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## **Flexural Wrinkling Strength of Lightly Profiled Sandwich Panels with Transverse Joints in the Foam Core**

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### **Summary**

In Australia, sandwich panels are commonly made of flat or lightly profiled steel faces and expanded polystyrene foam cores, and are increasingly used as roof and wall claddings in industrial and commercial buildings. Flexural wrinkling is often the governing criterion in the design of these panels. The use of lightly profiled faces is expected to increase the flexural wrinkling stress considerably whereas the presence of joints between the polystyrene foam slabs in the transverse direction introduces a reduction to the flexural wrinkling stress. Therefore a series of full scale experiments and finite element analyses were conducted to evaluate the effects of lightly profiled faces and transverse joints on the flexural wrinkling stress of panels subjected to a lateral pressure loading. The results showed that properly designed lightly profiled panels provided considerable improvement to flexural wrinkling strength over flat panels whereas the presence of transverse joints caused a significant reduction to wrinkling strength. This paper presents the details of this investigation, the results and comparison with available theoretical and design solutions.

**Keywords:** Lightly profiled sandwich panels, Flexural wrinkling, Transverse foam joints, Experiments, Finite element analyses

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## **Introduction**

In Australia, sandwich panels are commonly made of flat or lightly profiled steel faces and expanded polystyrene foam cores (Figure 1), and their use as roof and wall claddings in industrial and commercial buildings has increased rapidly in recent times. Flexural wrinkling is a unique failure associated with flat and lightly profiled sandwich panels when subjected to bending or compressive loads. This wrinkling failure occurs well below the yield stress of the compressive face, and thus is the governing design criterion for most sandwich panels (Figure 2).

The use of lightly profiled faces is expected to increase the flexural wrinkling stress considerably. However, the method of using polystyrene foam slabs leads to the presence of transverse joints in sandwich panels. The presence of these transverse joints implies an imperfection in the continuously supported plane of the faces (i.e., a ‘gap’ between the foam slabs and a ‘step’ due to the difference in the heights of foam slabs) as shown in Figure 3. The joint introduces two potential sources of weakening into the panel. The ‘gap’ between the foam slabs in the transverse direction introduces a possible reduction in shear strength of the panel. Davies and Heselius (1993) indicate that if such a joint were to extend completely across the panel, the shear

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strength along that line would be close to zero. The second potential source of weakening is associated with the effect the 'gap' and 'step' has on the flexural wrinkling failure stress of the panel. The third possible weakening is due to the interaction of shear and wrinkling effects.

Extensive sandwich panel research has been carried out in Europe and America that has led to many useful research papers and design documents (Davies, 1987,1993, Hassinen, 1995, ECCS, 2000). ECCS (2000) recommends that no transverse joints should extend across more than one quarter of the panel width. However, the Australian manufacturers have been using full-width foam joints and therefore initiated an investigation to determine the weakening effects of these transverse foam joints for lightly profiled sandwich panels when used as simply supported single spans. A series of full scale experiments and finite element analyses were conducted to evaluate the effects of lightly profiled faces and the transverse joints on the flexural wrinkling stress of panels subjected to lateral pressure loading. This research considered mainly the weakening effect on the flexural wrinkling strength, and not the effects due to shear or the interaction of shear and wrinkling. This paper presents the details of this investigation, the results and comparison with available theoretical and design solutions.

### **Wrinkling Theory**

Wrinkling is a unique form of local instability of the flat and lightly profiled steel faces of a sandwich panel associated with short waves of buckling. It can occur when a sandwich panel as a whole is subjected to a compression load or when one of the

steel faces is in membrane compression under bending of the panel (Figure 2). If failure is to occur due to wind load, this unique wrinkling failure appears to be the most likely failure mode for flat and lightly profiled sandwich panels used as roof and wall claddings. The compressive face of a sandwich panel in bending develops a series of buckling waves at a relatively low stress level, well below the yield stress of the steel face. The amplitudes of these waves gradually increase until one of them yields and forms a wrinkle at the location of greatest bending moment and/or imperfections in the panel. The flexural wrinkling failure occurs rather suddenly, and this characteristic must be considered in design. The thin flat and lightly profiled steel faces of sandwich panels carry membrane forces only (no moment). Their failure mode, therefore, is considered like that of a thin compressed beam on an elastic foundation (i.e., the foam core).

Equation 1 gives the theoretical expression for the wrinkling stress  $\sigma_{wr}$  of sandwich panels with flat faces that was based on an energy method (Davies et al., 1991). It is assumed that the steel face is supported on an infinitely deep elastic support (foam core). It is important to realise that Equation 1 has been derived assuming a continuous elastic medium.

$$\sigma_{wr} = 1.89 \left\{ \frac{8(1 - \nu_c)^2}{12(1 - \nu_f^2)(1 + \nu_c)(3 - 4\nu_c)^2} \right\}^{\frac{1}{3}} (E_f E_c G_c)^{\frac{1}{3}} \quad (1)$$

where  $E_f$  and  $E_c$  are the modulus of elasticity of the face and foam core;  $\nu_f$  and  $\nu_c$  are the Poisson's ratio of the face and foam core, and  $G_c$  is the shear modulus of the foam core. When  $\nu_c = 0$  and  $\nu_f = 0.3$ , Eqn.1 becomes

$$\sigma_{wr} = 0.819(E_f E_c G_c)^{\frac{1}{3}} \quad (2)$$

Lightly profiled faces are generally considered those with a profile depth of 1 to 2 mm. Even with such a small depth of profile, a significant increase in both the wavelength of buckling and wrinkling stress can result according to several researchers (Davies, 1993, Hassinen, 1995, Kech, 1991). Although lightly profiled panels can significantly increase the wrinkling stress, flat plate buckling between the folds can occur as the depth or the spacing of the profile increases. This must be considered in design. The current design procedures for a lightly profiled face in compression are based on modifying the methods utilised for flat faces. The analysis for flat faces is simply modified by taking into account the bending stiffness of the lightly profiled face. The effect of foam core depth is again disregarded (Davies et al. 1991), resulting in Equation 3.

