Behaviour and Design of Sandwich Panels
Subject to Local Buckling and Flexural
Wrinkling Effects

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By

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KEYWORDS

Sandwich panels, fully profiled faces, lightly profiled faces, foam core, local buckling, flexural wrinkling, interactive buckling, slender plates, effective width, rib depth, flat plate width, experimental investigation, finite element analysis
ABSTRACT

Sandwich panels comprise a thick, light-weight plastic foam such as polyurethane, polystyrene or mineral wool sandwiched between two relatively thin steel faces. One or both steel faces may be flat, lightly profiled or fully profiled. Until recently sandwich panel construction in Australia has been limited to cold-storage buildings due to the lack of design methods and data. However, in recent times, its use has increased significantly due to their widespread structural applications in building systems. Structural sandwich panels generally used in Australia comprise of polystyrene foam core and thinner (0.42 mm) and high strength (minimum yield stress of 550 MPa and reduced ductility) steel faces bonded together using separate adhesives. Sandwich panels exhibit various types of buckling behaviour depending on the types of faces used. Three types of buckling modes can be observed which are local buckling of plate elements of fully profiled faces, flexural wrinkling of flat and lightly profiled faces and mixed mode buckling of lightly profiled faces due to the interaction of local buckling and flexural wrinkling. To study the structural performance and develop appropriate design rules for sandwich panels, all these buckling failure modes have to be investigated thoroughly. A well established analytical solution exists for the design of flat faced sandwich panels, however, the design solutions for local buckling of fully profiled sandwich panels and mixed mode buckling of lightly profiled sandwich panels are not adequate. Therefore an extensive research program was undertaken to investigate the local buckling behaviour of fully profiled sandwich panels and the mixed mode buckling behaviour of lightly profiled sandwich panels.

The first phase of this research was based on a series of laboratory experiments and numerical analyses of 50 foam-supported steel plate elements to study the local buckling behaviour of fully profiled sandwich panels made of thin steel faces and polystyrene foam core covering a wide range of b/t ratios. The current European design standard recommends the use of a modified effective width approach to include the local buckling effects in design. However, the experimental and numerical results revealed that this design method can predict reasonable strength for sandwich panels with low b/t ratios (< 100), but it predicts unconservative strengths for panels with slender plates (high b/t ratios). The use of sandwich panels with high
*b/t* ratios is very common in practical design due to the increasing use of thinner and high strength steel plates. Therefore an improved design rule was developed based on the numerical results that can be used for fully profiled sandwich panels with any practical *b/t* ratio up to 600. The new improved design rule was validated using six full-scale experiments of profiled sandwich panels and hence can be used to develop safe and economical design solutions.

The second phase of this research was based on a series of laboratory experiments and numerical analyses on lightly profiled sandwich panels to study the mixed mode buckling behaviour due to the interaction of local buckling and flexural wrinkling. The current wrinkling formula, which is a simple modification of the methods utilized for flat panels, does not consider the possible interaction between these two buckling modes. As the rib depth and width of flat plates between the ribs increase, flat plate buckling can occur leading to the failure of the entire panel due to the interaction between local buckling and wrinkling modes. Experimental and numerical results from this research confirmed that the current wrinkling formula for lightly profiled sandwich panels based on the elastic half-space method is inadequate in its present form. Hence an improved equation was developed based on validated finite element analysis results to take into account the interaction of the two buckling modes. This new interactive buckling formula can be used to determine the true value of interactive buckling stress for safe and economical design of lightly profiled sandwich panels.

This thesis presents the details of experimental investigations and finite element analyses conducted to study the local buckling behaviour of fully profiled sandwich panels and the mixed mode buckling behaviour of lightly profiled sandwich panels. It includes development and validation of suitable numerical and experimental models, and the results. Current design rules are reviewed and new improved design rules are developed based on the results from this research.
(a) International Refereed Journal Papers


(b) International Refereed Conference Papers


(c) Invited Paper for International Conference


(d) Non-Refereed Publications


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### NOTATIONS

- $a$: half-wave buckle length
- $A_f$: cross-sectional area of the face per unit width, cross-sectional area
- $b$: width of flat plate or width of panel
- $b_{eff}$: effective width of flat plate element
- $b_{eff,i}$: effective width of element $i$
- $b_i$: width of plate element $i$
- $b_s$: horizontal width of inclined element
- $B$: overall panel width
- $B_{eff}$: bending stiffness of an equivalent column
- $B_f$: flexural rigidity of the face per unit width
- $B_T$: bending stiffness
- $C$: non-dimensional parameter
- $C_f$: foundation coefficient
- $D$: flexural rigidity
- $D_c$: flexural rigidity of the core
- $D_f$: flexural rigidity of the faces
- $e$: distance between centroid of steel faces
- $E_c$: Young’s modulus of foam
- $E_f$: Young’s modulus of steel
- $f_{Cc}$: characteristic compressive stress
- $f_{Cv}$: characteristic shear strength of the core
- $f_{Fc}$: compression strength of profiled face
- $f_y$: yield stress of steel
- $F$: support reaction
- $g$: self weight of the panel
- $G_c$: shear modulus of foam core
- $G_{ct}$: shear modulus of the core for long term loading
- $h$: depth of the equivalent column, depth of the core
- $h_i(z)$: functions
- $I$: second moment of area
- $I_{eff}$: effective second moment of area
- $k_{mn}$: a parameter
- $K$: numerical constant, local buckling coefficient
- $K_{eff}$: effective buckling coefficient of equivalent column
- $K_{inc}$: increased value of buckling coefficient
- $K_p$: numerical constant
- $L$: length or span of the sandwich panel
- $L_s$: support width
- $m, n$: numbers of terms
- $M$: bending moment
- $M_f$: bending moment in the lightly profiled face
- $M_a$: applied mid-span bending moment
- $N_1, n_2$: numbers of the webs
- $p$: applied stress
- $q$: uniformly distributed load
- $R$: non-dimensional stiffness parameter, support reaction
- $R_G$: degree of reflection relative to magnesium oxide = 100%
\( s_{w1}, s_{w2} \) lengths of the webs of the profiled faces
\( S \) shear stiffness
\( t \) thickness of steel
\( t_1, t_2 \) thicknesses of the faces
\( t_c \) thickness of core
\( T_1 \) outside temperature
\( T_2 \) inside temperature
\( u, v, w \) displacements
\( U \) total potential energy
\( U_B \) strain energy of bending in the plate
\( U_C \) strain energy in the core
\( V \) work done by applied compressive force
\( V_{F1}, V_{F2} \) shear forces in faces 1 and 2
\( w_b \) deformation due to bending
\( w_s \) deformation due to shear
\( w_u \) wind load per unit length
\( y \) distance from the neutral axis
\( y_{max} \) distance of centroid from extreme fibre
\( \alpha \) empirical factor whose value is greater than 1
\( \beta \) non-dimensional parameter
\( \delta \) constant
\( \varepsilon \) strain
\( \phi \) ratio of half-wave buckle length to plate width
\( \phi(t), \phi_i \) creep coefficient
\( \gamma \) non-dimensional parameter
\( \gamma_m \) material factor
\( \varphi \) constant
\( \lambda \) slenderness parameter
\( \mu \) constant
\( \nu_c \) Poisson’s ratio of polystyrene foam
\( \nu_f \) Poisson’s ratio of steel
\( \rho \) ratio of effective width to actual width of the plate
\( \sigma \) stress
\( \sigma_c \) bending stress in the core, compressive stress in core
\( \sigma_{Ccd} \) compressive stress in the core over a support
\( \sigma_{cr} \) critical buckling stress
\( \sigma_{cr,i} \) buckling stress of the face \( i \)
\( \sigma_f \) bending stress in the face
\( \sigma_{f(max)} \) maximum face stress
\( \sigma_{f(midspan)} \) mid-span face stress
\( \sigma_{Fd} \) tensile or compressive stress in the face
\( \sigma_k \) buckling stress of equivalent column averaged over full panel width
\( \sigma_{wr} \) flexural wrinkling stress, interactive buckling stress
\( \tau_{Cd} \) shear stress in the core
\( \tau_{Fdi} \) shear stress in the web of a profiled face layer
\( \psi \) ratio of the stresses at the ends of the inclined plate element
The work contained in this thesis has not been previously submitted for a degree or diploma at any other higher education institution. To the best of my knowledge and belief, the thesis contains no material previously published or written by another person except where due reference is made.

Signed: 

Date:
CHAPTER 1.0 INTRODUCTION

1.1 Background

Sandwich panels are becoming more and more popular in the construction industry due to their widespread structural applications in both commercial and residential building systems. Structural sandwich panels consist of two thin, stiff and strong sheets of dense material separated by and bonded rigidly to the centre core of lighter, weaker, less stiff and low-density material. The two thin sheets are usually called ‘faces’ and the inner thick layer is called ‘core’. Faces are commonly made of steel and the core material may be polyurethane, polyisocyanurate, expanded polystyrene, extruded polystyrene, phenolic resin, or mineral wool.

The use of sandwich panels offers many advantages as it leads to structures that are lightweight, cost effective and durable. In the past, sandwich panels have been commonly used in many aeronautical applications. In recent years, they have been used as structural building components in many industrial and office buildings in Europe and the USA. Their use has now been extended to residential building construction due to their ability to improve the structural and thermal performance of houses. Until recently, sandwich panel construction in Australia has been limited to cold-storage buildings due to lack of design methods and data. However, in recent times, the sandwich panels have been increasingly used in building structures (see Figure 1.1), particularly as roof and wall cladding systems. They are also being used significantly as internal walls and ceilings.

There has been growing interest in the use of sandwich panels in the construction of building and other structural systems due to their wide range of advantages. Some of the advantages such as high strength-to-weight ratio, reduction in cost of framework and foundation, and superior structural efficiency make the sandwich panel an excellent load-carrying member in building. These panels are very efficient for thermal and sound insulation. Further, they possess many other advantages such as long spanning capability, mass productivity, transportability, fast erectability, prefabricatability, durability, and reusability. These characteristics render such panels
very useful even in an environment subject to extreme temperatures, in places where erection time and labour need to be minimized, and in many other difficult situations.

(a) Woolworth State Office, South Australia

(b) Sandwich Panel (Subaaharan, 1998)

Figure 1.1 Sandwich Panels in Buildings

This figure is not available online. Please consult the hardcopy thesis available from the QUT Library.
Sandwich panels can be manufactured using different processes. Some of the commonly used processes (Davies, 1987a) are:

- **Continuous process:** In this method, foam is continuously injected to the mould made of steel faces in an arrangement creating all necessary environments. The foam filled continuous mould is then cut into the required length. This manufacturing process has the capacity to produce large quantities of panels of consistent quality. But very few continuous foaming lines are in operation because of the high cost involved in the process.

- **Individual production in horizontal mould:** The moulds are positioned horizontally so that injected foam falls on to the lower face and rises to meet the upper face. The drawback of this process is that it may result in different degrees of adhesion to the two faces and non-uniform foam properties. Further, the foam may spread and rise unevenly, trapping small pockets of air and giving rise to an increased susceptibility to blistering.

- **Individual production in vertical mould:** Moulds are positioned vertically and the foam rises in contact with both faces. This process of manufacturing can overcome some of the problems associated with horizontal moulds.

- **Separate production of cores from face:** Foam cores are produced separately from the faces and the three components (two faces and core) are glued together by the use of adhesive to produce the complete panel. This method is considered to be more economical.

