from 4 to as high as 126 for the largest b/t ratio considered in this investigation.

Figs. 5(a) and (b) in the format of K vs R curves present the comparison of results from the finite element analyses and experiments with the design buckling Eq. (4) for higher strength steel G550 and softer lower strength steel G250, respectively. Results for both G550 and G250 steels compared reasonably well with the design buckling Eq. (4), and there was little difference in the results between G550 and G250 steels presented in Figs. 5(a) and (b). Eq. (4) agrees quite well with the results from this investigation for polystyrene foam for R values up to 200. Therefore it can be concluded that Eq. (4) combined with the simplified R value defined by Eq. (2c) can be used in the design of Australian sandwich panels using polystyrene foam and high strength steels although these equations were developed for polyurethane foam and softer low strength steels. For higher R values greater than 200, Eq. (4) appears to overestimate the buckling coefficient K.

Since Eq. (4) and the finite element analyses assume that the primary buckling mode is always critical (n=1), finite element analyses for all cases in Tables 2 and 3 were repeated using models of full width (b instead of $b/2$) in order to verify this assumption. Although Davies *et al.* (1991) indicates that this is true only for the range of R from 0 to 200, the analytical and experimental results from this investigation confirm that this assumption is also valid for higher values of R up to 600. This extends the applicability of Eq. (4) to higher values of R. However, since Eq. (4) was derived for R values in the range of 0 to 200, an improved equation for a larger range of R of 0 to 600 was derived using the same procedure described earlier in this paper (Davies *et al.* 1991). In this procedure, Eq. (3) was first solved for ϕ for varying R values up to 600, the buckling coefficient K was found using Eq. (2a) and finally an improved equation given by Eq. (7) was obtained based on curve fitting.

$$K = [16 + 4.76R^{1.29}]^{1/2} \tag{7}$$

Predicted results from the improved Eq. (7) and Davies *et al.*'s (1991) theory described in Section 2 are also included in Figs. 5(a) and (b), and are compared with results from Eq. (4) developed by Davies *et al.* (1991), FEA and experiments. As seen in Figs. 5(a) and (b), Eq. (7)

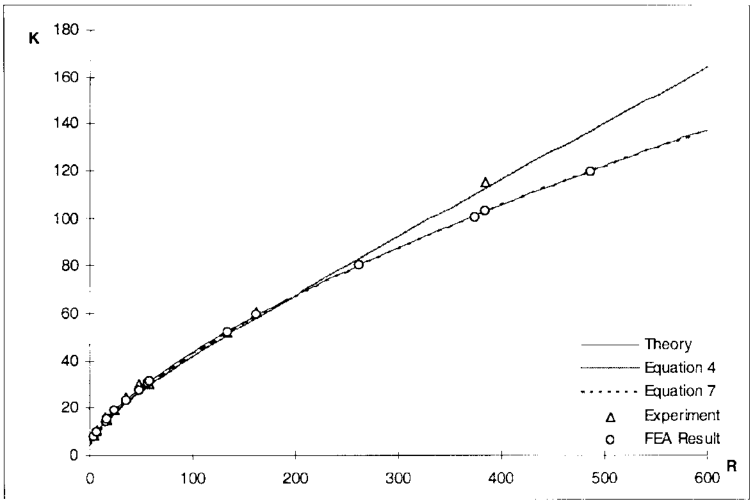


(a) G550 Steel

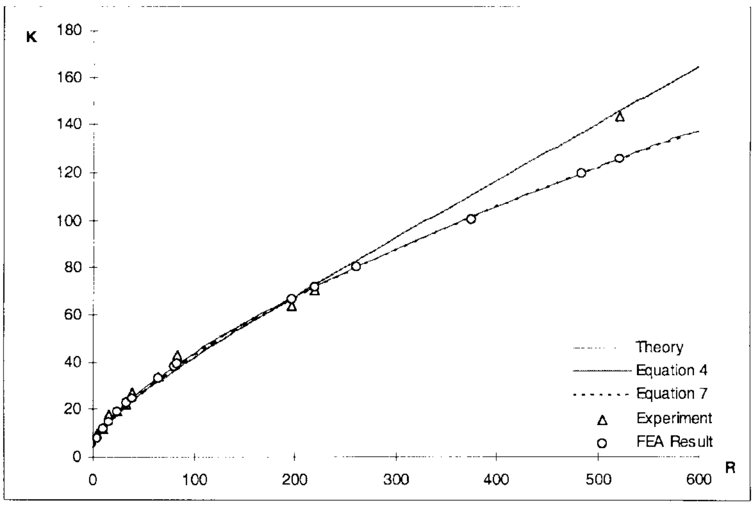
(b) G250 Steel

Fig. 5 Buckling coefficient K vs simplified R

provides a better agreement with results obtained from Davies *et al.*'s (1991) theory and FEA than Eq. (4) for higher values of R beyond 200. When compared with these results, Eq. (4) is not conservative for R values greater than 200, but Eq. (7) is conservative for all values of R up to 600. When all these results in Figs. 5(a) and (b) were re-plotted using the exact R values using Eq. (2b), the agreement between the results from Eq. (7), Davies *et al.*'s (1991) theory and FEA improved further (see Figs. 6(a) and (b)). Therefore Eq. (7) combined with the exact expression for R defined by Eq. (2b) should be recommended. However, the differences are rather small and appear only for higher R values. Hence, for the sake of simplified design equations, it may not be necessary to change the current use of Eq. (4) combined with the simplified expression for R defined by Eq. (2c) (ECCS 1991).



(a) G550 Steel



(b) G250 Steel

Fig. 6 Buckling coefficient K vs exact R

6. Conclusions

Local buckling behaviour of steel plate elements supported by a polystyrene foam core was investigated using finite element analyses and laboratory experiments. Finite element analyses included the composite behaviour of thin steel plate elements and polystyrene foam core using full-length models and appropriate smaller models based on symmetry. Local buckling stresses and associated half-wave buckling lengths obtained from experiments and finite element analyses agreed well. These results were then compared with the design buckling stress formulae available in European design documents, based on which appropriate design recommendations have been made on their suitability for Australian sandwich panels. An improved buckling formula has been developed to include more slender plate elements supported by foam core.

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