$$\sigma_{wr} = \frac{1.89}{A_f} \left[ \frac{8(1 - \nu_c)^2 E_c G_c B_f}{(1 + \nu_c)(3 - 4\nu_c)^2} \right]^{\frac{1}{3}} \quad (3)$$

where  $A_f$  = cross-sectional area of face per unit width

$$B_f = \text{flexural rigidity of face per unit width} = \frac{E_f I_f}{b}$$

When  $\nu_c = 0$  and  $\nu_f = 0.3$ , Eqn.3 becomes

$$\sigma_{wr} = \frac{1.82}{A_f} (E_c G_c B_f)^{\frac{1}{3}} \quad (4)$$

For design purposes, a coefficient of 0.65 is used instead of 0.819 for flat faced panels in Eqn.2 (a reduction factor of about 0.8), and  $E_c$  and  $G_c$  are the characteristic values, ie. the 5% fractile values of the population (ECCS, 2000). This is due to practical considerations such as finite core depth, lack of flatness of the face, and non-linearity and lack of homogeneity of the core material (Davies et al., 1991), but not intended for the presence of transverse joints in the foam. For lightly profiled panels, a coefficient of 0.95, (ie. a reduction factor of approximately 0.5) is used to allow for the practical considerations mentioned above, and not for the transverse foam joints. Since the analysis is based on a simple modification for flat panels, the interaction between local buckling of the flat parts and buckling of the complete face is not included in the analysis for lightly profiled faces. Kech (1991) has developed an improved model for lightly profiled faces that takes into account the interaction between the two buckling modes. However, further testing of lightly profiled panels and an exact analysis is deemed to be required (Davies, 1993). The reduction factor of 0.5 for lightly profiled panels is considered too severe by the sandwich panel manufacturers and designers. It is considered that more accurate wrinkling stress values can be obtained by testing.

The current design formulae/ methods as outlined above do not account for transverse joints in the foam core of flat and lightly profiled sandwich panels. They have been derived for panels with continuous foam cores. Therefore this investigation is aimed at determining both the effects of lightly profiled faces and transverse joints on the flexural wrinkling stress of sandwich panels. The effect of transverse joints on flat faced panels is given in McAndrew and Mahendran (1999).

## **Experimental Investigation**

### **Details of Experiments**

Twenty-eight sandwich panels with lightly profiled faces were considered in this study. Both profiles, ribbed and satinlined, shown in Figure 1 were included. The foam thicknesses and grades considered were 75, 150 and 200 mm, and SL (average measured density =  $13.1 \text{ kg/m}^3$ ) and M (average measured density =  $18.1 \text{ kg/m}^3$ ), respectively. The span was varied from 2700 to 4700 mm, but the width of all panels was 1215 mm. Steel thickness was also varied: 0.4, 0.5 and 0.6 mm. The joint location was varied, however, it was kept at midspan for the majority of test panels. Panels were manufactured with two types of foam joints, full-width butt and scarf joints, as shown in Figure 3. For each panel, the gap and step imperfections were measured. Table 1 presents the details of lightly profiled panels tested in this study.

A vacuum chamber was used to produce a uniformly distributed transverse loading of the panels (see Figure 4), enabling flexural wrinkling failures (see Figure 2) to occur in bending. Test panels were simply supported over 70-mm wide rectangular hollow sections and were not restrained by the timber casing. In each test, the steel face to be tested was placed on the top so that it was subjected to a compression force. Once the panel was positioned in the chamber, a polyethylene sheet was placed loosely over the panel. The sheet was sealed to the sides of the timber casing and a vacuum pump used to decrease the air pressure in the chamber. A pressure transducer and five displacement transducers were used for each test. The support reactions were not measured as the test panels were simply supported. In some tests, strain gauges were

used at midspan and/or at the joint to validate the stresses calculated from pressure gauge measurements. Since these results agreed well, strain gauges were not used on other test panels. Experimental data was collected continuously by a computer-controlled loading system until panel failure.

For all panels without joints, flexural wrinkling occurred within the middle third of the panel. It is interesting to note that no panel without a joint actually failed at midspan at the location of greatest bending moment. Most of the panels with joints failed at the joint. The others failed within 300 mm of the joint. In all the tests, the panels collapsed suddenly with no post-wrinkling strength, indicating the nature of these flexural wrinkling failures (see Figure 2). Failure involved a distinct inward fold of the steel with tensile failure of the foam occurring directly beside the fold. This failure type occurred in every test panel under bending. The development of short buckling waves as predicted by wrinkling theory was not observed due to the sudden nature of the failure and small amplitude and length of the buckling waves. However, for panels with transverse joints, some buckling waves were observed at the joints prior to flexural wrinkling failure. No glue failure was apparent in the tests. Table 1 shows the results for the lightly profiled panels tested in this investigation.

The measured suction pressure at failure ( $p$ ) in Table 1 was used to calculate the wrinkling stress  $\sigma_{wr}$  for each test using the following equation.

$$\sigma_{wr} = M / (e A_f) \quad (5)$$

where  $M$  = maximum moment in the panel =  $(p \times \text{panel width} \times \text{span} + W) \times \text{span}/8$



$A_f$  = cross-section area of profiled face

$e$  = Distance between centroids of the steel faces

The selfweight of panel 'W' was also included in the calculations.

The following measured average material properties of foam were used in the calculation of theoretical wrinkling stresses using Eqns. 2 and 4:  $E_c = 3.44$  MPa and  $G_c = 1.72$  MPa,  $\nu_c = 0$  for SL Grade foam and  $E_c = 5.40$  MPa and  $G_c = 2.70$  MPa,  $\nu_c = 0$  for M Grade foam. The Young's modulus  $E_c$  of the foam was taken as the average of the tension and compression moduli. The material properties of steel faces were assumed to be  $E_f = 200,000$  MPa and  $\nu_f = 0.3$ . The design wrinkling stresses were based on characteristic values:  $E_c = 2.77$  MPa and  $G_c = 1.385$  MPa,  $\nu_c = 0$  for SL Grade foam and  $E_c = 4.76$  MPa and  $G_c = 2.38$  MPa,  $\nu_c = 0$  for M Grade foam. The design and theoretical wrinkling stresses (Eqn.4) are then compared with experimental stresses in Table 1.