In Australia, sandwich panels are generally produced by the last method described above. Expanded polystyrene foam cores are manufactured separately and the high strength steel faces are attached to the foam core by applying a suitable adhesive (see Figures 1.2 (a) and (b)). Although this is the easiest and most economical process, greater attention should be given to achieve the perfect bonding between faces and core. For a satisfactory structural performance and better composite action, adhesive bond plays a vital role.
In the building industry, the steel faces of sandwich panels are generally used in three forms: flat, lightly profiled, and profiled. Typical cross-sections of sandwich panels with three different forms of faces are shown in Figure 1.3. Lightly profiled sandwich panels are considered to be those with rib depth of less than 2 mm. Flat and lightly profiled panels are generally used as wall claddings. For roof construction, it
is usually recommended that at least one face be profiled in order to constrain the tendency of most core materials to creep under sustained load (Davies, 1993).

![Figure 1.3 Sandwich Panels with Various Faces](image)

The faces of sandwich panels serve various purposes. They provide architectural appearance, structural stiffness, and protect the relatively vulnerable core material against damage or weathering. Tensile and compressive forces are supported almost entirely by faces. Flat and lightly profiled faces can carry only axial forces, as their bending stiffness is negligible whereas profiled faces can carry both axial forces and bending moments. Similarly, the core of the sandwich panel has many functions. Essentially, it keeps the faces apart and stabilizes them against local failure, and provides a shear connection between faces. Further, it increases the moment of inertia of the cross-section of the outer faces and enables them to work together as a single beam. Both the critical buckling stress and the ultimate compressive strength of the compressed face are significantly raised by the presence of the core in comparison to similar plates without such support. Hence, the sandwich panel represents an excellent example of the optimum use of dissimilar materials.

Under the action of different loadings such as gravity, wind, snow, temperature gradient, and others, the steel plate elements of sandwich panels are susceptible to various buckling failures due to axial compression and/or bending actions depending on the type of steel faces used. Essentially, three types of buckling behaviour can be observed which are local buckling of steel plate elements of fully profiled sandwich panels, flexural wrinkling of flat and lightly profiled sandwich panels, and mixed
mode buckling of lightly profiled sandwich panels. A thorough investigation of all these buckling behaviours is the important and essential step for the accurate analysis and safe design of sandwich panels in any structural system.

The static behaviour and strength of sandwich panels is based on the composite action of the three structural layers, namely the two faces and the core (Davies, 2001). Various factors such as influence of shear flexibility of the core on the global behaviour, influence of the core in restraining local buckling of the faces, influence of temperature-induced stresses and deflections, creep and so on make the design of structural sandwich panels different from other light-weight building components (Davies, 1993). A large number of studies have been undertaken in sandwich construction to investigate their buckling behaviour and develop rational design procedures. As a result of extensive research in the past few decades especially in Europe, a design document called “European Recommendations for Sandwich Panels, Part 1: Design” (CIB, 2000) has been developed for the design of sandwich panels with metal faces and various types of foam core. This has enabled the designer and builders to use sandwich panels safely in many structural systems.

However, this design document is mainly based on the metal faces and polyurethane or polyisocyanurate cores and not intended for very thin and high strength steels. In contrast to the European practice of using thicker steel faces and polyurethane foam core, Australian sandwich panels are comprised of very thin and high strength steel faces (0.42 mm thickness and minimum yield stress of 550 MPa) and expanded polystyrene foam core which are bonded together. Hence there is a need to verify the applicability of European recommendations to Australian panels in order to develop the necessary confidence among Australian manufacturers and designers. Despite the increasing interest in the use of sandwich panels, the lack of design rules and standards has confined their use to specific areas. This clearly indicates the need for research to investigate all types of buckling behaviour of such composite panels, examine the current design procedures and develop new design rules in order to overcome the inconveniences faced by the designers and manufacturers.
1.2 Problem Definition

In the past, many research programs were conducted on various aspects of the behaviour of sandwich panels. Despite these studies, there are several major issues, which have not been addressed adequately in any of the research conducted to date. In summary, the following issues have influenced the selection of the proposed research area.

Local buckling (see Figure 1.4) of plate elements in fully profiled sandwich panels is significantly improved by the presence of the foam core. However, fully profiled sandwich panels are always susceptible to local buckling failure under the action of compression, bending or a combination of these loading actions. This local buckling phenomenon is treated in design by utilizing the modified conventional effective width approach (CIB 2000), originally developed for the plain plate elements without any foam core. But recent research (Jeevaharan, 1997) has shown that the modified effective width approach can not be extended to the fully profiled sandwich panels with increasing plate slenderness. The effective width approach developed by Winter (1947) was based on many tests and studies of local buckling and postbuckling strengths of steel plates. This implies that this method can be applied to sandwich panels that have plate elements with low plate width to thickness (b/t) ratios (<200) as they exhibit considerable postbuckling strength. With the increasing b/t ratio of the foam-supported steel plates, there is either very little or no postbuckling strength. Therefore the extension of the conventional effective width method to sandwich panels with slender plates can not represent the true ultimate strength behaviour.
If the plate elements in sandwich panel have very high $b/t$ ratios, the strength will be governed by wrinkling failure and can be evaluated using the well-established wrinkling formula (CIB, 2000). However, the $b/t$ ratios of steel plate elements generally used in fully profiled sandwich panels do not fall in the wrinkling region. Therefore, the wrinkling formula cannot be applied to the plate elements in the profiled sandwich panels as it will underestimate the ultimate strength. Davies and Hakmi (1990) proposed this current modified effective width formula based on limited number of tests that did not include a wider range of plate $b/t$ ratios. Jeevarahan (1997) conducted compression tests to investigate the problem associated with this design formula. Unfortunately, his experimental results were inadequate to eliminate the inadequacy of the design formula and associated problems. Thus, extensive experimental and analytical studies need to be undertaken to investigate thoroughly the local buckling phenomenon of foam-supported steel plate elements used in fully profiled sandwich panels with a wider range of $b/t$ ratios and to modify or develop new design rules that can be applied for any practical sandwich panels used in any structural systems with improved accuracy.

**Figure 1.5 Flexural Wrinkling of Sandwich Panels (McAndrew, 1999)**

For the flat and lightly profiled sandwich panels, flexural wrinkling (see Figure 1.5) is an extremely important design criterion as the behaviour of these panels is governed mainly by flexural wrinkling. A well-established analytical solution based on elastic half-space method exists for the design of flat panels; however, the
analytical solution for the wrinkling of lightly profiled panels is less well developed. The current design method (CIB, 2000) for the wrinkling of lightly profiled panels is based on simple modifications of the method utilized for flat panels to take into account the flexural stiffness of the lightly profiled faces. But recent work (McAndrew, 1999) has identified the inaccuracy of extending this theoretical model to the lightly profiled sandwich panels.

In lightly profiled sandwich panels, as the depth or spacing of the ribs/profiles increases, flat plate buckling between the ribs can occur leading to the failure of the entire panel due to the interaction between local buckling and wrinkling modes (see Figure 1.6). However, the current wrinkling formula is based on the assumption of pure wrinkling of lightly profiled panels and does not consider the interaction between the two buckling modes. To obtain a safe design solution, this mixed mode type buckling failure behaviour should be taken into account in the design of lightly profiled sandwich panels. CIB (2000) recommends that the current design formula for lightly profiled panels is applicable only when the $b/t$ ratio of flat plates between the ribs is less than 100. But the current ribbed profile generally used in Australia does not meet this requirement. As McAndrew (1999) did not investigate the mixed mode type failure of lightly profiled sandwich panels in detail, further research needs
to be conducted using a series of experiments and numerical analyses to investigate the interactive buckling behaviour of lightly profiled panels with varying rib depths and spacings in order to develop a new or improved design formula that will take into account all the practical limitations including interaction between the two buckling modes.

The European Recommendations for Sandwich Panels (CIB, 2000) is the only document available for the design and testing of sandwich panels. This document is usually concerned with sandwich panels having metal faces and polyurethane or polyisocyanurate core materials. But in Australia, sandwich panels are usually comprised of very thin and high strength steel faces and extended polystyrene foam core. In such circumstances, there is a need to verify the applicability of European recommendations to Australian panels in order to develop the confidence among Australian manufacturers and designers. Hence, the research work towards achieving accurate design recommendations for the sandwich panel construction in the Australian environment with a thorough investigation of all the above mentioned buckling modes would be a significant contribution to the construction industry.

1.3 Objectives and Scope of the Research Program

Overall Objective:

The main purpose of this research is to undertake a thorough investigation of the buckling and ultimate strength behaviour of fully profiled and lightly profiled sandwich panels comprised of thinner and high strength steel faces and a comparatively thicker polystyrene foam core, and to develop appropriate design rules and recommendations for the safe design of sandwich panels in any structural systems. Specific objectives of this research are listed next:

Specific Objectives:

- To investigate analytically and experimentally the local buckling behaviour of slender plate elements supported by a polystyrene foam core as used in fully
profiled sandwich panels. On the basis of this study, alternative design formulation for local buckling of foam-supported slender plate elements will be developed.

- To investigate analytically and experimentally the mixed wrinkling and local buckling behaviour of lightly profiled sandwich panels. On the basis of this study, an alternative wrinkling formula that will take into account all the practical limitations including interaction between the wrinkling and local buckling modes will be developed.

- To recommend safe design rules for sandwich construction in the Australian context using the results from detailed investigations of various buckling modes of sandwich panels. This will help to develop the confidence among the designers and manufacturers to use sandwich panels in any structural and building systems with a greater degree of safety.

Only those sandwich panels manufactured from thin steel faces and expanded polystyrene foam cores have been considered in this investigation. In addition to the various buckling failures mentioned earlier, there are also other failure modes such as yielding of metal faces, shear failure, crushing of the panels, failure of the fasteners, blistering etc. which must be considered in the design of sandwich panels. Similarly, various aspects of sandwich panel behaviour under environmental impact, fire and creep loading and durability issues are also important in the design of sandwich panels. However, this research project is limited to the investigation of fully profiled and lightly profiled sandwich panels under local buckling and flexural wrinkling modes. It is believed that CIB (2000) addresses the other design issues reasonably well.

1.4 Method of Investigation

The research was mainly based on a series of laboratory experiments followed by numerical studies. Laboratory experiments included both full-scale and small-scale tests. For the small-scale tests, sandwich panel specimens required for the tests were
prepared in the laboratory, i.e. thin cold-formed steel faces were glued to the polystyrene foam cores with suitable adhesives. Flat, lightly profiled, and profiled faces were used as per the requirements to fulfil the stated objectives. For the full-scale experiments, required sandwich panels were directly ordered from sandwich panel manufacturing companies.

Local buckling investigation of foam supported steel plate elements as used in fully profiled sandwich panels was conducted using compression tests on the Tinius Olsen Testing Machine. In conducting these tests, $b/t$ ratios, thicknesses, steel grades etc. of the foam supported steel plate elements were taken as variable parameters. Full-scale tests on fully profiled sandwich panels were conducted by applying a uniformly distributed wind pressure loading using a specially constructed vacuum test rig. Mixed mode type buckling of lightly profiled sandwich panels was investigated experimentally by compression tests on foam supported lightly profiled steel faces using a Tinius Olsen Testing Machine. In conducting these experiments, the depth and spacing of the ribs, the thickness of steel etc. were taken as variable parameters.

For both the local buckling investigation of fully profiled sandwich panels and mixed mode type buckling investigation of lightly profiled sandwich panels, finite element analyses were carried out using the finite element programs ABAQUS and MSC/PATRAN. ABAQUS was used for numerical computation whereas PATRAN was used as a pre-processor (creation of the models) and a post-processor (visualization of the results). Available theoretical solutions were also evaluated. Experimental results and available theoretical solutions were used as the bench mark data to calibrate the finite element models created and analysed in this study. Finally, finite element analysis results were used to develop appropriate design formulae for the safe design of sandwich panels to be used in any building systems.

