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Finite element Investigation of Design Formula for Thin Walled Plates Supported by Foam under Buckling Load

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ABSTRACT:
The thin-walled plate and foam of low material is components of sandwich panels, one of the attractive engineering structures mixed between different two materials to accomplish high ultimate strength integrated with light weight structures. Usually, the sandwich panels composed of two thin steel faces with ultra-strength are supported and separated between them by foam core. This paper deals with presently design expression of thin-walled steel faces supported by foam simulated using finite element analysis (FEA). Numerous waves of FE models introduced to explore and analyze the adequacy of present approach for thin-walled design. Also, investigation of FE simulation model containing geometry, dimensions, loading system and supporting pattern are presented. The FEA model provides a good acceptance with experimental results. However, for sandwich plate element with low (b/t) ratios, the formulae of design appeared safety solution, while for high (b/t) ratios (slenderness plate) becomes over estimated. Finally, formula of designing sandwich plate with local buckling is proposed in unification form.

KEYWORDS:Finite element simulation, local buckling; multi-waves model, Geometrical imperfection, Elastic buckling and ultimate strength of sandwich panel.

1- INTRODUCTION

In the recent years, it have been observed an increase in practical application of sandwich panels in civil engineering construction. The sandwich panels most consist of two steel plate and the relatively soft interior material. The faces carry normal stresses, while the three principal roles of the core are to carry shear stress, to protect the compressed faces against buckling and to provide thermal insulation. However, the use of flexible cores in sandwich structure causes more complex in their behaviour than that for plain plates. Davies and Hakmi (1990,1991) [1,2] investigated the sandwich panel subjected to uniform compression forces. They represented the panel as a simply supported plate resting on half space linear elastic foundation. Authors indicated that, the results of formula did not matched their corresponding practical results. Davies (1993) [3] introduced a good reviewing state of art in terms of structural load bearing capacity of sandwich panels design. Mainly two cases of the sandwich panel analysis are investigated. Mahendran and Jeevaharan (1999) [4] investigated the sandwich panels made of high steel strength (i.e. Australian panels). They adopted experimental and numerical programs for steel faces having different strength of yielding and thickness stiffened by polystyrene core. Modified design buckling formula is introduced based on FEA results. Mahendran and McAndrew (1999 and 2001) [5 and 6] considered wrinkling mode of failure for sandwich panels faces by plain steel sheets supported by polystyrene strips. In FEA, they used a half-wave buckled model to simulate a sandwich panel without joints which agreed well with the theoretical results. Both experimental and FEA with half-wave model results showed that, the face thickness and spume thickness ≥ 75mm do not influenced the wrinkling stress of flat face. Mahendran and McAndrew (2003) [7] expanded previous studies to comprise the flexural wrinkling failure mode with lightly profiled steel sheets and transverse joints. They carried out a groups of tests using two types of profiled face namely ribbed and satinlined. Pokharel and Mahendran (2002, 2004,2005,2008) [8,9,10,11] carried out extensive experimental test with various steel grade and plate dimensions to comprised wide margin for (b/t) ratio. They concluded acceptance results to employ the FEA model to simulate the experimental tested cases. They use half-waved model with wide margin for (b/t) ratio to introduce design formula of local buckling of sandwich panel.

J.S. Moita et al, (2015) investigated nonlinear analysis of multi-layers of sandwich structures in terms of buckling and geometry using finite element simulation. They indicated a specific assumptions to simulate multi-layers through displacement compatibility with allowing difference behaviours of each layer. They concluded the FES model depends on the ratio core to face thickness[12].

In this paper, a unified design formula is presented based on finite element analysis investigations. Essentially, the ultimate strength of sandwich panel is related to (b/t) ratio and material properties of faces and core. The waves-length model used for FEA was verified through comparison with experimental results.

2- LINEAR ELASTIC BUCKLING

As the core thickness is large compared with the face plate thickness, it is customary to consider only one plate face together with the core material. Thus, in elastic buckling analysis of sandwich panel, steel thin faces resting on the foam core with large thickness can be assumed as plates supporting by flexible basis as shown in Fig. 1. Rectangular plate with simply supporting all edges is submitted to compressive load (p) on opposite transverse sides. The critical stress of elastic buckling(σ_{crt}) of plate shown in Fig.1 as follows[1]:

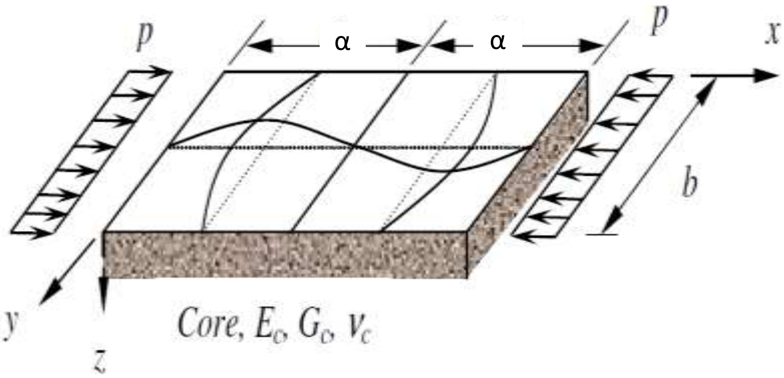

Fig. 1 Plate On Elastic Foundation [1]

$$\sigma_{crt} = Z \frac{\pi^2 E_s}{12(1-\upsilon_s{}^2)} \left(\frac{t}{b}\right)^2 \quad (1)$$

$$K = \left[\frac{1}{\phi} + n^2\phi\right]^2 + R\phi\sqrt{(1+n^2\phi^2)} \quad (2)$$

$$R = \frac{24(1-\upsilon_s{}^2)(1-\nu_c)E_c}{\pi^3(1+\nu_c)(3-4\nu_c)E_f} \left(\frac{b}{t}\right)^3 \text{(actual)} \quad (3)$$

$$R = \frac{12(1-\upsilon_s{}^2)\sqrt{E_c G_c}}{\pi^3 E_s} \left(\frac{b}{t}\right)^3 \text{(simplified)} \quad (4)$$

where:σ_{crt} = critical elastic buckling stress; E_s=plate elasticity constant of; E_c=core elasticity constant; G_c =core shear constant; v_s=poison's ratio of the plate; v_c=poison's ratio of the core; b =width of the plate; t_s =plate thickness; K=parameter of buckling; R=dimensionless stiffness parameter; n =term number buckling mode; ($\phi = \alpha/b$); α= length of half-wave buckle; and, b=width of the plate. As the halved wave buckling length decreases, the buckle parameter (K) increases which causes reduction in the critical buckling stress (σ_{crt}). Thus, the first critical buckling mode is obtained by minimizing the buckle parameter K regarding to ϕ (i.e. $\partial K/\partial\phi = 0$). Then Eq. (2) can be written as:

$$2n^4\phi - \frac{2}{\phi^3} + \frac{R(2n^2\phi^2+1)}{\sqrt{n^2\phi^2+1}} = 0 \qquad (5)$$

Eq.(5) is complicated to solve for ϕ, thus appropriated approximation method is used and the buckling parameter K can be obtained. However, for practical purposes, explicit mathematical formulae are required, and the following expressions have been proposed for the solutions of Eq.(5):

- By Hassinen (elastic half-space model) [13]:

$$K = 4.0 - 0.4150\,R + 0.7030\,R^2 \text{ where } R = \sqrt[3]{\frac{E_c}{E_s}}\left(\frac{b}{t}\right) \qquad (6)$$

- By Hassinen (simplified elastic half-space model) [13]:

$$\mathrm{K} = 4.0 - 0.4740\,\mathrm{R} + 0.9850\,\mathrm{R}^2 \text{ where } \mathrm{R} = \sqrt[6]{\frac{\mathrm{E_c G_c}}{\mathrm{E_s}^2}}\left(\frac{\mathrm{b}}{\mathrm{t}}\right) \qquad (7)$$

- Davies and Hakmi (simply basis modelling) [1]:

$$\mathrm{K} = \sqrt{16 + 11.8\mathrm{R} + 0.055\mathrm{R}^2} \qquad (8)$$

3- REVIEW OF LOCAL BUCKLING DESIGN FORMULA

The critical buckling stress does not introduced sufficient base for design, but it can be used as valuable factor. For compression members, the major design criterion based on local buckling, since the (b/t)ratios are large in cold-formed steel. Also, after elastic local buckling is occurred most of loading is supported by parts adjacent to boundary of the plate. Consequently, the effective width principle is considered in which on two strips of the plate, the maximum edge stress works in uniform shape while central regions is unloaded which is provided by Winter (1947) and the design formula listed as [1]:

$$b_{eff} = \rho b$$

$$\rho = \frac{1}{\lambda}\left[1 - \frac{0.22}{\lambda}\right] \quad \rightarrow \quad \lambda > 0.6730$$

$$\rho = 1.00 \quad \rightarrow \quad \lambda \leq 0.6730$$

$$\lambda = 1.0520\sqrt{\frac{f_y}{E_s Z}}\left(\frac{b}{t}\right) \qquad (9)$$

where: f_y=yield stress of face; The buckling parameter K is fixed for the plate unsupported by core (e.g. conditions with of simply supported plate K=4), while in the case of plate stiffened by core the K affected as the material properties of core foam and steel sheets and (b/t) ratio are changed. However, effective width techniques is expanded for corrugated sheets by improved the buckling parameter K values [1]. Davies and Hakmi (1990) carried out expanded experimental program on steel beam made of thin-walled steel sheets with supporting compression flange by foam to examine the ability to apply the effective width approach on sandwich panels. They concluded that, with increasing values of (b/t) ratios, the values of buckling parameter listed in Eq.(8) becomes unaccepted in terms of safety compared with test results observed. Thus, they provided modification of Eq.(8) by using *0.6R* instead of *R* as follows [1]:

$$K = \sqrt{16.0 + 7.0R + 0.02R^2} \qquad (10)$$

Mahendran and Jeevaharan (1999) proposed Eq.(11) for *K* based on FEA for steel sheets stiffened by soft material as follows [4]:

$$\mathrm{K} = \sqrt{16.0 + 4.76\mathrm{R}^{1.29}} \qquad (11)$$

Recommendations of European, Part-I:(CIB 2000) was established empirical reduction factor based on Davies and Hakmi recommendations[5], with simply expression to obtain the dimensionless argument *R* as stated in Eq.(12) [14]:

$$K = \sqrt{16.0 + 7.0R + 0.002R^2} \quad \text{for} R = 0.350\left(\frac{b}{t}\right)^3 \frac{\sqrt{E_c G_c}}{E_f} \qquad (12)$$

Pokharel and Mahendran proposed an improved effective width design formula Eq.(13) and Eq.(14) based on a the same series of test results and FEA as follows[10,8]:

$$\begin{aligned}
& b_{eff} = \rho b \\
& \rho = \frac{1}{\lambda}\left[1.0 - (\frac{0.22}{\lambda})\right]\frac{1}{\beta} \quad \rightarrow \quad \lambda > 0.673 \\
& \rho = 1.00 \qquad\qquad \rightarrow \lambda \leq 0.673 \\
& \lambda = 1.052\left(\frac{b}{t}\right)\sqrt{\frac{f_y}{E_f K}} \quad \beta = 0.0038\left(\frac{b}{t}\right) + 0.7627
\end{aligned} \qquad (13)$$

$$\rho = \frac{0.34}{\lambda}\left[1 + \frac{7.71}{\beta} - \frac{12.72}{\beta^2} + \frac{5.35}{\beta^3}\right] \qquad (14)$$

$$\lambda = 1.0520\sqrt{\frac{f_y}{E_f K}}\left(\frac{b}{t}\right)$$

$$\beta = 1.052\left(\frac{b}{t}\right)\sqrt{\frac{f_y}{E_f}}$$

$$\rho \geq 1.0 \quad b_{eff} = b$$

$$if\ \rho < 1.0 \quad b_{eff} = \rho b$$

4- EXPERIMENTAL DATA

Calibration of FEA method as an powerful analysis method to explore the behaviour of sandwich panel for local buckling and post-buckling conditions is carried out by considering the Pokharel's experimental programme and samples as details are specified in Table.1 with formal diagram of testing apparatus is shown in Fig. 2 [15].

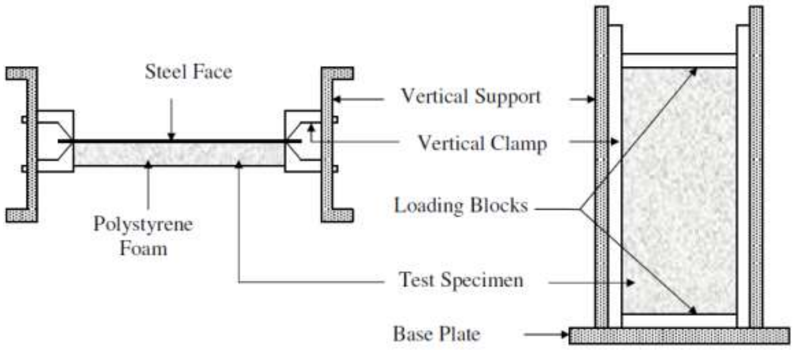

Figure.2 Formal diagram of test apparatus [15]

Table.1: Pokharel's Experimental Programme [2]

Test	B (mm)	Steel Plates G550					Steel Plates G250				
		thickness (mm)		measuring		(b/t)	thickness (mm)		measuring		(b/t)
		Spec.	bmt	fy (MPa)	Es (GPa)		Spec.	bmt	Fy (MPa)	Es (GPa)	
1	50	0.95	0.95	637	226	52.6	1	0.93	326	216	53.8
2	50	0.8	0.8	656	230	62.5	0.8	0.73	345	217	68.5
3	50	0.6	0.6	682	235	83.3	0.6	0.54	360	218	92.6
4	50	0.42	0.42	726	239	119	0.4	0.39	368	220	128.2
5	80	0.95	0.95	637	226	84.2	1	0.93	326	216	86
6	80	0.8	0.8	656	230	100	0.8	0.73	345	217	109.6
7	80	0.6	0.6	682	235	133.3	0.6	0.54	360	218	148.1
8	80	0.42	0.42	726	239	190.5	0.4	0.39	368	220	205.1
9	100	0.95	0.95	637	226	105.3	1	0.93	326	216	107.5
10	100	0.8	0.8	656	230	125	0.8	0.73	345	217	137
11	100	0.6	0.6	682	235	166.7	0.6	0.54	360	218	185.2
12	100	0.42	0.42	726	239	238.1	0.4	0.39	368	220	256.4
13	120	0.95	0.95	637	226	126.3	1	0.93	326	216	129
14	120	0.8	0.8	656	230	150	0.8	0.73	345	217	164.4
15	120	0.6	0.6	682	235	200	0.6	0.54	360	218	222.2
16	150	0.95	0.95	637	226	157.9	1	0.93	326	216	161.3
17	150	0.8	0.8	656	230	187.5	0.8	0.73	345	217	205.5
18	150	0.6	0.6	682	235	250	0.6	0.54	360	218	277.8
19	150	0.42	0.42	726	239	357.1	0.4	0.39	368	220	384.6
20	180	0.6	0.6	682	235	300	0.6	0.54	360	218	333.3
21	180	0.42	0.42	726	239	428.6	0.4	0.39	368	220	461.5
22	200	0.95	0.95	637	226	210.5	1	0.93	326	216	215.1
23	200	0.8	0.8	656	230	250	0.8	0.73	345	217	274
24	200	0.6	0.6	682	235	333.3	0.6	0.54	360	218	370.4
25	200	0.42	0.42	726	239	476.2	0.4	0.39	368	220	512.8
Note: bmt –base metal thickness											