## **Experimental Results and Discussion**

The failure stress at midspan  $\sigma_{wr}$  is used in the comparison of results as this is the maximum stress in the panel that initiates wrinkling. The mechanism for the observed reduction in flexural wrinkling capacity due to the presence of joints is not obvious. The calculated stresses at the wrinkle based on failure pressures were considerably less than the corresponding midspan stresses when the joint was located away from midspan. However, larger localised stresses could have been present at the joints, which caused the premature failures at the joints. The failure mechanism for panels with joints must be considered to be different to the theoretical mechanism whereby

buckling waves are formed with one becoming unstable at the critical stress to form a fold. Further research is being conducted to study the failure mechanism in detail.

Test results for the Ribbed Panels (Panels 1 and 5) show that experimental stresses are considerably less than the theoretical stresses. The expected increase in wrinkling stress due to the lightly profiled face was not realised. As seen in Equations 3 and 4, the wrinkling stress depends on the flexural rigidity parameter  $B_f$  of the steel face, which is very sensitive to the variations in rib/ridge height of the steel face. In the case of both ribbed and satinlined panels, the specified rib/ridge height was 0.7 and 0.8 mm, respectively (Figure 1), and thus even small reductions in these heights such as 0.2 mm could lead to considerable reductions to  $B_f$  and thus wrinkling stresses (see Table 2 in Section 4). However, the finite element analyses of ribbed panels showed that the reduction was mainly due to the local buckling of ribbed faces.

Test results for the Satinlined Panels (Panels 4, 6, 7, 18, 23, 26) show that experimental wrinkling stresses in many cases agreed with the higher theoretical stresses. For some panels, the measured ridge heights varied across the panel width and were below the nominal value of 0.8 mm. This explains the reduced wrinkling stresses in these cases. Therefore it can be concluded that satinlined profiles are capable of increasing the wrinkling strength of sandwich panels in comparison to flat panels, however, manufacturers should adopt a quality control process to ensure that small ridge heights are achieved in practice. The glued interface between the foam core and steel faces must be continuous and have adequate strength.

Test results indicated that ribbed and satinlined sandwich panels, which contained transverse joints, had reduced flexural wrinkling capacity than panels which contained no joints (by up to 30%). This is shown in Table 1 by comparing Panel 1 ( $\sigma_{wr} = 98.6$  MPa) which contained no joints with Panels 2 and 3 ( $\sigma_{wr} = 79.6$  and  $80.4$  MPa) with transverse joints. Similarly Panel 23 can be compared with Panels 19, 20, 21 and 22. For a large proportion of other panels with joints, failure occurred at stresses less than for panels without joints. The panels with joints failed at the joint in almost all cases. These observations indicate that panels with transverse joints have an inherent weakness at the joint. On average, the reduction to wrinkling stresses is about 20%. Therefore, for panels containing joints, designers should be aware that the design equation may not be conservative and that a more generous empirical reduction factor may need to be applied. Another approach is to modify the transverse joint (glued or shorter width joints) so that an improved form of connection between the slabs of foam is achieved.

The use of scarfed joints appeared to improve the wrinkling stress in some cases, but there is hardly any difference between the scarfed and butt joints in most other cases. In fact, in a number of tests, full-width butt joints (Panel 10) performed better than scarfed joints (Panel 8). However, the latter may be preferred with regard to possible shear failures when the joints are near the support.

Experiments confirmed that the effects of foam thickness and span on the flexural wrinkling failure stress of a sandwich panel are negligible. It was found that increasing the foam thickness beyond 75 mm lead to only a marginal increase in wrinkling strength for sandwich panels considered in this investigation. In contrast,

the steel thickness appeared to affect the wrinkling stress unlike for flat panels. It was found that the location of joints within the span did not alter the wrinkling stress. However, for panels with full-width joints near a support, shear failure can become the critical failure mode. Panel 17 was an excellent example of a shear failure where the failure stress was very low. This should be avoided in design.

The design wrinkling stresses are in general too conservative (see Table 1). In fact, the design stress for some ribbed panels using the coefficient of 0.95 in Equation 4 is less than that for flat panels obtained using the coefficient of 0.65 in Equation 2, and thus the latter design stress has been used. This is because the design stress of lightly profiled panels cannot be smaller than that of flat panels with the same steel face thickness. These observations indicate the conservative nature of current design equations for lightly profiled panels.

The presence of step and gap imperfections can lead to a larger region where the steel face is not supported by the foam. The step and gap imperfections for all panels with transverse joints were measured (Table 1), however, they should be considered as estimates only. Although the gap size was almost zero for all test panels, there was an immediate reduction to wrinkling strength. Although the effect of increasing gap size was not determined, the gap size should be kept to a minimum. The step sizes varied from 0 to 1.0 mm in the tests, but no meaningful trend could be observed. Therefore a finite element analysis was carried out to study the effects of these imperfections, and the following section describes this.

### **Finite Element Analyses**

Flexural wrinkling behaviour of sandwich panels with transverse joints was also investigated using finite element analyses (FEA). It was considered that FEA could be used to isolate the effects of joints and lightly profiled faces in greater detail. A finite element program Abaqus (HKS, 1998) was used for this study. S4R5 shell elements with four nodes and five degrees of freedom per node were chosen for the steel faces whereas C3D8 3D brick elements with eight nodes and three degrees of freedom per node were used to model the foam. The material properties of foam as measured in the experimental study were used in the analysis (see last section) whereas for steel they were assumed to be  $E_f = 200,000$  MPa and  $\nu_f = 0.3$ . Both materials were considered to be isotropic. Since wrinkling occurs in the elastic region (well below the yield stress of steel faces) and is a bifurcation phenomenon with little postbuckling strength, an elastic buckling analysis was used. This analysis did not include the effects of any nonlinear variation and the lack of homogeneity of foam material properties or panel imperfections. However, it was considered adequate as the numerical results combined with experimental results could be used to assess the effects of such imperfections. The numerical analyses of ideal models could still be used to investigate the effects of important parameters such as joints.

The initial finite element model simulated a steel face with all four sides simply supported subject to an end compression load as for the theoretical approach using energy method so that it can be calibrated against the available theoretical predictions (Eqn.4). The foam core was considered to be sufficiently wide and deep to simulate the conditions assumed in the theoretical method. A single half-wave buckle was modelled with appropriate boundary conditions including that of symmetry. Figure 5a

shows the model geometry (half the panel width  $b/2$  x half-wave buckling length  $a/2$  x model depth = foam depth  $t_c$  + steel face  $t_f$ ) and mesh used by McAndrew and Mahendran (1999) for panels with flat faces. This enabled a finer mesh to be used while enabling both the wrinkling stress and the associated half-wave length to be compared with theoretical predictions. The FE model for flat panels was first used to investigate the effects of various parameters and thus to decide the final model. A convergence study was undertaken to justify the adequacy of the mesh used. The minimum buckling stress was obtained by varying the model length  $a/2$ . Figure 5b shows the buckling mode of the half-wave buckle model.