1.5 Layout of Thesis

The detailed investigations of various buckling modes of fully profiled and lightly profiled sandwich panels using an extensive series of experimental studies and finite element analyses and development of improved design rules for the design of safe
sandwich panel structures are presented in this thesis as seven different chapters. The contents of each chapter are described next:

**Chapter 1**  This chapter presents a brief introduction of sandwich panels including areas of applications, advantages, manufacturing processes, various buckling behaviour and design aspects, problem definitions for this research, overall and specific objectives of the study and methods of investigations.

**Chapter 2**  A summary of current literature relating to various aspects of sandwich panels, independent readings and critical analyses of previous findings are presented in this chapter. The broad areas included in this chapter are past research and development into sandwich panels, various buckling failure modes of sandwich panels, development of buckling stress formulae, methods of investigation of buckling behaviour, design procedure and considerations, testing methods, various failure modes including yielding of metal face, shear failure of the core, blistering, failure of fasteners, crushing of the panel, other influencing factors such as durability, temperature, creep etc., fire resistance behaviour and theory and application of numerical methods including finite element method.

**Chapter 3**  This chapter describes the experimental investigation carried out to investigate the local buckling behaviour of foam supported steel plate elements as used in the profiled sandwich panels. All the experimental results are presented, current design methods are reviewed, and critical analysis and interim design solutions are outlined.

**Chapter 4**  The detailed finite element analysis to investigate the local buckling behaviour of foam-supported flat plate elements as used in the fully profiled sandwich panels is discussed and the results are presented. New design rules for the design of sandwich panels subjected to local buckling effects that were developed based on finite element analysis results are presented in detail. It also includes details of calibrating the
finite element models against experimental and available analytical results.

**Chapter 5** A series of full-scale experiments on fully profiled sandwich panels carried out to examine the accuracy of the new local buckling design rule and to confirm the inadequacy of the current design rule is presented in this chapter.

**Chapter 6** This chapter describes and presents details of a series of experiments and finite element studies conducted to investigate the mixed mode type buckling behaviour of lightly profiled sandwich panels. All the results are presented, current design rules are reviewed and the new design rule is presented.

**Chapter 7** In this chapter, a summary of the most significant findings of this research and recommendations for further research are presented.
CHAPTER 2.0    LITERATURE REVIEW

2.1  General

Due to considerable structural importance and architectural appearance, a large number of publications dealing with structural sandwich panels are in existence. This chapter aims to provide a brief review of the sandwich panels in various aspects mainly based on previous research. A review of research on local buckling and post buckling behaviour of thin plates as used in fully profiled sandwich panels and available effective width formulae is presented. The theory and design formulae associated with the flexural wrinkling of the flat and lightly profiled sandwich panels is outlined. All the available design procedures, considerations, and testing methods of the sandwich panels are discussed. European recommendations (CIB, 2000) will be referred to any relevant discussions throughout the chapter. Sandwich elements may be one-dimensional (eg. beam, column, and strut) or two-dimensional (eg. panels). Similarly, foam cores can be isotropic or orthotropic and metal faces may be thick or thin. The discussions and review are focussed only on sandwich panels with isotropic foam core and thin metal faces.

A brief overview of the experimental methods and finite element studies carried out by previous researchers to investigate the buckling behaviour of sandwich panels is presented. The review is also extended to the various modes of failure the sandwich panels may experience. These may include yielding of the metal face, shear failure of the core, failure of fasteners, blistering, crushing of the panel, failure at a point of connection, and so on. Various other influencing factors such as thermal bowing, durability and creep are discussed. A brief summary is also provided on the research related to the fire resistance behaviour of sandwich panels.

2.2  Buckling Modes of Sandwich Panels

Under the action of different loading such as gravity, wind, snow, temperature gradient, and others, sandwich panels experience various types of buckling failures. The buckling failure mode depends on the type of steel faces used in sandwich
panels. Basically three types of buckling modes can be observed which are local buckling of plate elements of fully profiled faces (Figure 2.1), flexural wrinkling of flat and lightly profiled faces (Figure 2.2), and mixed mode buckling of lightly profiled faces due to the interaction of local buckling and flexural wrinkling (Figure 2.3). To develop the safe design rules and standards for sandwich panels, all these buckling failure modes have to be investigated thoroughly.

**Figure 2.1 Local Buckling of Sandwich Panels**

**Figure 2.2 Flexural Wrinkling of Sandwich Panels**

**Figure 2.3 Mixed Mode Buckling of Sandwich Panels**
The conventional design treatment for the local buckling phenomenon of fully profiled sandwich panels utilizes the concept of effective width. But recent research (Jeevaharan, 1997) has shown that conventional effective width formula can not be extended to fully profiled sandwich panels with increasing $b/t$ ratio (Figure 2.4). Davies and Hakmi (1992) proposed the enhanced effective width formula based on a very limited number of tests. These tests did not cover the wide range of plate $b/t$ ratios. The compression tests conducted by Jeevarahan (1997) aimed at investigating the design problems were inadequate as the experimental results had insufficient data to review the design rule.

Flexural wrinkling failure, which is the governing criteria for flat and lightly profiled panels, can be addressed successfully by the elastic half-space theoretical model in the case of flat panels. In the current practice, lightly profiled sandwich panels are also designed by using elastic half-space method with simple modification considering pure wrinkling failure. But recent research (McAndrew, 1999) has identified the inaccuracy of extending this approach to the lightly profiled sandwich panels with increasing rib/ridge heights and spacing of the profiles (Figure 2.5).
Previous research including that of McAndrew (1999) have indicated that lightly profiled sandwich panels with increasing depths and spacings of rib/ridge do not fail due to pure wrinkling. Instead, failure may occur due to the interaction of local buckling and flexural wrinkling as shown in Figure 2.3. CIB (2000) recommends that the current design formula for lightly profiled panels is applicable only when $b/t$ ratio of flat plates between the rib/ridge is less than 100. But the current ribbed profile used in Australia does not meet this requirement. Finite element analysis conducted by McAndrew (1999) showed that such ribbed panels failed at a lower strength than predicted by the flexural wrinkling formula. However, McAndrew (1999) did not investigate the design aspect of lightly profiled sandwich panels subjected to mixed mode type buckling failure.

### 2.3 Early Research and Development of Sandwich Construction

According to Hakmi (1988), Wood Fairbairn was the first person to use the idea of sandwich composites in the construction industry when he suggested a new design of Menai and Conway tubular bridges consisting of metal sheets adjacent to a layer of low-density material (Fairbairn, 1849). Later on, the same principle was suggested by von Karman and Stock (1884) and it has been used successfully in various fields. Since before the Second World War, sandwich construction has been widely used in aircraft and many structural applications. The structural analysis of sandwich panels with thin flat facings has been investigated as early as 1940’s, particularly for aeronautical applications (Allen, 1969; Gough et al., 1940; Hoff and Mautner, 1956). However, research and development of sandwich panels began only in late 1960s, pioneered by Chong and his associates (Chong and Hartsock, 1993). Hartsock published the method of analysis for sandwich panels with formed facings in 1969. Chong and Hartsock (1974) presented a method to predict the localised wrinkling instability of such panels. Hartsock and Chong (1976) conducted analysis of sandwich panels with foamed faces. Chong et al. (1977) investigated flexural behaviour and thermal stresses for both simple and continuous span panels.
In the past, a number of researches were conducted on various aspects of the behaviour of sandwich panels. Most of these researches were focussed on investigating the buckling behaviour of sandwich panels including local buckling, flexural wrinkling, etc. Some researches were conducted to model the sandwich panels using different numerical techniques such as finite strip, finite-layer, finite prism, and finite element analyses, whereas others concentrated on the investigation of suitable testing methods of sandwich panels, fire resistance behaviour, temperature effects, and so on. The various research works carried out by a number of researchers in this area are described next.

Gough et al. (1940) considered the problem of local wrinkling of the faces of sandwich struts by considering various cases of wrinkling instability with several combinations of face and supporting medium, taking into account distortion of the cross-section of the isotropic core. They assumed that the middle surface of each face is completely inextensible. This means that they did not consider the effect of bending, and therefore had only considered stress failure by instability in shear. Also they assumed that the core is attached to the middle of the faces instead of the inner side. They presented their results in the form of curves.

Williams et al. (1941) carried out the study on general problem of instability of sandwich struts with isotropic and non-isotropic (aeolotropic) cores. They modified the theory of Gough et al. (1940) by taking the length into consideration. They carried out a theoretical investigation of the possibilities of sandwich panels for transmitting compressive end loads, and their behaviour under such conditions. This analysis also identified the in-phase (anti-symmetrical waves) mode of instability over long wave lengths with the modified Euler mode, but it indicated that the short wave length mode should also occur in-phase (Hakmi, 1988).

Leggett and Hopkins (1942) modified the one dimensional theory presented by Williams et al. (1941) to suit sandwich plates taking both the dimensions into consideration (x and y directions). They assumed the following deflection equation:

\[ w = C \sin \frac{\pi x}{l} \sin \frac{n \pi y}{b} \]  

(2.3.1)
where \( l \) is a variable representing the semi-wave length in the direction of loading, and \( b \) is the width of the plate. They used the strain energy method but failed to present the formula for the critical load, and stated that “the results cannot, however, be expressed easily in algebraic form, so the details of the work are omitted”.

Hopkins and Pearson (1944) presented a report in which they discussed the elastic behaviour of sandwich plates with isotropic faces and presented experimental results for 12 sandwich panels designed for use as aircraft flooring with plywood faces and an onazote core. They employed the potential energy method to solve the elastic wrinkling of sandwich plates using the following series as a deflection equation:

\[
w = \sum \sum A_{mn} \sin(m \pi x / a) \sin(n \pi y / b) \quad (2.3.2)
\]

Taking maximum stress at the centre of the panel, stress in \( x \) direction was given as:

\[
\sigma_{ix} (\text{max}) = \frac{E_f}{2(1-\nu^2)} (-)^{m+n} \pi^2 t \sum \sum (k_{mn} + \epsilon)[(m / a)^2 + (n / b)^2 ](-)^{(m+n-2)/2} A_{mn} \quad (2.3.3)
\]

where \( \sigma_{ix}(\text{max}) \) is the maximum central stress in \( x \) direction, \( t \) is the thickness of the core, \( a \) is the length of the plate, \( b \) is the width of the plate, \( \epsilon \) is the strain at the centre of the plate, \( \nu \) is the Poisson’s ratio of the faces, and \( k_{mn} \) is a parameter whose value is given by a complicated formula.

A theory for buckling of sandwich panels with an isotropic core was presented by Reissner (1948) by neglecting the bending stiffness of the face layers normal to the planes, and taking the two faces into consideration. He used the finite deflection differential equations to obtain plate-buckling differential equations that include the effect of transverse shear stress deformation on the buckling load for anisotropic core. The differential equations that he presented are a modification of the fundamental equations for large deflections of a thin homogeneous plate derived by von Karman (1884).
Bijlaard (1951) introduced the method of split rigidities for the analysis of the elastic and plastic stability of sandwich plates for anti-symmetrical buckling. He split the rigidity of the structure into two cases. The first case is where the plate has its actual bending rigidity, but the core is assumed to be infinitely rigid against shear. The second case considered the core to have its actual shearing rigidity, but the plate is assumed to be infinitely rigid against bending. The total deflection splits into two parts $w_1$ and $w_2$ corresponding to cases 1 and 2. He discussed the case of Euler type buckling (overall buckling) and for the case of wrinkling he adopted the formula suggested by Hoff and Mautner (1948) for sandwich struts.

Nardo (1953) presented an elastic theory (small deflection theory) for the buckling loads of flat rectangular sandwich panels with loaded edges clamped, unloaded edges simply supported and the faces loaded edgewise in one direction. He used Hoff’s differential equation and employed the strain energy method to solve the problem. The results were presented as a series of design charts, which gave the buckling load as a function of the plate aspect ratio and the ratio of core to face thickness of 15 to 25. Each curve is valid for a particular value of the parameter representing the ratio of shear rigidity to bending rigidity.