5- FINITE ELEMENT MODEL

PATRAN 2010 is the finite element simulation (FES) software used as pre-and post-processer to explored behaviour of local buckling of sandwich panels under uniaxial load. The sandwich panel was simulated by four nodded quadrilateral-shell element (CQUAD4) to model the steel surface plate while eight nodes solid brick element (CHEXA) to the foam core. In FES method, the full-scale model is the easiest model (L×b), but the disadvantage related is that either low level of accuracy is found or a large computational time needed. In order to reduce difficulties, a half-length model (L/2×b/2) is used with corresponding boundary conditions due to symmetry as shown in Fig.s (3and 4). In addition, the halved multi-wave modelling (2.5α×b/2) can be used as reduced model as shown in Fig. 5, and the modelling length is variable and depended on the foam and faces properties and the (*b/t*) ratio. The current design expressions are examines and creative design formulae are achieved using multi-wave model and details is specified later.

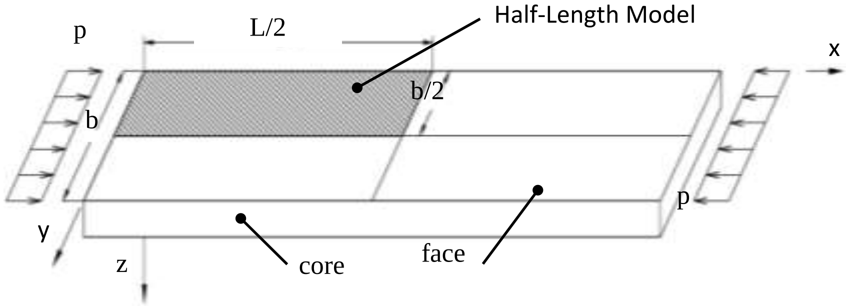

Figure.3 Principle of half-length model

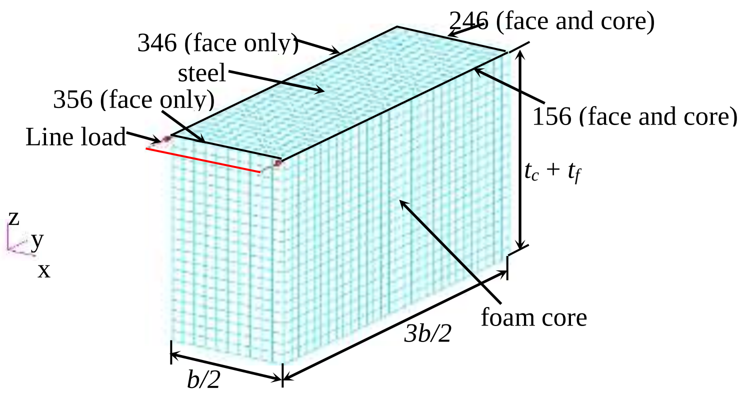

Fig. 4 Finite element half-length model with boundary conditions

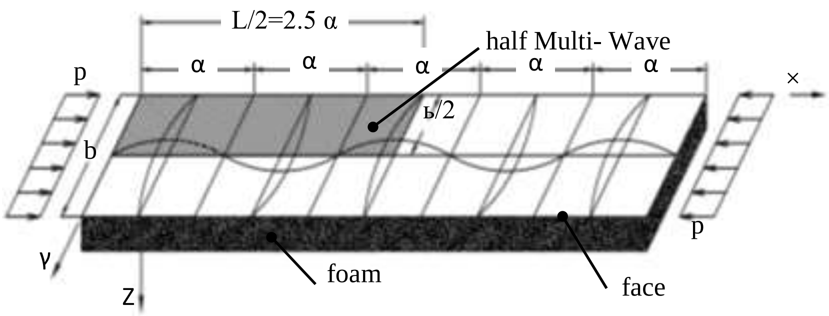

Fig. 5 Principle of half-multi-waves buckle model

6- NUMERICAL CONVERGENCE STUDY

Generally, the results of FEA model can be compared with experimental results to select element size achieved by FEA. Also, the computation time is increased as mesh density increased, thus an appropriate mesh size of FEA model must be chosen. Table.2 shows results of the numerical convergence study for H-L model for specimen no.10 (see Table.1).