The results showed that a model using  $b/2 = 300$  or more and  $t_c = 75$  mm or more (SL grade) gave a wrinkling stress of 87 MPa that lied within 1% of the theoretical prediction of 86.6 MPa, thus matching the wrinkling theory assumption of infinitely wide and deep panels. The half-wave length  $a/2$  of 24 mm also agreed well with the theoretical prediction of 23.8 mm. All these results confirm that a half-wave buckle model can be successfully used to model the wrinkling behaviour of sandwich panels. Therefore the lightly profiled panels were also modelled by considering a single half-wave buckle with load and boundary conditions identical to those used for the flat panels. For these panels, it was found that stresses penetrated deeper into the foam core than was the case for flat panels. This had the effect of reducing the wrinkling stress slightly even for 75 mm deep panels (for flat panels this happened for 50 mm panels). Therefore both steel faces (top in end compression and bottom no load) were modelled to simulate more precisely panels used in practice. Figures 6 (a) and (b) show the Satinlined and Ribbed models used in the FEA.

Flexural wrinkling behaviour of lightly profiled panels was investigated in detail using the same procedure used for flat panels. The minimum buckling stress was obtained by varying the model length  $a/2$ . Full-scale experiments and theoretical predictions indicated that rib/ridge height has a large influence on the wrinkling stress of lightly profiled panels, and therefore this was investigated using FEA. Table 2 shows the FEA results for varying rib/ridge heights (0.4 to 1.25 mm) for both ribbed and satinlined panels with 0.6 mm steel thickness. For satinlined panels, some experimental results did not agree with theoretical predictions (Eqn.4). However, the FEA results compare well with the theoretical predictions with a 2% maximum difference for ridge heights up to 1 mm. For increasing ridge heights of thinner steel faces (0.4 mm), the theory over-predicted the wrinkling stresses. Both FEA and theoretical results indicate that the wrinkling stress increases are noticeable for small increments of rib/ridge height. Figure 7a illustrates the typical buckling mode of satinlined panels. It is very similar in form to the buckled shape of flat panels.

In contrast, the typical buckling mode of ribbed panels modelled shown in Figure 7b is quite different to those of flat and satinlined panels. Local buckling across the panel width can be seen in the regions where the compressive face has steel elements that are 78.5 mm in length (see Figure 1). This local buckling phenomenon has thus contributed to the significant difference between the FEA and theoretical results reported in Table 1. For ribbed panels with 0.6 mm steel and SL grade foam, the difference in results varied from 7.4 to 33.6%. However, the FEA results agreed reasonably well with the experimental results, for example, for Panel 1, the FEA and experimental stresses were 99.8 and 98.6 MPa.

A theoretical local buckling stress of 98 MPa was calculated using the method recommended by Davies and Hakmi (1990,1992) for ribbed panels with 0.4 mm steel and SL grade foam. This agrees well with the corresponding FEA result of 91.2 to 97.1 MPa for varying rib heights of 0.4 to 1.25 mm. However, the theoretical local buckling stresses of 127.8 and 115.9 MPa for the case of 0.6 and 0.5 mm steel faces do not agree with the FEA predictions of 92.8-110.5 and 92.5-104.2 MPa. Therefore it can be concluded that for ribbed panels with steel thickness of 0.5 mm or more, a combined mechanism of wrinkling and local buckling contributes to failure. This explains the differences between the theoretical predictions and FEA results, and the lower than expected failure stresses for ribbed panels. The ribbed section geometry considered here can be improved if the slender plate elements (78.5 mm) are eliminated. ECCS (2000) suggests that the  $b/t$  ratio of plate elements should be less than 100 for the wrinkling formula for lightly profiled faces to be used (Eqn.4).

In order to model the gap and/or step imperfections due to the presence of transverse joints (see Figure 3), a larger model was chosen as the half-wave buckle model could not be used. In this case, a longer length  $\ell$  was chosen to include the joint at mid-length while being able to accommodate sufficient multiple buckles. As for the half-wave buckle model for lightly profiled panels, both steel faces were included. Half the width ( $b/2$ ) was modelled with appropriate boundary conditions. Figure 8a shows the model used in the case of flat panels. The panel length  $\ell$  of 300 mm was found to be adequate to obtain accurate results (Figure 8b). Even though the model was large, resulting in a slightly coarser mesh density, results were still found to be adequate.



The gap imperfection was studied first using the model shown in Figure 8a with the gap placed at mid-length. The model dimensions used were:  $b/2 = 300$  mm,  $\ell = 300$  mm and foam thickness  $t_c = 75$  mm. The gap was modelled by leaving a physical gap between the solid elements representing the foam slabs on either side of the gap. It was also modelled with solid elements of very low stiffness and the results were identical. Effects of the gap size on the wrinkling stress were studied by analysing the model with varying physical gaps.

The results in Figure 9a show that when a discontinuity in the form of a transverse joint exists in the foam core of a sandwich panel, a significant reduction in wrinkling stress (14 to 16%) occurred even when the gap was very small. With increasing gap size from 0.1 to 10 mm, the reduction was very gradual after the initial significant reduction. This observation was the same and magnitudes were similar for steel faces of varying thicknesses (0.4 to 0.6 mm), and foam of varying grade (SL and M) and thickness (50 to 200 mm). Examination of the buckling mode of panels containing a gap shows that buckling becomes concentrated at the gap location. The location of gap within the panel span was considered another important parameter, but its effects on the wrinkling stress were minimal as observed in the experiments.

Gap imperfections for satinlined models were modelled in a similar manner as for flat panels described above. The immediate reduction to wrinkling strength due to gap imperfections was of a very similar magnitude (14 to 16%) as for flat panels. As the gap size increases from 0.1 mm to 10 mm, the reduction was only gradual.