Kan and Huang (1967) attempted to obtain an approximate solution for the post buckling (large deflection) problem of sandwich plates using the differential equation given by Reissner (1948), and employing the method of successive approximation to solve the resultant equation. They did not consider the imperfection problem, and nothing was mentioned in their paper about the shape of the deflection equation that they used or the critical wavelength. Their solution was based as they stated “upon the smallness of central deflection ratio for the cases of uniformly loaded, clamped and rectangular sandwich plates considering large deflection”. This means that they used the large deflection theory and assumed that the magnitude of the central deflection is the same as the thickness of the core. They used the ratio of the central deflection to the thickness of the core as a perturbation parameter, and employed the perturbation method to solve the problem. They failed to present a formula, which governs the wrinkling of sandwich plates. Instead they gave a numerical example for a specific case, and expressed the approximate results as load-deflection curves, and
presented an approximate equation for the case of square plates in which the load was given as a function of the central deflection (Hakmi, 1988).

Kao et al. (1973) presented a large deflection solution by modifying the Reissner (1948) differential equation by adding a third equation to them using the principle of complementary energy. They transformed the three governing differential equations into three systems of non-linear algebraic equations using a finite difference approach solved by successive iterations. But with this analysis no formula for the critical stress was reported.

2.4 Local Buckling of Thin Plates as used in Profiled Sandwich Panels

Thin steel faces supported by a thick foam core can be considered as a plate on elastic foundation. Mathematically the problem can be modelled as shown in Figure 2.6. A simply supported rectangular plate is subjected to an applied stress $p$ along the two transverse edges. The longitudinal edges of the plate are assumed to be simply supported. The length of the plate in the x-direction is generally large compared with the width. The solution for the buckling starts from the following relation given by Timoshenko and Gere (1961) for a thin plate without core. The buckled shape can be represented by a double sine series as given by (Davies and Hakmi 1990):

$$w = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} a_{mn} \sin \frac{m \pi x}{a} \sin \frac{n \pi y}{b}$$  \hspace{1cm} (2.4.1)
Considering only a single term in the x-direction, the relation can be modified as:

\[ w = \sum_{n=1}^{\infty} a_n \sin \frac{n \pi x}{a} \sin \frac{n \pi y}{b} \]  \hspace{1cm} (2.4.2)

where \( a \) is the unknown half wavelength of buckling mode. In order to evaluate the linear buckling stress of a long, uniformly compressed plate, generally the first term of the series is sufficient to represent the buckling mode. Other terms in the series are needed only in the non-linear, post-buckling range. The strain energy of bending in the plate is written by:

\[ U_n = \frac{\pi^4 abD}{8} \sum a_n^2 \left[ \frac{1}{a^2} + \frac{n^2}{b^2} \right]^2 \]  \hspace{1cm} (2.4.3)

where \( D \) is the flexural rigidity of the plate given by:

\[ D = \frac{E_f t^3}{12(1 - \nu_f^2)} \]  \hspace{1cm} (2.4.4)

The work done by the applied compressive force during buckling is

\[ V = \frac{\pi^2 pt}{8} \left[ \frac{b}{a} \right] \sum a_n^2 \]  \hspace{1cm} (2.4.5)

The evaluation of strain energy in the core can be achieved using three different ways. In the first approach, Winkler assumption is used. In this assumption, the foundation coefficient \( C_f \) is considered to have a simple constant value. Due to the difficulty involved in defining the constant \( C_f \) this method is of little value. The second method utilises the principle of elastic half-space within which all three displacements in three different directions are included. The third method also utilises the principle of elastic half-space, but only the most important stress components are included in the analysis and a single displacement function is used.
for the deformation of the core. This third method is also termed as ‘simplified’ method.

Using the elastic half-space method, the supporting foundation is modelled as an elastic-half space within which the displacements in the three directions of x, y, and z are expressed by three displacement functions:

\[
\begin{align*}
    u(x, y, z) &= h_1(z) \cos \alpha z \sin \alpha z \cos \alpha y \\
    v(x, y, z) &= h_2(z) \sin \alpha z \cos \alpha y \\
    w(x, y, z) &= h_3(z) \sin \alpha z \sin \alpha y
\end{align*}
\]  

(2.4.6)

The functions \( h_i(z) \) are determined on the basis of Navier’s equations for homogeneous, isotropic material. By the use of this equation, the following mathematical expression can be obtained for the foundation coefficient \( C_f \).

\[
C_f = \frac{2(1-\nu_c)\pi E_c}{(1+\nu_c)(3-4\nu_c)} \sqrt{\frac{1}{a^2 + \frac{n^2}{b^2}}}
\]  

(2.4.7)

This expression shows that the foundation coefficient \( C_f \) depends on the width of the plate and the buckle shape. The strain energy in the core material is found by the following expression:

\[
U_c = \frac{1}{2} \int_0^b \int_0^a C_f w^2 \, dx \, dy
\]  

(2.4.8)

After substituting the value of \( C_f \) from Equation 2.4.7 in 2.4.8, the final expression for the strain energy in the core can be found as:

\[
U_c = \frac{\pi ab(1-\nu_c)E_c}{4(1+\nu_c)(3-4\nu_c)} \sum a_n^2 \left[ \frac{1}{a^2 + \frac{n^2}{b^2}} \right]^\frac{1}{2}
\]  

(2.4.9)

Now, the total potential energy of an axially compressed plate undergoing buckling on an elastic support is given by:
\[ U = U_b + U_c - V \]  \hspace{1cm} (2.4.10)

Substituting the values from Equations 2.4.3, 2.4.5, and 2.4.9 to 2.4.10, the expression for total potential energy is written by:

\[
U = \frac{\pi^4 abD}{8} \sum a_n^2 \left[ \frac{1}{a^2} + \frac{n^2}{b^2} \right]^2 + \frac{\pi ab(1-\nu_c)E_c}{4(1+\nu_c)(3-4\nu_c)} \sum a_n^2 \left[ \frac{1}{a^2} + \frac{n^2}{b^2} \right]^{2/3} \]
\[ - \frac{\pi^2 pt}{8} \left[ \frac{b}{a} \right] \sum a_n^2 \]  \hspace{1cm} (2.4.11)

Minimising this total potential energy \( U \) with respect to the coefficient \( a_n \) in turn gives the buckling stress, \( \sigma_{cr} \).

\[
\sigma_{cr} = \frac{\pi^2 a^2 D}{t} \left[ \frac{1}{a^2} + \frac{n^2}{b^2} \right]^2 + \frac{2a^2(1-\nu_c)E_c}{\pi t(1+\nu_c)(3-4\nu_c)} \left[ \frac{1}{a^2} + \frac{n^2}{b^2} \right]^{1/3} \]
\[ - \frac{\pi^2 pt}{8} \left[ \frac{b}{a} \right] \sum a_n^2 \]  \hspace{1cm} (2.4.12)

Introducing \( \phi = a/b \), critical buckling stress, \( \sigma_{cr} \), in the elastic region is reduced to

\[
\sigma_{cr} = K \frac{\pi^2 E_f}{12(1-\nu_f^2)} \left[ \frac{t}{b} \right]^2 \]  \hspace{1cm} (2.4.13)

where \( K \) is the buckling coefficient and given by

\[
K = \left[ \frac{1}{\phi} + n^2 \phi \right]^2 + R\phi \left[ 1 + n^2 \phi^2 \right]^{1/3} \]  \hspace{1cm} (2.4.14)

and

\[
R = \frac{24(1-\nu_f^2)(1-\nu_c)E_c}{\pi^2(1+\nu_c)(3-4\nu_c)E_f} \left[ \frac{b}{t} \right]^3 \]  \hspace{1cm} (2.4.15)

If the simplified foundation model is used, the simplified value of \( R \) is given by
Critical buckling stress itself does not provide any satisfactory basis for design, but it can be used as a useful design parameter. It is well known that in cold-formed steel design, the width to thickness ratios are usually large, hence local buckling becomes a major design criterion for compression members. Buckling of these elements may occur at a stress level lower than the yield stress of steel. However, for the plates with a considerably low $b/t$ ratio, the elastic local buckling does not represent the collapse of the members. Failure will occur at a load higher than the elastic-buckling load. Initial buckling is followed by redistribution of internal stresses enabling the members to carry increasing loads due to the post-buckling strength. Thus, post-buckling behaviour is important for the optimum design of cold-formed steel members and this raises a significant analytical problem. For the cold-formed steel members without any foam support, such local and postbuckling problems are treated for design purposes by utilising the concept of effective width. A widely used effective width formula in many national and international standards is the “Winter” formula. The principal associated with this is: the width $b$ of the compressed element is replaced by the reduced value of the width $b_{\text{eff}}$ when calculating the section properties for use in the design calculations as shown in Figure 2.7. The “Winter” formula takes the form:

$$R = \frac{12(1-\nu^2)}{\pi^3} \sqrt{\frac{E_r G_c}{E_f}} \left[\frac{b}{t}\right]^3$$  \hspace{1cm} (2.4.16)
$b_{\text{eff}} = \rho b$

\[
\rho = \begin{cases} 
1 & \text{for } \lambda > 0.673 \\
1 - \frac{0.22}{\lambda} & \text{for } \lambda \leq 0.673 
\end{cases}
\]

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

where $b_{\text{eff}}$ = effective width of the plate, $f_y$ = yield stress of steel, $E_f$ = Young’s modulus of steel, $t$ = thickness of the steel plate, $K$ = buckling coefficient ($= 4.0$ for simply supported longitudinal edges).

This effective width approach can be extended to the profiled faces of sandwich panels by modifying the buckling coefficient $K$ to take into account the core support. It is obvious that the value of buckling coefficient $K$ will increase due to the plates stiffened by core material and thus raising the value of buckling stress $\sigma_{cr}$. As seen from Equations 2.4.15 and 2.4.16, the influence of the composite action between faces and core is modelled by the dimensionless stiffness parameter $R$. In both actual and simplified methods, the critical buckling stress $\sigma_{cr}$ can be found by minimising the buckling coefficient $K$ with respect to the wavelength parameter $\phi$. Hence the condition $\partial K/\partial \phi = 0$ from Equation 2.4.14 gives

\[
2n^4 \phi - \frac{2}{\phi^2} + R(2n^2 \phi^2 + 1)(n^2 \phi^2 + 1) \frac{1}{\phi^2} = 0
\]

(2.4.18)

If the elastic support given by the core is ignored, the buckling coefficient in Equation 2.4.13 has the value $K = 4.0$. If the elastic support provided by the core is utilised in the evaluation of the critical buckling stress of the face, Equation 2.4.18 can be solved for $\phi$ using a suitable numerical method and $K$ can be evaluated. It is found that in the range of $0 < R < 200$, the primary buckling mode with $n = 1$ is always critical (Davies et al., 1991). For the practical design purpose, a number of explicit mathematical formulae have been proposed for the solution of Equation 2.4.18 to determine the enhanced buckling coefficient $K$ in sandwich panels with profiled faces. These mathematical approximations are given next:
1. By Davies et al. (1991) based on the half-space assumption

\[ K = 4 - 0.415R + 0.703R^2 \quad \text{with} \quad R = \frac{b}{t} \left[ \frac{E_c}{E_f} \right]^{1/3} \quad (2.4.19) \]

2. By Davies et al. (1991) based on the simplified foundation model

\[ K = 4 - 0.474R + 0.985R^2 \quad \text{with} \quad R = \frac{b}{t} \left[ \frac{E_c G_c}{E_f^2} \right]^{1/6} \quad (2.4.20) \]

3. By Davies and Hakmi (1990) based on the simplified foundation model

\[ K = [16 + 11.8R + 0.055R^2]^{1/2} \quad \text{with} \quad R = \frac{12(1 - \nu_f^2)\sqrt{E_c G_c}}{\pi^3 E_f} \left[ \frac{b}{t} \right]^3 \quad (2.4.21) \]

4. By Davies and Hakmi (1990) based on the simplified foundation model by replacing \( R \) with \( 0.6R \)

\[ K = [16 + 7R + 0.02R^2]^{1/2} \quad (2.4.22) \]

5. By Mahendran and Jeevaharan (1999) based on the simplified foundation model to include a greater range of \( R \) from 0 to 600

\[ K = [16 + 4.76R^{1.29}]^{1/2} \quad \text{with} \quad R = \frac{12(1 - \nu_f^2)\sqrt{E_c G_c}}{\pi^3 E_f} \left[ \frac{b}{t} \right]^3 \quad (2.4.23) \]

In the current European Recommendations for Sandwich Panels, Part I: Design (CIB, 2000), the following formula has been recommended for predicting the value of \( K \)

\[ K = [16 + 7R + 0.02R^2]^{1/2} \quad \text{with} \quad R = 0.35 \frac{\sqrt{E_c G_c}}{E_f} \left[ \frac{b}{t} \right]^3 \quad (2.4.24) \]
The enhanced buckling coefficient $K$ derived from one of the equations from 2.4.19 to 2.4.24 is used in the effective width method by substitution into the general Equation 2.4.17 for plates.