Table.2: Numerical Convergence Study

Element mesh ize(mm)		Number of Elements	σ_{crt}(MPa)	σ_{ult}(MPa)
Shell	Solid			
25×25	25×25×25	60	129.63	319.35
15×15	15×15×15	240	122.43	214.40
10×10	10×10×10	825	120.57	184.04
10×10	10×10×5	1575	120.46	182.95

7- VERIFICATION OF FEA MODEL

To verify the FEA model, all samples were experimented by Pokharel and Mahendran are simulated using H-L model. Firstly, the buckling stress is obtained by carrying out linear elastic buckling analysis, then the first bucked mode is used as a shape of imperfection with ($0.1t_f$). Table.3 shows the results of buckling and ultimate stresses for FEA model comparing with the experimental results with G550 and G250 steel plates, respectively.

7-1 RESULTS VARIATION OF ELASTIC BUCKLING STRESS (σ_{crt})

The average ratio between buckling strength of FES model and tested programme were calculated (1.03) and (0.98) of the steel plate G250 and G550, respectively. The consistent variation parameters were (0.08) for G250 and (0.06) for G550.

7-2 RESULTS VARIATION OF ULTIMATE STRENGTH (σ_{ult})

For ultimate strength, the average ratio between FES and tested models of ultimate strength (σ_{ult}) were determined (0.91) for G250 and (0.84) for G550. And the consistent parameters of variation were (0.08) for G250 and (0.07) for G550. Through all presented comparing results, it is approved that the FEA method and H-L model are introduced sufficient accuracy and power full technique to model the non-linear or post buckling behaviour of sandwich panels.

8- MULTI-WAVED MODEL

In Pokharel and Mahendran experimental programme, the steel plate was restrained along the four edges while allowing the foam without any constraint, since the core continuing in both dimensions. According to local buckling behaviour of sandwich panel, initially at elastic buckling many waves is constructed in steel face, and their numbers and wavy length depend on (b/t) ratio and material properties [8]. At the post buckling stage these waves are rearranged depending upon the stress redistribution; until one of these waves reaches ultimate strength, then panel fails. In this paper, M-W model is investigated for different number of waves and (b/t) ratio with constant material properties(for face;E=233000MPa, υ=0.3, G=89615.40MPa, and F_y=675MPa, and for foam core; E=3.8MPa, υ=0.08, G=1.77MPa) as shown in Table.4. The results show that, for high (b/t) ratios, the model effects the ultimate strength which is expected due to redistribution during post buckling stage. Also, the model 5-waved model represented a suitable model for ultimate strength of M-W model, hence the ultimate strength become constant for different (b/t) ratio.

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Table.3: Comparing results of FEA andtests specimens for H-L Model

No.	Steel Plate-G550					Steel Plate-G250				
	(b/t) ratio	σ_{crt} (MPa)		σ_{ult} (MPa)		(b/t) ratio	σ_{crt} (MPa)		σ_{ult} (MPa)	
		FES	Exp.	FES	Exp.		FES	Exp.	FES	Exp.
1	52.60	337.53	336.20	400.10	434.30	53.80	313.82	272.50	273.02	285.40
2	62.50	264.22	293.00	358.65	428.30	68.50	223.41	221.90	236.1	251.50
3	83.34	189.17	238.30	305.42	305.00	92.60	160.68	151.90	196.37	201.50
4	119.00	133.44	141.90	264.26	264.30	128.20	120.63	146.70	164.6	186.20
5	84.20	177.96	170.30	259.48	279.20	86.00	168.33	171.20	186.82	201.60
6	100.00	149.11	155.80	222.73	275.00	109.60	131.17	152.60	153.23	185.40
7	133.34	118.51	112.90	190.09	223.30	148.10	107.6	98.10	130.41	159.00
8	190.50	100.70	97.00	163.15	186.00	205.10	94.87	88.10	111.84	149.00
9	105.34	138.05	139.40	203.67	232.60	107.50	131.33	123.40	148.25	178.10
10	125.00	120.46	132.80	182.87	203.40	137.00	109.9	121.40	130.03	162.10
11	166.67	103.24	102.00	153.18	181.80	185.20	95.18	90.90	110.21	148.10
12	238.10	91.67	87.60	129.20	184.00	256.40	87.18	76.90	94.9	111.30
13	126.34	116.96	122.10	173.61	205.20	129.00	112.32	119.40	129.77	150.20
14	150.00	106.03	119.80	156.36	203.50	164.40	98.18	98.60	114.09	126.40
15	200.00	92.93	91.00	129.73	169.90	222.20	87.71	85.20	96.93	113.30
16	157.90	100.24	104.60	143.27	158.10	161.30	96.88	96.80	110.82	125.50
17	187.50	93.10	96.10	129.37	172.80	205.50	87.45	87.20	98.18	101.90
18	250.00	85.16	83.90	109.25	133.90	277.80	80.46	76.50	86.11	91.20
19	357.10	79.88	79.00	93.05	119.20	384.60	76.81	67.20	78.29	80.30
20	300.00	79.71	78.60	96.53	122.60	333.30	75.86	73.60	78.98	86.00
21	428.60	76.04	77.80	84.54	118.40	461.50	73.4	64.10	73.69	78.30
22	210.50	83.63	88.20	113.62	136.50	215.10	83.26	84.90	91.77	91.60
23	250.00	81.58	79.90	103.57	117.10	274.00	77.68	72.70	83.04	78.00
24	333.34	76.94	71.40	90.37	118.00	370.40	73.46	65.70	75.55	71.90
25	476.20	73.98	80.00	80.50	100.10	512.80	71.53	64.60	71.51	75.60