In order to verify whether the panel length chosen had any effect on the wrinkling stress results, it was also varied from 300 to 600 and 1000 mm, but the results confirmed the observations made using the 300 mm length model. Models containing step imperfections were then created. They had the same geometry, mesh and boundary conditions as for the models with gap imperfections. Since the step size was not expected to be greater than 1 mm in practice, it was used in the model with varying gap widths of 1, 5 and 10 mm. Initial task was to study only the effect of step imperfection, and hence both the steel face and foam were modelled as continuous, but with a step in the steel face geometry and a gap in the foam core with solid elements. The results showed that there was little change to the wrinkling stress and the mode of buckling due to the presence of step imperfections when compared with the results of flat and satinlined panels with no gap/step imperfections. This study was continued with gap and step imperfections, which showed that the wrinkling stresses were reduced by similar magnitudes observed with the models with only the gap imperfections. These results demonstrate that gap imperfections rather than step imperfections is the major contributing factor to the loss of wrinkling strength when transverse joints are present in a sandwich panel.

The model used in the FEA and the theoretical methods of Davies et al. (1991) assumed infinitely wide and deep panels. The sandwich panels used as claddings in buildings are considerably wide and have foam thicknesses greater than or equal to 50 mm with another steel face in tension. Therefore it is expected that the theoretical predictions from Eqns. 2 and 4 and FEA models used so far could adequately predict the wrinkling stress of these sandwich panels. However, these panels are in fact subjected to uniform lateral wind pressures rather than end compression loads.

Therefore it may be necessary to investigate the wrinkling behaviour of sandwich panels subjected to a uniform lateral pressure loading.

For this purpose, a finite element model shown in Figure 10 was used with a finer mesh at midspan region. Both steel faces and the foam core were modelled with appropriate boundary conditions. The model length  $L/2$  and width  $b/2$  were chosen as 1400 mm and 600 mm, respectively (simulates standard test panel sizes) whereas the foam thickness  $t_c$  and steel face thickness  $t_f$  were taken as 200 mm and 0.6 mm, respectively. The wrinkling stress determined from this model was 90.3 MPa. The slight difference in wrinkling stress compared with earlier results from FEA and theoretical prediction of 87.1 and 86.6 MPa might have been due to the difference in loading (lateral pressure loading versus end compression loads) and the coarser mesh used. However, the difference is small and thus continued use of theoretical formulae (Eqn.2) and FEA using simplified models (Figures 5 and 6) is satisfactory in determining the wrinkling stress of sandwich panels as used in practice. The FEA result of 90.3 MPa compares well with the average wrinkling stress of 92.4 MPa from experimental results for flat panels with SL grade foam and 0.6 mm steel face.

The same half-length model subjected to a uniform lateral pressure was also used for ribbed and satinlined panels without transverse joints. The results are very similar to those obtained from a half-wave buckle length model subject to end compression load and theoretical predictions based on an energy method (Eqn.4). They confirmed the increase in wrinkling stress for satinlined panels, and the occurrence of local buckling and associated reduction in buckling stresses for ribbed panels.

In order to study the effects of gap and step imperfections, a full-length model was used, but with other details remaining the same as for the half-length panels without joints as shown in Figure 10. Appropriate boundary conditions were used to enable panel length ( $L$ ) and width ( $b/2$ ) to be modelled. A finer mesh was used at midspan as for half-length models. The full-length model was first validated by comparing the wrinkling stress from this model without a gap or step. The result of 92.4 MPa compared well with that of the half-length model of 90.3 MPa. Despite the coarser density of the mesh used, the results are similar to that obtained from the half-wave buckle length model of 87.1 MPa. Following this, the effect of gap and/or step imperfections was investigated using the full-length model with the gap at mid-length for flat panels. As for the models based on end compression loads, an immediate reduction to wrinkling strength occurred due to the presence of a gap imperfection. However, this reduction was in the range of 22-24% for steel faces with 0.4 to 0.6 mm thicknesses, compared with 14-16% obtained from the models with end compression loads. The difference in loading could have been the reason for the difference in the results for the effect of joints. However, the higher strength reduction predicted by the full length models seem to agree with those obtained in the experiments on panels with full-width butt joints with an average reduction value of 20%. Figure 9b shows the reduction to wrinkling stress as a function of gap size for full-length models.

The full length models were also used to study the effect of step imperfections of up to 1 mm. They confirmed the previous result from the models with end compression loads that step imperfections alone had no effect on wrinkling stress. Full-length models of lightly profiled panels were not used to study the effect of step and gap

imperfections as other results obtained indicate that the reductions observed for flat panels will apply to lightly profiled panels.

## **Conclusions and Recommendations**

The following conclusions and recommendations can be drawn from this study which included extensive experimental and numerical investigations of the wrinkling behaviour of lightly profiled sandwich panels:

- For satinlined panels without joints, theoretical and FEA results are in good agreement, and Equation 3 can be used to predict the wrinkling stress. When the ridge height exceeds 1 mm, results from FEA need to be considered. As for flat panels, the use of characteristic or guaranteed minimum material properties  $E_c$ ,  $G_c$  and a reduction factor of 0.8 to allow for other imperfections are recommended. However, further experimental tests are recommended for satinlined panels without joints to ensure that the theoretical wrinkling stress is achieved. The specified ridge height must be attained during panel production. Until this is resolved, the use of current design equation (ECCS, 2000) is recommended.
- Both FEA and experiments showed that when a discontinuity in the form of a full-width transverse joint was present in the foam core anywhere within the panel span, a significant reduction in wrinkling stress occurred even when the ‘gap’ imperfection was very small (0.1 mm). With increasing gap size from 0.1 to 10 mm, the reduction was gradual. The FEA showed that ‘step’ imperfections did not cause any reduction to wrinkling capacity. Therefore, based on the FEA results for panels subjected to a uniform lateral pressure load, a further reduction factor of 0.76 is recommended to Equation 3 to allow for the ‘gap’ and ‘step’

imperfections. The magnitudes of ‘gap’ and ‘step’ imperfections created during panel production should be kept to a minimum. It must be noted that this investigation did not consider the weakening effects due to shear and interaction of shear and wrinkling, and was limited to simple spans. The sandwich panel manufacturers should consider using the recommendation in ECCS (2000) that states, “no joint should extend across more than one quarter of the panel width”. The joints should not be located near high shear and/or bending regions.