In the past few decades, several studies have been carried out by different researchers to investigate the local buckling behaviour of fully profiled sandwich panels. Linke (1978) presented an important theoretical analysis for elastic buckling and post-buckling behaviour of sandwich plates with special boundary conditions. He used the energy method, and considered the initial waveness of the sandwich plate. Newton’s iterations were used to solve the resulting equations. However, he could not formulate any critical load formulae for the elastic or the plastic buckling.

Davies (1987b) presented a report on axially loaded sandwich panels. This report first considers appropriate methods of analysis for sandwich panels subject to combined axial compressive load and bending moment and an exact finite element solution has been derived. Some previously unpublished tests were described and the results compared with the theoretical values. Davies concludes that great care must be taken in the design of load bearing wall panels because of the interaction between local and global buckling and their sensitivity to the eccentricity of load.

Hakmi (1988) carried out a detailed study on local buckling of sandwich panels. This research was concerned with the local buckling of thin plates stiffened by an isotropic medium. The study commences with a review of previous research into structural aspects of sandwich construction. The problem under consideration originated in a theoretical analysis carried out by Linke (1978) in Germany using an approximate energy method. Hakmi developed a completely new theory for the linear and non-linear buckling of sandwich plates based on an exact series solution of the governing differential equations.

Davies and Hakmi (1990) presented a report on local buckling of profiled sandwich plates. In this report, the local buckling behaviour of a compressed plate element supported by a relatively weak isotropic medium has been considered. As sandwich panels used in building construction typically consists of two metal faces and a
foamed plastic core, one or both faces may have a trapezoidal or similar profile for either structural or aesthetic reasons. When such a panel is subjected to static loading the profiled face may be compressed and therefore is liable to failure by local buckling. They have presented some practical solutions to address such a problem.

Davies and Hakmi (1992) conducted a series of tests on foam-filled, thin-walled steel beams to study the postbuckling behaviour. The conventional design treatment for the local buckling of sandwich panels utilises the concept of effective width. Their experimental study investigated the extension of the effective width concept to the plate elements supported by plastic foam material. The test results were then used to formulate an enhanced effective width formula (Equations 2.4.21 and 2.4.17). However, this formula is applicable only for relatively low b/t ratios of steel plate elements, generally less than about 100 and can not be extended to thinner plates. They proposed this enhanced effective width formula based on very limited number of tests. Thus, extensive experimental and analytical studies need to be conducted to make an appropriate modification to the existing effective width formula, so that it can be used to slender plates with enhanced accuracy.

Uy and Bradford (1996) investigated the local buckling of a plate in contact with a rigid restraining medium such as concrete. They conducted experiments on concrete filled steel column to study the local and post buckling behaviour of steel plates in composite steel-concrete member. A semi-analytical finite strip method was used to study the behaviour numerically and a post local buckling model was established based on the effective width principle to determine the strength of a concrete filled box section. As concrete is the rigid medium, it restrains the free formation of buckles in the steel plate and forces them to form away from the concrete. However, in sandwich panel, foam core is the flexible medium and it does not restrain the free formation of buckles in steel faces. Therefore formation of buckles occurs both towards the foam and away from the foam making the complete sine wave. Because of this fundamental difference in buckling behaviour of steel faces in rigid and flexible mediums, the design principles established for composite steel-concrete member can not be applied to sandwich panels.
Sironic et al. (1999) presented a new solution for the buckling of an infinitely long plate experiencing in-plane loading, glued to an elastic foundation. The accuracy of Winkler model which ignores the presence of shear stress in the foundation and between the foundation and the plate was reviewed. The Pasternak model, an improved Winkler model by the addition of shear term to the governing equation, was described. The results obtained from the new solution were then compared with those from Winkler’s and Pasternak’s models. An attempt was made to unify the newly presented model and the Winkler/Pasternak models so as to obtain a formulation exhibiting the accuracy of the proposed model with simplicity of the Winkler/Pasternak models. Further, it was shown that the behaviour of the plate is affected by the foundation depth when it is less or equal to the half-wave buckle length and beyond that there is no effect on the buckling stress. In Sironic et al.’s (1999) model for an infinitely long plate, the foundation was restrained at the base and it is mainly concerned with Euler type of buckling. Therefore it can not be extended to a sandwich panel subjected to local buckling and flexural wrinkling failures as the elastic foundation in this case is considered infinitely deep with free base. However, a separate study can be undertaken to investigate the possibility of extending the application of Pasternak model to sandwich panels.

During 1995 to 1999, QUT researchers conducted two research projects to investigate the buckling behaviour of sandwich panels. These projects produced two Masters theses entitled “local buckling behaviour and design of sandwich panels in buildings” by Manohara Jeevaharan (1997), and “behaviour and design of sandwich panels for flexural wrinkling” by Duncan McAndrew (1999). The second project was a collaborative research project with industry partner James Hardie Building Systems. These two projects provided the basic background and hence a convenient starting point to continue the research in this area.

Jeevaharan (1997) carried out an extensive study on the local buckling behaviour and design of sandwich panels in buildings. An investigation using finite element analyses and laboratory experiments was carried out on steel plates of varying yield stresses and thicknesses supported by polystyrene foam core. This research was aimed to investigate the validity of European Design Recommendations for Australian sandwich panels made of high strength steel and polystyrene foam core.
European Design Recommendations for foam filled steel members are based on polyurethane foam core and low tensile steel. The European design recommendations were published by the European Conventional Steelwork (ECCS) and recently a new version (CIB, 2000), has been published. This document was produced by an international committee of experts and has considerable credibility within Europe. In fact, this standard is being used as de facto National Standards, where no alternative document exists.

Based on the series of tests and FEA results, Mahendran and Jeevaharan (1999) presented a detailed report on local buckling behaviour of steel plate elements supported by a plastic foam material. In order to extend the applicability of the design Equation 2.4.21 to higher values of $R$ (ranging from 0 to 600), they proposed an improved design Equation 2.4.23 to determine the buckling coefficient $K$ of the steel plate elements supported by foam core. Design Equation 2.4.21 presented by Davies et al. (1990) is valid only for a range of $R$ from 0 to 200 and $b/t$ ratio less than 120 (Davies and Hakmi, 1992). Although Davies et al. (1991) indicated that the assumption that the primary buckling mode is always critical ($n = 1$) is true only for a range of $R$ from 0 to 200, the analytical and experimental results of Mahendran and Jeevaharan (1999) confirmed that this assumption is also valid for higher value of $R$ up to 600. Compared with the design Equation 2.4.21 proposed by Davies et al (1991), this improved Equation 2.4.23 is more suitable to find the enhanced buckling coefficient $K$ for the plates with higher $b/t$ ratios. However, this equation is also inadequate to determine the true value of effective widths for the slender plates.

### 2.5 Flexural Wrinkling of Flat Faced Sandwich Panels

Chong and Hartsock (1974) presented a method for predicting the flexural buckling of foam-filled sandwich panels with light gauge cold-formed metal faces. The study pointed out some of the important issues such as critical wrinkling stress can be computed from the properties of the face and core, that the shear modulus of plastic foam cores can be computed simply from lateral deformation measurements made during compression tests, and the critical wrinkling stress may serve as a design basis for the allowable compressive stress in the face.
Chong (1986) conducted studies on sandwich panels with cold-formed thin facings. This study described the structural behaviour including flexural stresses/deflections, axial stability, and thermal stresses, summarising more than a decade of research. The methods used in this research were analytical (boundary-value), numerical (finite-strip, finite-layer, finite prism), and experimental (full-scale testing).

Sandwich panels with flat faces are of particular interest to the aerospace industry and hence flexural wrinkling has been investigated by many authors (Davies, 1993). Analytically, wrinkling stress can be developed by treating the core as an elastic half-space. If the width $b$ of the thin plate in Figure 2.6 increases to infinity the sandwich panel with a wide flat face is obtained. The buckling stresses of such wide flat-faced sandwich panels can be determined by using the same elastic half-space principle as used before in the profiled sandwich panels. Thus, flexural wrinkling strength for flat faces is obtained by simplifying Equation 2.4.12 in the following form:

$$\sigma_{wr} = \frac{\pi^2 D}{a^2 t} + R \frac{a}{\pi t}$$  \hspace{1cm} (2.5.1)

where $R = \frac{2(1-\nu_c)E_c}{(1+\nu_c)(3-4\nu_c)}$ (half-space)  \hspace{1cm} (2.5.2)

$$R = \left[ E_c G_c \right]^{1/2}$$ (simplified)  \hspace{1cm} (2.5.3)

Minimising $\sigma_{wr}$ with respect to the half wave buckle length $a$ gives

$$\frac{d\sigma_{wr}}{da} = -\frac{2\pi^2 D}{a^3 t} + \frac{R}{\pi t} = 0$$

$$a = \pi \left[ \frac{2D}{R} \right]^{1/3} = \pi t \left[ \frac{E_f}{6(1-\nu_f^2)R} \right]^{1/3}$$  \hspace{1cm} (2.5.4)

$$\sigma_{wr} = \frac{1.89}{t} \left[ DR^2 \right]^{1/3} = 1.89 \left[ \frac{E_f R^2}{12(1-\nu_f^2)} \right]^{1/3}$$  \hspace{1cm} (2.5.5)

Substituting $R$ into Equation 2.5.5, the expression for wrinkling stress becomes
\[ \sigma_{wr} = 1.89 \left( \frac{2(1-\nu_f)^2}{3(1-\nu_f^2)(1+\nu_c)(3-4\nu_c)^2} \right)^{1/3} \left( E_f E_c G_c \right)^{1/3} \text{ (half-space) \hspace{1cm} (2.5.6)} \]

\[ \sigma_{wr} = 1.89 \left( \frac{1}{12(1-\nu_f^2)} \right)^{1/3} \left( E_f E_c G_c \right)^{1/3} \text{ (simplified) \hspace{1cm} (2.5.7)} \]

If the values of Poisson’s ratio are assumed to be \( \nu_f = 0.3 \) and \( \nu_c = 0.25 \), then the following approximate expressions for the wrinkling stress are obtained.

\[ \sigma_{wr} = 0.823 \left( E_f E_c G_c \right)^{1/3} \text{ (half-space) \hspace{1cm} (2.5.8)} \]

\[ \sigma_{wr} = 0.852 \left( E_f E_c G_c \right)^{1/3} \text{ (simplified) \hspace{1cm} (2.5.9)} \]

Practical considerations, such as the lack of flatness of the face and non-linearity of the core material, usually imply that the theoretical wrinkling stress is not achieved and wrinkling takes place at a stress lower than that predicted by Equations 2.5.8 and 2.5.9. For practical purposes, the wrinkling stress \( \sigma_{wr} \) can be given by

\[ \sigma_{wr} = K \left( E_f E_c G_c \right)^{1/3} \text{ \hspace{1cm} (2.5.10)} \]

where \( K \) is a numerical constant less than 0.823 and may be determined experimentally for a particular product. For practical design, the constant \( K \) is given by a value of 0.65 (CIB, 2000). This means a further reduction factor of about 0.8.