Table.4: Comparing Results of Multi-Waved Model

(b/t)	σ_{crt} (MPa)			σ_{u} (MPa)		
	number of waves			number of waves		
	one	three	five	One	three	Five
62.5	271.71	272.57	272.65	345.91	346.55	345.93
125	134.09	134.80	134.41	201.71	203.46	203.35
200	109.06	109.81	109.91	136.34	144.23	144.19
250	103.54	104.26	104.36	120.99	129.35	129.74
500	96.63	96.97	97.03	96.80	104.47	104.67

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In addition, Table.5 shows comparison results of different foam core thickness of M-W model with different (b/t) ratios. It seems that, the foam thickness greater than 75mm is sufficient for simulation with marginal effect which agrees with that used by Pokharel and Mahendran [9].

Table5. Comparison With Different Foam Thickness

(b/t)	σ_{crt} (MPa) foam core thickness (mm)			σ_{ult} (MPa) foam core thickness (mm)		
	50	75	100	50	75	100
62.5	272.37	272.56	272.57	346.52	346.55	346.55
125	134.78	134.80	134.81	203.45	203.45	203.29
200	109.31	109.79	109.81	143.94	144.21	144.53
250	104.13	104.25	104.26	129.17	129.33	129.35
500	96.74	96.96	96.97	103.99	104.44	104.47

Also, Table.6 shows the results of single and double face cross section, the marginal effect is appeared on elastic buckling stress and ultimate strength which is confirmed Davies's principle.

Table6. Comparison Single And Double Faces Model

(b/t)	σ_{crt} (MPa)		σ_{ult} (MPa)	
	Single-face	Double-faces	Single-face	Double-faces
62.5	271.56	271.50	345.54	345.53
125	135.79	135.77	202.28	202.41
250	105.25	105.25	128.34	128.28
500	97.96	97.96	103.46	103.42

9- DESIGN FORMULA USING M-W MODEL

The proceeding results showed the verification of M-W model as acceptable FES model to analyse sandwich panel under local buckling load, and Fig.6 shows the geometry, boundary conditions and pattern load of model. The design formula of local buckling of sandwich panel is formulated as a relation between (b_{eff}/b) and (b/t) ratios, this relation is affected by mechanical properties of face and foam core.

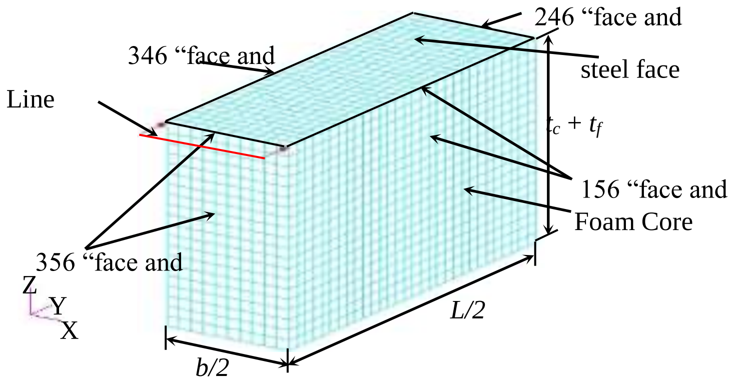
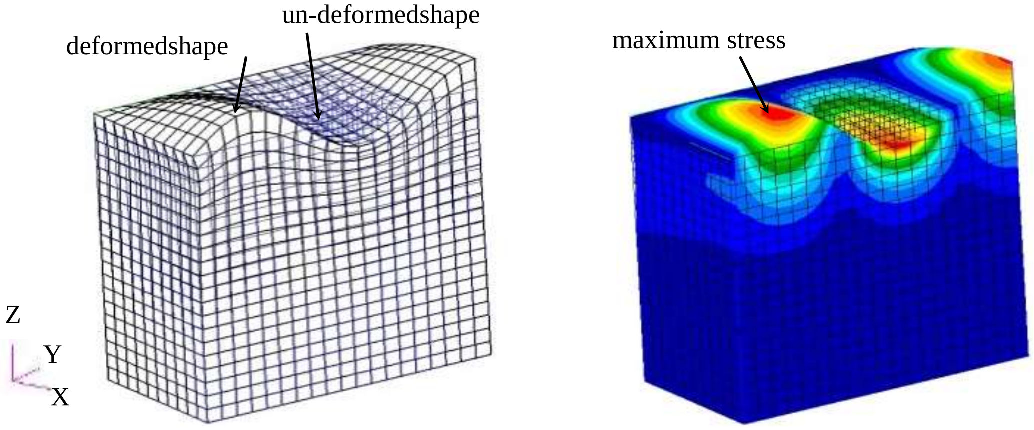