- The ribbed panels failed at stresses lower than that predicted by the wrinkling formula because of local buckling. The current ribbed profile should therefore be designed for local buckling effects in combination with FEA results and not based on the wrinkling formula for lightly profiled panels. The current ribbed profile does not meet the ECCS (2000) requirement of  $b/t < 100$  for flat plates of lightly profiled panels. If it is appropriately modified, the ribbed panels can also provide considerable improvement to flexural wrinkling strength over flat panels.
- The results have demonstrated the potential improvement to flexural wrinkling strength due to satinlined and ribbed panels over flat panels provided the profile geometry has been optimally designed and that the design profile is achieved in practice. The manufacturers should adopt quality control procedures to guarantee not only the profile geometry, but also the material properties of foam ( $E_c$ ,  $G_c$ ).
- This study has shown that the theoretical equations for flat and lightly profiled panels based on end compression loads adequately predict the wrinkling stress of panels subjected to uniform lateral pressures.

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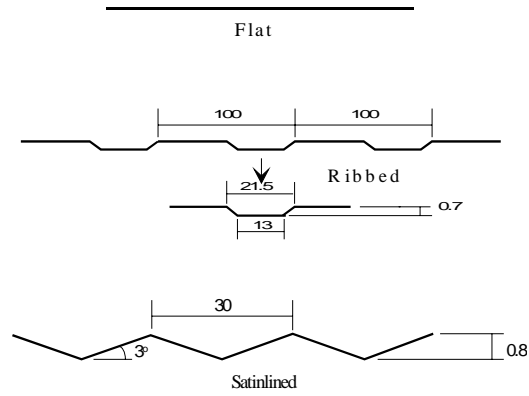
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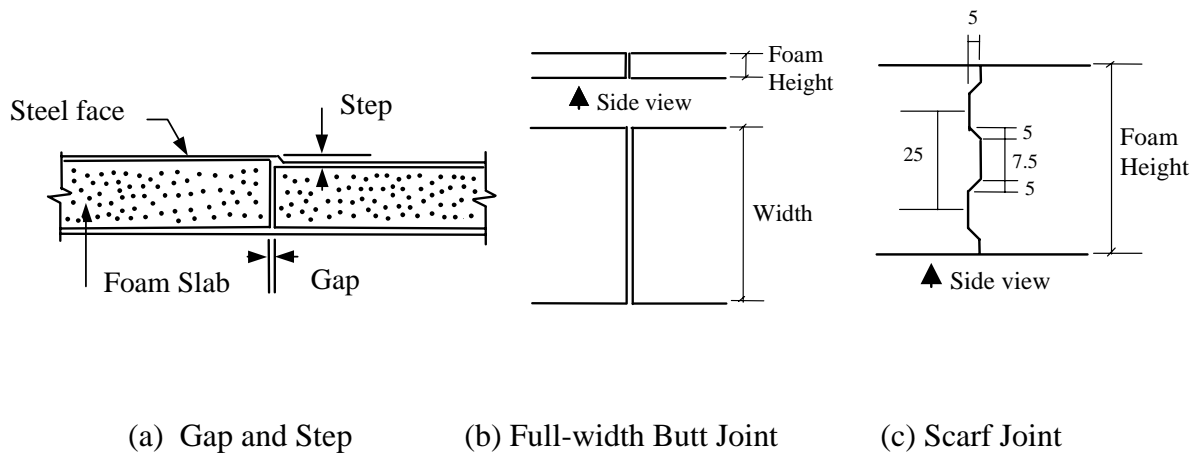




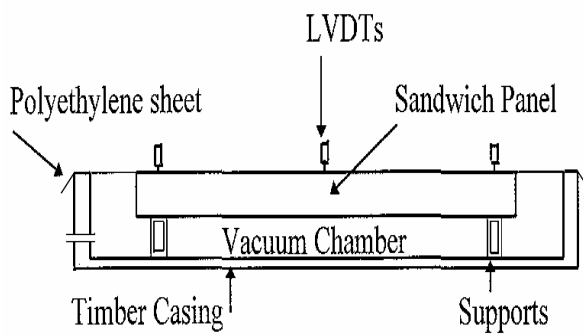
**Figure 1. Face Profiles of Flat and Lightly Profiled Sandwich Panels**



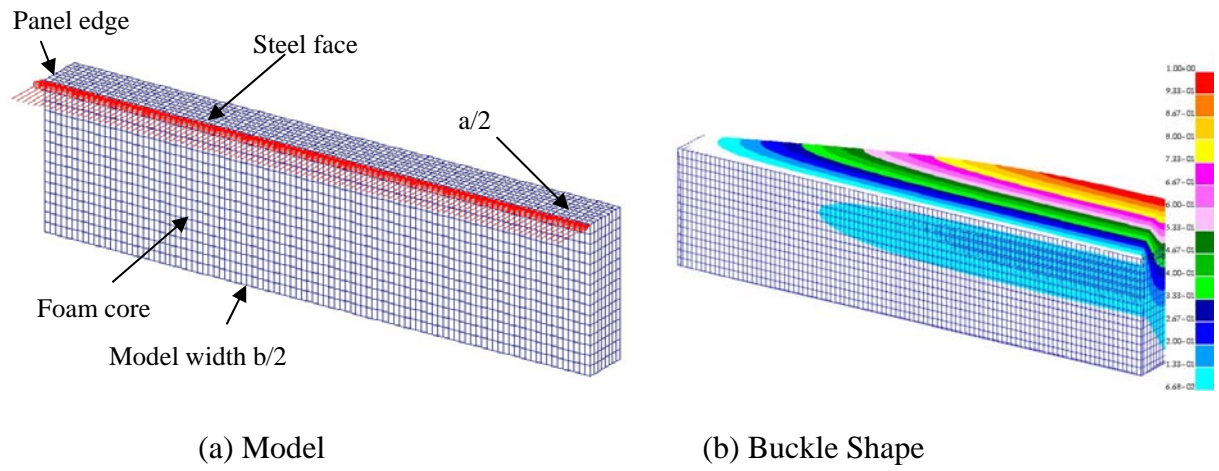
**Figure 2. Flexural Wrinkling Failure**