### 2.6 Flexural Wrinkling of Lightly Profiled Sandwich Panels

Lightly profiled sandwich panels are generally considered to be those with faces of profile depth of up to 2 mm. Even with such a small depth of profile, a significant increase in wrinkling stress can result according to several researchers (Davies, 1993; Hassinen, 1995; Kech, 1991). The current design procedures for lightly profiled faces in compression are based on modifying the methods utilised for flat faces. Two
practical methods are used for obtaining the design values of wrinkling stress of lightly profiled sandwich panels.

(a) In the first method the analysis for flat faces is simply modified by taking into account the bending stiffness of lightly profiled face, and the effect of foam core depth is disregarded (Davies et. al, 1991). The derivation process of this equation is based on the elastic half-space method. The design recommendations adopted in the current European standard (CIB, 2000) is based on this method. Hence, considering the effect of bending stiffness, Equation 2.5.1 can be modified as:

$$\sigma_{wr} = \frac{\pi^2 B_f}{a^2 A_f} + \frac{R a}{\pi A_f} \tag{2.6.1}$$

where $A_f$ is the cross-sectional area of the face per unit width ($\approx t$), $B_f$ is the flexural rigidity of the face per unit width ($=E_f I_f/b$), $a$ is the half-wavelength of the buckling mode and $R$ is the parameter defined by Equation 2.5.2. Minimising the critical buckling stress $\sigma_{wr}$ with respect to the half-wave buckle length $a$, Equation 2.6.1 reduces to

$$\frac{d\sigma_{wr}}{da} = -\frac{2\pi^2 B_f}{a^3 A_f} + \frac{R}{\pi A_f} = 0$$

$$a = \pi \left[ \frac{2B_f}{R} \right]^{1/3} \tag{2.6.2}$$

Substituting $a$ in Equation 2.6.1 gives

$$\sigma_{wr} = \frac{1.89}{A_f} \left[ B_f R^2 \right]^{1/3} \tag{2.6.3}$$

Substituting $R$ into this equation the expression for flexural wrinkling strength becomes
\[
\sigma_{cr} = \frac{1.89}{A_f} \left( \frac{8(1-\nu_c)^2}{(1+\nu_c)(3-4\nu_c)^2} \right)^{1/3} \left( E_c G_f B_f \right)^{1/3} \quad (2.6.4)
\]

If the values of Poisson’s ratio are assumed to be \( \nu_f = 0.3 \) and \( \nu_c = 0.25 \), then the following approximate expression for the wrinkling stress is obtained.

\[
\sigma_{wr} = \frac{1.825}{A_f} (E_c G_f B_f)^{1/3} \quad (2.6.5)
\]

If the value of Poisson’s ratio are assumed to be \( \nu_f = 0.3 \) and \( \nu_c = 0 \), then the following approximate expression for the wrinkling stress is obtained.

\[
\sigma_{wr} = \frac{1.82}{A_f} (E_c G_f B_f)^{1/3} \quad (2.6.6)
\]

From Equations 2.6.5 and 2.6.6, it is obvious that the constant (i.e. 1.825 and 1.82) does not change significantly with change to the Poisson’s ratio of the foam core. Because of the practical considerations such as imperfections, material non-linearity, and inadequacy of the analysis, the theoretical wrinkling stress is not achieved and wrinkling takes place at a stress lower than that predicted by the above equations. Further, the interaction between local buckling of the flat plates and the buckling of the complete face is not included in the analysis. Due to these constraints, European recommendation (CIB, 2000) has proposed a semi-empirical equation with a large reduction factor of about 0.5 for the wrinkling stress.

\[
\sigma_{wr} = \frac{0.95}{A_f} (E_c G_f B_f)^{1/3} \quad (2.6.7)
\]

(b) In the second method, a factor \( \alpha \) is applied to the equation for a flat face and is given by:

\[
\sigma_{wr} = 0.65\alpha (E_f E_c G_c)^{1/3} \quad (2.6.8)
\]
The value of $\alpha$ is greater than 1 ($\alpha > 1$) and is determined by testing. The major drawback of this technique is that $\alpha$ is not a constant for a particular profile but tends to vary with the thickness and the properties of the core. Since current theoretical methods have several limitations, European recommendations for sandwich panels (CIB, 2000) states that design by testing is the most proficient method with regard to sandwich panels with lightly profiled faces.

Mahendran and McAndrew (2000) investigated experimentally and analytically the behaviour and design of lightly profiled sandwich panels for flexural wrinkling. The aims of this research were to investigate the flexural wrinkling behaviour of sandwich panels, which contain transverse joints and the increase in wrinkling capacity of lightly profiled panels over flat panels. In Australia, expanded polystyrene is commonly used as the core material introducing several new considerations such as adhesive bond, the presence of transverse joints in the foam core, and so on. In these circumstances, this project revealed some useful information in the application of sandwich panels in the building systems. Extensive experimental and finite element analyses were carried out to determine the effects of transverse joints of the foam core and lightly profiled faces on the wrinkling capacity of sandwich panels. More details of their work including some important results are given in Section 2.8.

2.7 Mixed Mode Buckling of Lightly Profiled Sandwich Panels

Light profiling of the sandwich panel can significantly increase the wrinkling stress. However, as the depth and spacing of the profiles increase, flat plate buckling between the ribs becomes possible (Davies, 1993). The finite element analyses and experimental studies conducted by McAndrew (1999) have also pointed out that local buckling was found to be occurring between the ribs of sandwich panels with higher spacing of ribs causing failure to occur at stresses lower than that predicted by the wrinkling formula. This clearly indicates that failure occurs in lightly profiled panels due to the interaction between the flexural wrinkling of overall panel and local buckling of the flat plates between the ribs.
Kech (1991) proposed an equation, which takes into account the interaction between two buckling modes in lightly profiled faces. This equation is considered to be an improvement to the earlier developed wrinkling stress formula. The model derived by Kech is based on considering the folded area, including effective width of the flat plates on either side of it, as an axially compressed column on an elastic foundation. The cross-section of this column and nomenclature are shown in Figure 2.8.

The wrinkling stress $\sigma_{wr}$ is taken to be 0.55 times the buckling stress $\sigma_k$ of this equivalent column averaged over the full panel width but not less than the flat face buckling stress, for which Kech uses Equation 2.5.10 with $K = 0.75$. The calculation of $\sigma_{wr}$ proceeds iteratively as follows:

$$\sigma_{wr} = \sigma_k \frac{b_{ef1} + b_{ef2} + 2b_s}{b_k} \quad (2.7.1)$$

where $b_{ef,i} = b_i \frac{\sigma_{cr,i}}{\sigma_k}$

$$\sigma_k = \frac{3}{2t} \sqrt{2B_{ef} K_{ef}^2} \quad (2.7.3)$$

is the effective width of element $i$ according to the von Karman equation.

but is not greater than the yield stress of the face material.
\[ K_{ef} = \frac{2(1 - \nu_c)E_c}{(1 + \nu_c)(3 - 4\nu_c)} \left[ \frac{\kappa h / (3 - 4\nu_c) + \sinh(\kappa h) \cosh(\kappa h)}{\sinh^2(\kappa h) - [\kappa h / (3 - 4\nu_c^2)]} \right] \]  
\hspace{2cm} (2.7.4)

where \( \kappa = \frac{\pi}{a} \)  
\hspace{2cm} (2.7.5)

and the wavelength of buckling \( a \) is given by

\[ a = \pi \sqrt{\frac{2B_{ef}}{K_{ef}}} \]  
\hspace{2cm} (2.7.6)

and where \( B_{ef} \) is the bending stiffness of an equivalent column (which depends on \( b_{ef,i} \)), \( \sigma_{cr,i} \) is the buckling stress of the face \( i \), which is given by:

\[ \sigma_{cr,i} = \frac{4\pi^2 E_f}{10.92} \left( \frac{t}{b_i} \right)^2 \]  
\hspace{2cm} (2.7.7)

where \( b_i \) is the width of the element \( i \) and \( h \) is the depth of the panel between the centroid of the faces. The second part of the equation for \( K_{ef} \) is a correction term to take into account the finite thickness of the core layer relative to the elastic half-space assumption used in the derivation of the buckling stress.

Kech claims this procedure to be valid for \( \nu/t \leq 10 \), which covers most practical cases. But this equation was verified by comparison with a limited number of test results and it was found that the method is valid for very small ratios of profile depth to plate thickness (Davies, 1993). The accuracy of Kech’s equation needs to be investigated by more extensive comparison with test results and more accurate analysis. Also Kech’s equation is very complicated to use in the design. Hence, further research needs to be conducted for lightly profiled panels to develop a more accurate equation that takes into account the interaction of the two buckling modes.
2.8 Methods of Investigation of Buckling Behaviour

Investigation of various buckling failures of sandwich panels is achieved by either experimental methods or numerical methods including finite element analyses. In the past, many researchers have conducted experiments on sandwich panels to investigate various aspects of sandwich panels. Similarly, investigations were done using analytical methods such as boundary-valued approach and numerical methods such as finite-strip, finite layer, and finite prism approaches. Some of the experiments conducted on sandwich panels to investigate the local buckling and flexural wrinkling behaviour are discussed here and some of the finite element studies available in the literature are reported.

2.8.1 Experimental Investigation

(a) Davies and Hakmi’s (1992) Test for Local Buckling

In order to investigate the extension of effective width concept to plate elements supported by plastic foam material, Davies and Hakmi (1992) conducted a series of bending tests on foam-filled steel beams. The tested beams were folded from galvanised steel sheet to the dimensions shown in Figure 2.9 (a). All dimensions were kept constant except for the width $b$, which was varied to allow breadth-to-thickness ratios for the compression flange in the range of 70 to 250. Both plain steel and foam-filled beams were tested. The density of the foam, and therefore its mechanical properties were also varied.

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Figure 2.9 Davies and Hakmi’s (1992) Bending Test
A total of 18 beams were tested, of which 13 were foam-filled. The beams were tested under four-point loading as shown in Figure 2.9 (b). Load was applied in increments by adding weights to hangers and, at each load level, the deflection at salient points was measured using dial gauges. Table 2.1 presents the summary of their results. In this table, the quoted failure load was the maximum load carried corresponding to local buckling of the compressed flange. The effective width of the compressed flange was determined by equating this load to the moment of resistance of the reduced cross-section. For each of the tested beams, the density, compression modulus and shear modulus were determined by testing. The measured material
properties were used in the evaluation of the test results. The yield stress was taken as constant with an average value of 281 N/mm².

In Figure 2.10, Davies and Hakmi’s (1992) test results are compared with their proposed formula developed based on these test results. For the plate elements of relatively low slenderness (less b/t ratio), the predictions are in close agreement. For plates of higher slenderness, their formula overestimates the plate strength. Hence, for the plates with higher b/t ratios, a more rigorous analysis and extensive series of tests covering a wider range of b/t values are required. However, Davies and Hakmi (1992) have suggested that the non-dimensional parameter R should be multiplied by an empirical reduction factor of 0.6 for the safe solution in the design practice. However, this also gives unconservative results for the slender plates.