Figure.6 Multi- Wave Model

Thus, a parametric study is carried out with constant values of material properties for foam core and faces (which is average values of Pokharel and Mahendran experimental programme). Table.7 shows the results conducted using M-W models. Figure.7 presents the typical buckling mode and stress distribution of multi-wave model.Table.7 shows a good agreement has been utilized from results of FES in comparison with the theoretical results obtained from Eq.(1), (2), (4) and (5).



a-buckled shape **b- stress distribution**

Figure.7: Typical buckled shape and stress distribution M-W model

Table.7: Comparison Results of Multi-Wave model for steel face Grade-550.

No.	*b* (mm)	t_f (mm)	*(b/t) ratio*	*a* (mm)		*L/2* (mm)	σ_{crt} (MPa)		σ_u (MPa)
				Theo.	FEA		Theo.	FEA	
1	50	0.8	63	41.8	42.4	106	278.70	272.57	346.55
2	50	0.7	71	39.4	40.0	100	233.64	227.65	309.75
3	50	0.6	83	36.3	36.8	92	195.70	189.82	273.76
4	50	0.5	100	32.4	32.8	82	164.67	159.00	238.42
5	50	0.4	125	27.6	28.4	71	140.17	134.80	203.29
6	100	0.7	143	49.7	50.8	127	130.23	125.09	177.26
7	100	0.6	167	43.8	44.8	112	121.74	116.74	160.21
8	100	0.5	200	37.4	38.4	96	114.64	109.81	144.23
9	150	0.7	214	52.8	54.4	136	112.58	107.79	137.61
10	100	0.4	250	30.6	31.6	79	108.89	104.26	129.35
11	200	0.7	286	54.0	55.6	139	106.51	101.88	120.57
12	150	0.5	300	38.7	40.0	100	105.79	101.25	119.16
13	200	0.6	333	46.7	48.0	120	104.45	99.94	114.46
14	250	0.7	357	54.7	56.4	141	103.72	99.17	111.83
15	150	0.4	375	31.3	32.4	81	103.25	98.85	111.15
16	200	0.5	400	39.2	40.4	101	102.71	98.29	109.15
17	250	0.6	417	47.2	48.4	121	102.40	97.96	107.82
18	300	0.7	429	55.1	56.8	142	102.20	97.70	106.94
19	200	0.4	500	31.6	32.8	82	101.29	96.97	104.47
20	300	0.5	600	39.7	40.8	102	100.51	96.18	101.80

10- EFFECTIVE WIDTH COMPARISON

The effective width ratio (b_{eff}/b)of thin steel plate stiffened by foam can be calculated using ultimate stresses given in Tables.7 divided by f_y as in the following formula:

$$\frac{b_{eff}}{b} = \frac{\sigma_u}{f_y} \tag{15}$$

Figure.8 shows the comparison results of (b_{eff}/b)ratios versus(b/t) ratio of FES model and that calculated by Eq.(9) using K values indicted by Davies and Hakmi (1990) and ECCS(2000) [14], and those evaluated from Eq.(13 and 14) using proposed by Pokharel (2002 and 2005, respectively).