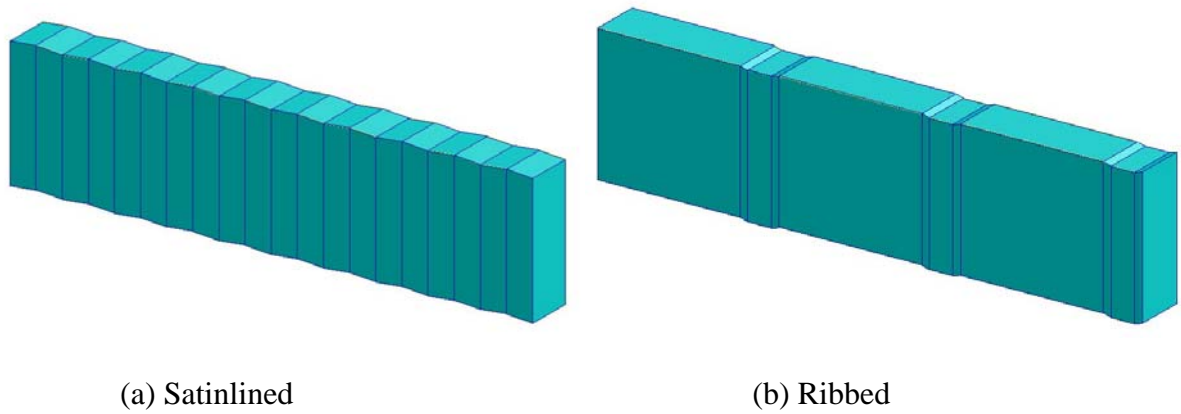
**Figure 3. Transverse Joints**



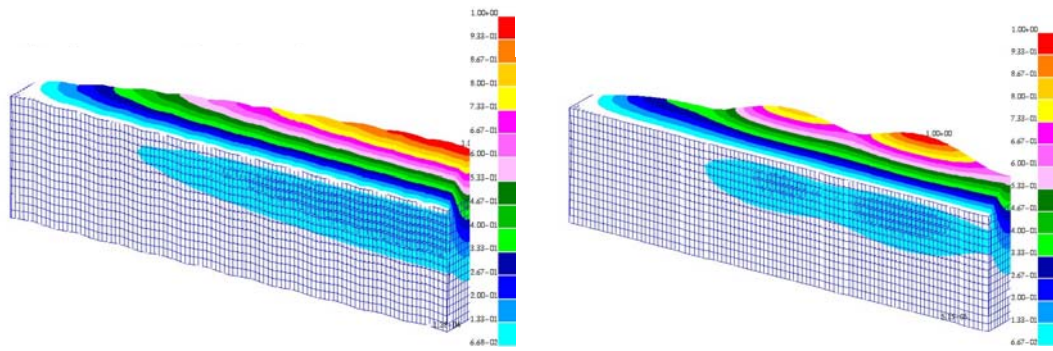
**Figure 4. Experimental Set-up**



**Figure 5. Half-wave Buckle Model of Flat Panels**



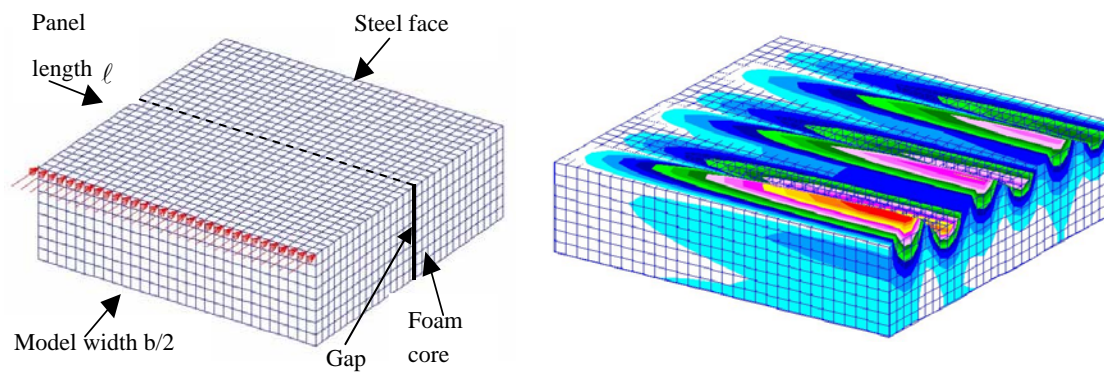
**Figure 6. Lightly Profiled Panel Models**



(a) Satinlined

(b) Ribbed

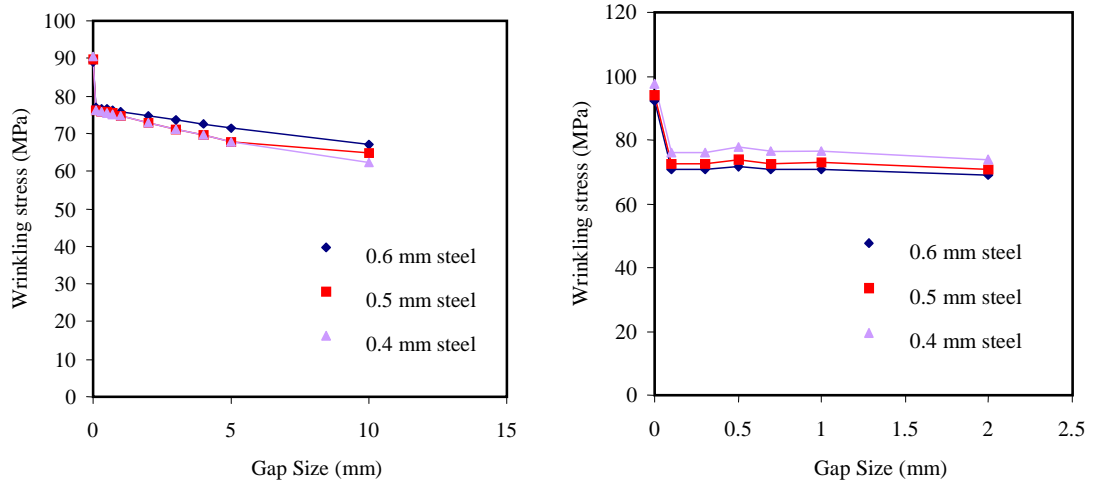
**Figure 7. Buckled Shape of Lightly Profiled Panels**