![Figure 2.10 Effective Width vs Modified Slenderness based on Davies and Hakmi’s Test (1992)](image)

(b) Test conducted at the Technical Research Centre of Finland for Local Buckling

The ultimate strength of compressed steel plates with and without core support was investigated at the Technical Research Centre of Finland (VTT) using compression tests with the apparatus shown in Figure 2.11. This test was reported by Davies et al. (1991) in their paper entitled “Face buckling stresses in sandwich panels”. The plate elements were simply supported along their longitudinal edges and were rigidly connected to the loading plattens at both ends.
The length to width ratio of the specimens varied between 3 and 18 and the width to thickness ratio \( b/t \) between 26 and 200. The modulus of elasticity and shear modulus of the foam varied between 1.6 and 8.5 N/mm\(^2\) and 1.3 and 2.6 N/mm\(^2\), respectively. After taking account of the effect of foam core, the experimentally obtained effective widths are compared with the Winter curve and the Winter curve modified by multiplying \( R \) by 0.6 as shown in Figure 2.12. Here the test results have a considerable scatter and are again unsafe in comparison with the unmodified Winter curve. These test results clearly indicate that the proposed formulae are not suitable for the slender plates (high \( b/t \) ratios).
(c) Jeevaharan’s (1997) Tests for Local Buckling

In order to verify the European design recommendations for the high strength steel and polystyrene foam core, Jeevaharan (1997) conducted an extensive series of compression tests. Steel plate elements with and without foam core were used in the investigation. Since one of the objectives of this investigation was to study the applicability of the design rules developed for mild steel to high strength steel, a low strength steel grade (G250) and a high strength steel grade (G550) were chosen. For each steel grade, four different nominal (specified) thicknesses of 0.4, 0.6, 0.8 and 1.0 mm were used. The width $b$ of the plate elements was varied to cover the large range of $b/t$ ratio (105 to 513) to enable the study of local buckling behaviour of very thin plate elements used in sandwich construction. Thickness of the foam core was taken as half the width of the plate element ($b/2$). A complete diagram of the test specimen and test rig is given in Figure 2.13.

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Figure 2.13 Compression Test (Jeevaharan, 1997)

From this study, it was concluded that the earlier developed formula (Equation 2.4.21) by Davies and Hakmi (1990) can be used for $R$-value up to 200. Based on the results, Mahendran and Jeevaharan (1999) proposed an improved design equation (Equation 2.4.23) to cover high $b/t$ ratios and $R$-values up to 600. However, their design formula is inadequate for steel plates with very high slenderness.

(d) McAndrew’s (1999) Test for Flexural Wrinkling
McAndrew (1999) conducted a series of tests on sandwich panels to investigate the effect of joints (gap and step imperfections shown in Figure 2.14) in the panels and the increase in wrinkling capacity of lightly profiled panels over flat panels. The experimental study included full scale bending tests using a vacuum test rig to determine the flexural wrinkling stresses, and small-scale tests to determine the mechanical properties of the foam. The material properties obtained from the small-scale tests were used to determine the theoretical wrinkling failure stresses.

**Figure 2.14 Step and Gap Imperfections in Sandwich Panels (McAndrew, 1999)**

Sandwich panels with three different compression faces, flat, ribbed and satinlined (see Figure 2.15) were considered in this study. All the test panels were manufactured by James Hardie Building systems Pty Ltd in Brisbane. The foam thicknesses considered were 75, 150, and 200 mm and the spans varied from 2700 to 4700 mm. The width of the panels was 1215 mm. Steel thickness considered were 0.4, 0.5, and 0.6 mm. Four different types of foam joints were investigated in the experimental study. These were full-width butt joints, half-width butt joints, full-width butt joints which were glued, and scarfed joints as shown in Figure 2.16.
A vacuum chamber was used to produce a uniformly distributed transverse pressure loading of the panels as shown in Figure 2.17, enabling flexural wrinkling failures to occur in bending.

Following the experimental study, Mahendran and McAndrew (2001) carried out some finite element studies as well. Based on experimental and finite element analysis results, some useful recommendations were made for the design of sandwich panels for flexural wrinkling. The presence of transverse joints in the foam core significantly reduced the wrinkling stress of sandwich panels tested in the experimental investigation. Therefore for flat sandwich panel with transverse joints, a further reduction factor of 0.76 based on FEA was recommended for panels with transverse ‘gap’ and ‘step’ imperfections. The reduction in strength due to gap size is
shown in Figure 2.18. To help eliminate the possibility of shear forces influencing panel failure, it was recommended that a joint should only extend half the width of a sandwich panel. From FEA results, it was observed that span has no influence on the wrinkling stress of sandwich panels, supporting theoretical predictions and experimental test results.

Figure 2.18 Wrinkling Stress vs Gap Size (Mahendran and McAndrew, 2001)

From their study, it was identified that a small increase to the rib/ridge height of the lightly profiled sandwich panels can significantly improve the flexural wrinkling strength of these panels. However, the results showed that flexural wrinkling stresses obtained from experiments did not agree with the theoretical prediction for the panel with increasing rib/ridge height and spacings of the ribs.

For the ribbed profiled panels, local buckling was found to be occurring in the flat plates of the panel causing failure to occur at stresses lower than that predicted by the wrinkling formula. The decrease in buckling strength is due to the interaction of two buckling modes as identified from the FEA results. Present formula for wrinkling stress (CIB 2000) is applicable only for $b/t$ ratios of flat plate of lightly profiled sandwich panels less than 100. Current Australian sandwich panels do not meet this requirement. Hence, it was recommended that further research needs to be conducted for lightly profiled panels to develop a wrinkling equation that takes into account the interaction between the two buckling modes.
2.8.2 Finite Element Analysis

There are several numerical methods available such as finite strip, finite layer, finite prism, finite difference, and finite element analysis approaches for the analysis and design of structures. Pomazi (1966) applied the finite difference technique to solve the bending problem of simply supported multi-layer plates. Jungbluth and Berner (1986) also described a finite difference technique, which appeared to be the favoured method in Germany. Chan and Cheung (1972) presented the finite strip solution for bending and vibration of plate structures. Later, the stability analysis of such method was extended by Foo (1977). Cheung et al (1982) utilised finite layer model to approximate the sandwich panel to enable a full three-dimensional analysis. Another technique was developed in the US in which thin faces are modelled by finite shell strips and relatively weak cores by finite prisms (Chong, 1986). However, for more general applications, the conventional finite element method offers the best approach (Davies, 1993).

Many finite element models have been proposed for the analysis of sandwich panels. Davies (1993) has pointed out that in many applications, the finite element method is approximate and it is necessary to use a large number of elements in order to obtain accurate solutions. But for three-layered sandwich beams, the solutions are exact and the minimum number of elements necessary to model the problem will give a precise solution.

Currently there are several finite element programs available for the analysis and design of structures. Commercially available finite element programs such as SAP, NASTRAN, NISA, ANSYS, and ABAQUS are capable of treating very complex problems.

Davies et al. (1991) used the finite element package, ABAQUS, when considering the post-buckling strength of sandwich panels with profiled faces. 64 shell elements with 8 nodes were used to model the steel face and 128 3D solid elements with 20 nodes in two layers to model the core. A buckling analysis was initially conducted followed by a non-linear analysis where an initial deflection was specified.
Jeevaharan (1997) used NASTRAN in his finite element analyses. PATRAN was used as pre- and post-processors. In the analyses, face element was modelled by shell elements and foam core was modelled by brick elements as shown in Figure 2.19.

McAndrew (1999) used the finite element program, ABAQUS, when calculating the flexural wrinkling strength of flat and lightly profiled sandwich panels. Steel faces were modelled as 4-node shell elements and foam core as 8-node solid elements (Figure 2.20). In the study, he considered only elastic buckling method of analysis. Finite element analysis was found to be an excellent tool to investigate the buckling behaviour of sandwich panels and helped to draw many important conclusions. FEA results showed that flat and satined panels that contain ‘gap’ imperfections have significantly less wrinkling strength compared to flat and satined panels without gap imperfections. For ribbed panels, local buckling was found to be occurring in the flat plates of the panel causing failure to occur at stresses lower than that predicted by the wrinkling formula. Hence, FEA results indicated the need for further research to determine wrinkling stress of such panels.
Figure 2.20 Finite Element Model used by McAndrew (1999)

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2.9 Method of Analysis for Sandwich Panels

Due to the difficulty of obtaining more exact solutions in deriving and solving the differential equations for the deflection of sandwich panels, theoretical analysis of such panels has been reduced significantly in recent times (Zenkert, 1995). The introduction of finite element analysis into sandwich panel design has presented another reliable method of analysis over complicated theoretical analysis. However, for calibrating finite element models, the results from analytical solutions play a very important role. This section presents the method of calculation of stresses and deflections based on theoretical analysis.

2.9.1 Flat and Lightly Profiled Panels

In sandwich panels with flat or lightly profiled faces, the bending stiffness of the faces can be neglected in comparison with the bending stiffness of the sandwich part of the cross-section. Normal engineering beam theory is applicable to the sandwich beam theory with a simple modification that involves inclusion of transverse shear deformation in the derivation of formulae. Total deformation of a sandwich panel consists of two parts: (a) deformation due to bending moments \( w_b \) and (b) deformation due to transverse shear forces \( w_s \).

\[
 w = w_b + w_s
\]

For example, the total deflection of a simply supported sandwich beam subjected to uniformly distributed load is given by:

\[
 w = \frac{5qL^4}{384D} + \frac{qL^2}{8S}
\]

where \( q \) is the uniformly distributed load, \( L \) the span of the sandwich panel, \( D \) the bending stiffness (= \( EI \)), and \( S \) the shear stiffness, which is given by:
where $e$ is the distance between centroids of steel faces, $b$ the width of panel, $G_c$ the shear modulus of core, and $t_c$ the thickness of core. The expressions for stress and strain in a sandwich beam can be given as that of a normal beam as given below:

$$\sigma = \frac{My}{I} \quad (2.9.4)$$

$$\varepsilon = \frac{My}{EI} = \frac{My}{D} \quad (2.9.5)$$

Flexural rigidity $D$ of the sandwich panels is given by the summation of flexural rigidity of the faces ($D_f$) and flexural rigidity of the core ($D_c$).

$$D = D_f + D_c \quad (2.9.6)$$

(a) For Flat Faces

$$D = \frac{E_f bt^3}{6} + \frac{E_c b t e^2}{2} + \frac{E_c b t e^3}{12} \quad (2.9.7)$$

where $E_f$ and $E_c$ are the modulii of faces and core, respectively, $t$ and $t_c$ the thicknesses of faces and core, respectively, and $b$ the width of panel. Substituting flexural rigidity $D$ into Equation 2.9.5 gives:

$$\varepsilon = \frac{12My}{2 E_f bt^3 + 6 E_f b t e^2 + E_c bt e^3} \quad (2.9.8)$$

Now, the bending stresses in faces ($\sigma_f$) and core ($\sigma_c$) can be given by:

$$\sigma_f = \frac{12My E_f}{2 E_f bt^3 + 6 E_f b t e^2 + E_c b t e^3} \quad \text{for} \quad \frac{t_c}{2} < y < \frac{t_c}{2} + t \quad (2.9.9)$$
\[ \sigma_c = \frac{12M yE_c}{2E_fbt^3 + 6E_f bte^2 + E_c bte_c^3} \quad \text{for} \quad 0 < y < \frac{t_c}{2} \]  

(2.9.10)

The thickness of the faces used in Australian sandwich panels is very small compared with the thickness of the core (i.e. \( t << t_c \)). Similarly, modulus of elasticity of the core is negligible compared to the steel faces (i.e. \( E_c << E_f \)). So the expressions for flexural rigidity and bending stresses can be simplified as follows:

\[ D = \frac{E_f bte^2}{2} \]  

(2.9.11)

\[ \sigma_f = \frac{2M y}{bte^2} \quad \text{for} \quad \frac{t_c}{2} < y < \frac{t_c}{2} + t \]  

(2.9.12)

\[ \sigma_c = \frac{2M yE_c}{E_f bte^2} \quad \text{for} \quad 0 < y < \frac{t_c}{2} \]  

(2.9.13)

The maximum face stress can be derived from Equation 2.9.12 and is given by:

\[ \sigma_f = \frac{2M}{bte^2} \left( \frac{t_c}{2} + t \right) \]  

(2.9.14)

The distance between the centroids of the faces, \( e = t + t_c \). By taking \( t + t_c/2 \approx e/2 \), the maximum face stress is given by Equation 2.9.15 where \( A_f \) is the area of cross-section of each face.

\[ \sigma_f = \frac{M}{bte} = \frac{M}{A_f e} \]  

(2.9.15)

(b) For Lightly profiled Faces

\[ D \approx \frac{E_f A_f e^2}{2} \]  

(2.9.16)
where $A_f$ is the area of cross-section of profiled face. The same assumptions of thin faces ($t << t_c$) and weak core ($E_c << E_f$) as taken before in Equation 2.9.11 is considered for Equation 2.9.16. Face stress for lightly profiled panels is given by Equation 2.9.4,

$$\sigma_f = \frac{M_y E_f}{D}$$ (2.9.17)

Assuming that $e/2$ is the distance from the centroid of the entire panel to the centroid of an individual face, maximum face stress can be given by:

$$\sigma_{f(max)} = \frac{M_{y_{max}} E_f}{D} = \frac{Me E_f}{2D}$$ (2.9.18)

After substituting the value of flexural rigidity $D$ from Equation 2.9.16 into Equation 2.9.18, following expression for the maximum face stress is obtained:

$$\sigma_f = \frac{M}{A_f e}$$ which is the same Equation 2.9.15 derived for flat panels.