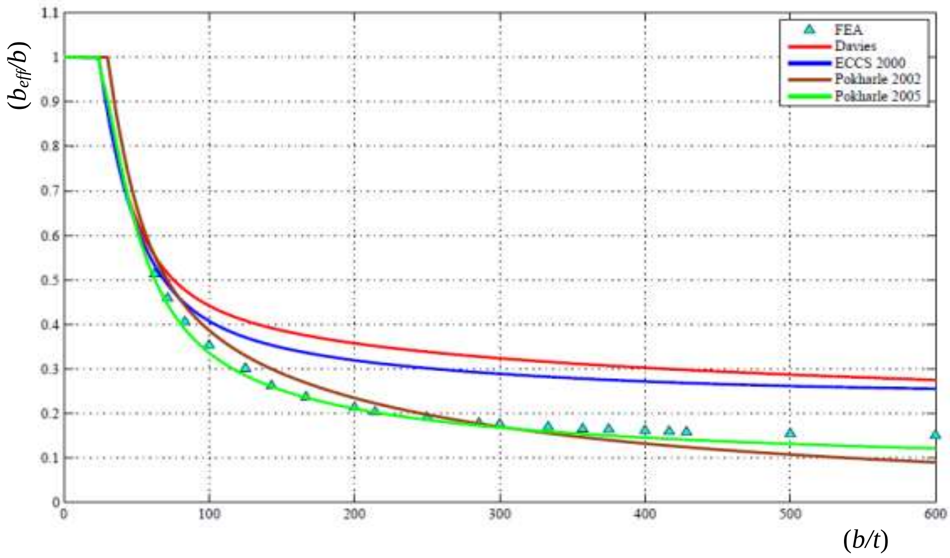


Figure8. Comparison results of FEA model with different models for G550

For low (b/t)ratios (<100), the results of (b_{eff}/b) ratio for FES model agreed well with results of Eq.(9), while for high slenderness (b/t)ratios (> 100), the current design formulae of ECCS-2000 results are overestimated in comparison with the results of FES model. In addition, Fig.9 appears that, the FEA results indicated using proposing design formula of Pokharel and Mahendran (2002 and 2005) by ABAQUS gives reasonable values of (b_{eff}/b) ratiofor moderate slenderness (b/t) ratios (<300), and underestimation for high slenderness (b/t) ratios (>300). However, these formulae of Pokharel and Mahendran neglected the effect of stress redistribution for high (b/t)ratios.

11- PREDICTION DESIGN FORMULA

In the previous articles, the buckling behaviour of sandwich panels was investigated which is the first step to predict a design formula. Based on FEA results using M-W model and best fit techniques, the basic relationship of (b_{eff}/b) and (b/t) ratios can be approximated as following:

$$\left({}^{b_{eff}}/_{b}\right) = \mu\left({}^{t}/_{b}\right)^{\gamma} + \delta \tag{16}$$

where: μ, γ, and δ constants related to material properties of face and core. To determine the unified values of μ, γ, and δ constants, all mechanical properties are assumed to be constant except the yield stress of steel faces due to its effects on the ultimate strength of sandwich panel. Figure.9 shows the best fitting results process to the data shown in Fig. 8 with the prediction design formula as following:

$$\left({}^{b_{eff}}/_{b}\right) = 70\left({}^{t}/_{b}\right)^{-1.23} + 0.15 \qquad (17)$$

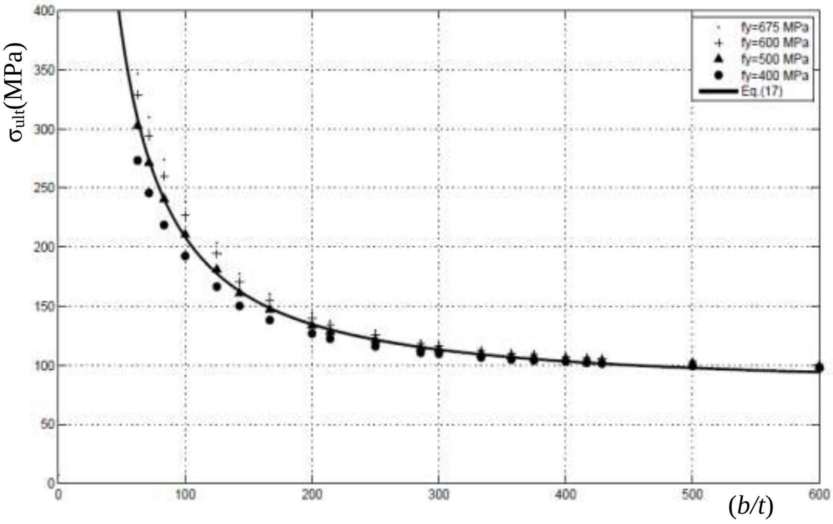


Figure9. Comparison results of FEA model with different models for G550

12- CONCLUSIONS

(1) The finite element method represents one of the powerful methods to investigate the profiled sandwich panel, and good agreement has been achieved in comparison with tested results utilized by Pokharel and Mahendran (2003) for elastic buckling and non-linear analysis.

(2) The multi-wave models are checked in terms of geometry, dimensions and boundary conditions to insure a good practical simulation of sandwich panels. Also, the wave length model is examined through number of waves (i.e. one, two, three, five, and seven waves) with constant mechanical properties for different (*b/t*) ratio. The 5-waves model presents a suitable model for obtaining the ultimate strength of sandwich panel.

(3) The use of foam core thickness greater than 75mm in FES model is sufficient thickness for simulation with only a marginal effect on elastic buckling stress and ultimate strength.

(4) The M-W model used to review the current design formula which is gave acceptable results for low (*b/t*) ratio whereas inadequacy agreement to higher slenderness or high (*b/t*) ratio.

(5) A prediction design formula of calculating the ultimate strength of sandwich panel is introduced basis on FES model in term of three constants factors depended on material properties of sandwich panel (i.e. faces and core).

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