(a) Model

(b) Buckle Shape

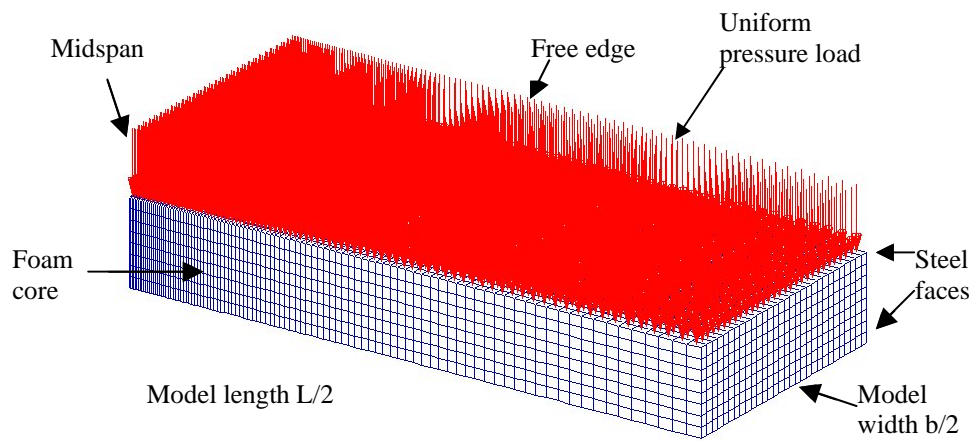
**Figure 8. Finite Element Model including the Gap Imperfection**



(a) Simple Models

(b) Full Length Models

**Figure 9. Effect of Gap Size on Wrinkling Stress**



**Figure 10. Model of a Sandwich Panel without Joints**

**Table 1. Details of Test Panels and Results**

Panel Details <sup>1</sup>	Joint Details <sup>2</sup>	Failure Press. p (kPa)	Wrinkling Stress (MPa)		
			Expt.	Theory	Design <sup>3</sup>
1/ 0.6R/ 75SL/ 2700	No	4.93	98.6	122.7	59.5
2/ 0.6R/ 75SL/ 2700	Butt/ 0/ 0.4/ 510	3.96	79.6	122.7	59.5
3/ 0.6R/ 75SL/ 2700	Butt/ 0/ 0.5/ 25	4.01	80.4	122.7	59.5
4/ 0.6S/ 75SL/ 2700	No	4.88	97.2	118.1	59.5
5/ 0.6R/200M/ 4700	No	4.22	96.4	165.7	85.4
6/ 0.6S/150SL/ 4660	No	3.70	110.5	118.1	59.5
7/ 0.6S/200M/ 4700	No	6.33	143.0	159.6	85.4
8/ 0.4S/ 75M/ 3700	Scarf/ 0/ 0.2/ 120	2.51	140.3	194.2	93.2
9/ 0.4S/ 75M/ 3700	Butt/ 0.5/ 0.1/ 55	2.65	147.7	194.2	93.2
10/ 0.4S/ 75M/ 3700	Butt/ 0/ 0.15/ 480	3.03	168.5	194.2	93.2
11/ 0.4S/ 75SL/ 3700	Scarf/ 0/ 0.4/ 910	1.46	83.1	143.8	65.0
12/ 0.4S/ 75SL/ 3700	Scarf/ 0/ 0.15/ 0	1.43	81.4	143.8	65.0
13/ 0.4S/ 75SL/ 3700	Scarf/0/0.35/1030	1.86	104.8	143.8	65.0
14/ 0.4S/ 75SL/ 3700	Scarf/ 0/ 0.2/ 370	1.72	97.2	143.8	65.0
15/ 0.5S/ 75SL/ 3000	Scarf/0.2/0.2 /315	2.51	74.6	128.5	59.5
16/ 0.5S/ 75SL/ 3000	Scarf/ 0/ 0.3/370	2.40	71.5	128.5	59.5
17/ 0.5S/ 75SL/ 3000	Butt/0.1/ 0.1/1060	1.74	52.6	128.5	59.5
18/ 0.5S/ 75SL/ 3000	No	2.96	87.6	128.5	59.5
19/ 0.5S/ 75M/ 3700	Butt/ 0/ 0.2/ 70	2.47	112.0	173.5	85.4
20/ 0.5S/ 75M/ 3700	Butt/ 0/ 0.15/1060	2.56	115.9	173.5	85.4
21/ 0.5S/ 75M/ 3700	Scarf/ 0/ 0/1050	2.80	126.4	173.5	85.4
22/ 0.5S/ 75M/ 3700	Scarf/ 0/ 0/ 10	3.53	158.3	173.5	85.4
23/ 0.5S/ 75M/ 3700	No	3.69	165.3	173.5	85.4
24/ 0.5S/ 75SL/ 3700	Butt/ 0/ 1.0/1040	2.07	94.4	128.5	59.5
25/ 0.4S/ 75M/ 3700	Scarf/ 0/ 0.4/ 990	1.84	104.0	194.2	93.2
26/ 0.4S/ 75M/ 3700	No	2.63	147.0	194.2	93.2
27/ 0.5S/ 75SL/ 3000	Butt/ 0/ 0.2/ 0	3.80	111.9	128.5	59.5
28/ 0.5S/ 75SL/ 3000	Scarf/ 0/ 0.1/ 830	3.86	113.0	128.5	59.5

1. Panel details are given as follows: Test Number/ Thickness and type of steel face, R – Ribbed & S – Satinlined/ Thickness and grade of foam/ Span in mm
2. Joint details are given as follows: Type of joint/ Gap size/ Step size/ Joint location from panel midspan in mm.
3. Design stresses = Greater of the values obtained from Equations 3 and 2 using the coefficients of 0.95 and 0.65, respectively.

**Table 2. Effect of Rib/Ridge Height on the Wrinkling Stress (0.6 mm steel)**

Rib/Ridge Height (mm)	$\sigma_{wr}$ Satinlined (MPa)				$\sigma_{wr}$ Ribbed (MPa)			
	SL		M		SL		M	
	Theory	FEA	Theory	FEA	Theory	FEA	Theory	FEA
0.4	95.1	95.7	128.4	128.4	100.6	92.8	135.8	123.6
0.55	103.1	102.5	139.2	136.6	110.9	96.2	149.9	127.1
0.7	-	-	-	-	122.7	99.8	165.7	130.5
0.8	118.1	116.2	159.6	154.9	-	-	-	-
1.0	130.9	128.4	176.8	171.1	146.6	106.4	198.0	136.3
1.25	147.2	144.0	198.9	192.2	166.3	110.5	224.5	140.0