![Figure 2.21 Simply Supported Beam with Uniformly Distributed Suction](image)

For the case of a simply supported sandwich panel under a uniformly distributed loading in the form of suction $q$ as shown in Figure 2.21, the following derivations for flat and lightly profiled panels which include the self weight $g$ of the panel are given and are directly related to Equation 2.9.15. The expression for moment at point $X$ is given by:
\[ M = (q + g) \frac{L}{2} X - (q + g) \frac{X^2}{2} \] (2.9.19)

Replacing \( X \) by \( L/2-x \), the above equation reduces to

\[ M = (q + g) \frac{L^2 - 4x^2}{8} \] (2.9.20)

Substituting the value of \( M \) into Equation 2.9.15, face stress can be expressed as:

\[ \sigma_f = \frac{M}{A_f e} = \frac{(q + g)(L^2 - 4x^2)}{8A_f e} \] (2.9.21)

Equation 2.9.21 gives the maximum face stress at any cross-section in the panel. At mid-span, the stress will be:

\[ \sigma_{f(midspan)} = \frac{(q + g)L^2}{8A_f e} \] (2.9.22)

Martikainen and Hassinen (1996) gave alternative equations for stress in the face of sandwich panel by simply replacing \( q \) with \( 2R/L \) where \( R \) is the support reaction.

\[ \sigma_f = \frac{(2R + gL)(L^2 - 4x^2)}{8A_f eL} \quad \text{(stress at location of wrinkle)} \] (2.9.23)

\[ \sigma_{f(midspan)} = \frac{(2R + gL)L}{8A_f e} \] (2.9.24)

### 2.10 Various Failure Modes of Sandwich Panels

As explained earlier, local buckling of profiled sandwich panels, flexural wrinkling of flat and lightly profiled sandwich panels, and mixed mode buckling of lightly profiled sandwich panels are some of the important failure modes. In addition to
these, a number of alternative modes of failure must be considered in the design of sandwich panels (Davies, 1993; CIB, 2000) and they include: (a) yielding of the metal faces, (b) shear failure of the core, (c) shear failure of the profiled face layer, (d) crushing of the panel at a point support or at a point or line load, (e) failure of the fasteners, (f) failure of the panel at a point of connection, (g) the attainment of a specified limiting deflection, and (h) blistering.

2.10.1 Yielding of the Metal Faces

The yield stress $f_y$ of the face material shall be taken as the guaranteed minimum value for the metal quality used according to the appropriate standard. Alternatively, it may be determined by testing in the laboratory. The tensile and compressive stress in the face shall satisfy the following equation:

$$\sigma_{Fd} \leq f_y / \gamma_M$$

(2.10.1)

where $f_y$ is the yield stress of the face material and $\gamma_M$ the material factor defined for the yielding failure mode. Sandwich construction introduces no additional considerations for tensile yield, and it is merely necessary to compare the calculated stresses with the yield stress after introducing the relevant load and material factors.

For flat and lightly profiled faces sandwich panels, compressive yield is rarely critical, as the wrinkling stress is generally significantly lower than the yield stress. For the profiled sandwich panels, yield of the outer part of the section may occur if the width to thickness ratio does not exceed certain limit. However, if the outermost plate element in a panel is in compression and its $b/t$ ratio exceeds $1.27 \sqrt{E_f / f_y}$, the panel may fail due to the effect of local buckling (CIB, 2000). Hence, compressive strength of the profiled faces depends on the yield stress of the face material, the $b/t$ ratio of the most stressed plane part of the profile, and the compressive and shear stiffness of the core material. CIB (2000) recommends that compressive strength of the profiled faces is determined either by using full-scale tests or by calculations based on the effective width approach.
2.10.2 Shear Failure of the Core

The only significant stresses in the foam core of a sandwich panel, apart from local influence near point or line loads, are the shear stresses necessary to obtain composite action between the two metal faces. The resulting shear stresses are very nearly uniform over the depth of the core and readily calculable. In the design process, the shear stress in the core has to satisfy the following equation (CIB 2000):

\[ \tau_{Cv} \leq f_{Cv} / \gamma_M \quad (2.10.2) \]

where \( f_{Cv} \) is the characteristic shear strength of the core material and \( \gamma_M \) the material factor defined for the shear failure in the core layer.

Due to inherent complicating factors associated with the foam properties of sandwich panels and the observation that foam properties are dependent on the particular machinery used and on the operating conditions, the characteristics of the shear strength of the core material should be determined by testing. The ultimate limit state of shear failure of the core may be determined using the average calculated shear stress at the section of maximum shear force (CIB, 2000). It must be verified that the adhesive will not fail before failure of the core itself, if the panels are produced using adhesives. If the panels are produced using discrete core materials with transverse joints, the shear strength must be assessed using a test on a complete panel with the most unfavourable position of joints.

There are a number of suitable test procedures to determine the shear strength and shear modulus of the core material. These methods include single span-dynamic, 4-pinned square, single span-static, and lapped test. It is important to note that results obtained from the standard tests based on lapped arrangements tend to be unduly pessimistic when used as design values. Basu (1976) determined the shear modulus of some continuously formed material by four alternative methods. The results obtained from these test methods are shown in Figure 2.22. During the test, the material had an average density of 50 kg/m³ (46 kg/m³ at the centre, 65 kg/m³ at the faces) and the results are summarised in Table 2.2.
Table 2.2 Shear Modulus of the Foam Core obtained from various Test Methods

<table>
<thead>
<tr>
<th>Dimensions of Test Specimens</th>
<th>Test Method</th>
<th>$G_c$ (N/mm$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$35 \times 110 \times 2000 \text{ mm}^3$</td>
<td>A (single span-dynamic)</td>
<td>5.5</td>
</tr>
<tr>
<td>$35 \times 35 \times 20 \text{ mm}^3$</td>
<td>B (4-pinned square)</td>
<td>4.6</td>
</tr>
<tr>
<td>$35 \times 140 \times 1000 \text{ mm}^3$</td>
<td>C (single span-static)</td>
<td>4.3</td>
</tr>
<tr>
<td>$35 \times 105 \times 420 \text{ mm}^3$</td>
<td>D (lapped test)</td>
<td>2.4</td>
</tr>
</tbody>
</table>

European recommendations for sandwich panels (CIB, 2000) recommends that single span-static (four point bending) tests using specimens cut from complete panels are to be preferred to the other tests for design purposes. The shear strength of the core decreases under long term loading. If a sandwich panel is designed to carry long-term permanent loads, reduction in shear strength should be taken into account.

2.10.3 Shear Failure of the Profiled Face Layer

In the design process, for a plain or lightly profiled sandwich panels, all of the shear force is considered to be carried by the core whereas for profiled faced sandwich panels, all of the shear force is considered to be carried by either the core or by the faces acting alone. The shear stress in the web of a profiled face layer should satisfy the following equation (CIB, 2000):
\[ \tau_{Fli} \leq f_{Fli} / \gamma_M \]  

(2.10.3)

where \( i = 1 \) and 2 are faces 1 and 2, respectively. The material factor \( \gamma_M \) for the shear failure in a profiled face is defined corresponding to the shear failure in a profiled face. The shear stresses in the webs of the face profile can be calculated using the equations given next:

\[
\tau_{F1} = \frac{V_{F1}}{n_1 s_{w1} t_1} \]  

(2.10.4)

\[
\tau_{F2} = \frac{V_{F2}}{n_2 s_{w2} t_2} \]  

(2.10.5)

where \( V_{F1} \) and \( V_{F2} \) are shear forces in faces 1 and 2, respectively, \( s_{w1} \) and \( s_{w2} \) are lengths of the webs of the profiled faces, \( n_1 \) and \( n_2 \) are numbers of the webs in the profiled faces in the panel width \( B \), and \( t_1 \) and \( t_2 \) are thicknesses of the faces 1 and 2, respectively.

2.10.4 Crushing of Panel at a Point Support or at a Point or Line Load

Points of support give rise to line loads on the panel, and such loads can also arise from other causes. If the loaded face is flat or lightly profiled, this force is mainly resisted by the core aided to a greater or lesser extent by bending of the metal face. For flat faces, a simple approach is to ignore the bending stiffness of the faces and, using a 45° angle of dispersion, to check the compressive stress at the mid-height of the core under factored loads is less than the compressive strength.

Lightly profiled faces can be treated similarly. A less conservative approach for lightly profiled panels is to take account of the bending stiffness of the face by treating it as a beam on an elastic foundation as given next (Davies, 1987a):

\[
\sigma_c = \frac{F \beta}{4} \left[ 1 + e^{-\lambda} (\cos \lambda + \sin \lambda) \right] \]  

(2.10.6)
\[ M_f = \frac{F}{8\beta} \left[ 1 + e^{-\lambda} \left( \cos \lambda - \sin \lambda \right) \right] \tag{2.10.7} \]

\[ \lambda = \beta L_s = \left[ \frac{E_s B}{4t_c E_f I_f} \right]^{1/4} L_s \tag{2.10.8} \]

where \( \sigma_c \) is the compressive stress in foam, \( F \) is the support reaction force, \( M_f \) is the bending moment in the lightly profiled face, \( L_s \) is the support width, and \( t_c \) is the depth of the core.

For profiled faces that are in contact with the support, there is little test information available. The recommended approach, which is conservative, is to ignore the beneficial effect of the core in preventing crippling of the web and to design on the basis of conventional procedures for profiled metal sheeting and decking (ECCS-TC7, 1983a). What little information is available suggests that it is not necessary to consider a reduction in capacity due to interaction between line loads and bending moments when this approach is used (Davies, 1987a).

In designing for crushing of the panel at a support, the compressive stress in the core layer over a support should satisfy the following equation:

\[ \sigma_{cd} \leq \frac{f_{Cc}}{\gamma_M} \tag{2.10.9} \]

where \( f_{Cc} \) is the characteristic compression strength and its value is determined experimentally, \( \gamma_M \) is the material factor for a crushing failure in a core. At a support, the distribution of compressive stresses in the core is taken into account by assuming the support reaction to cause a uniformly distributed stress at mid-depth of the core. The compressive stress at an end and an intermediate support are (CIB, 2000):

\[ \sigma_{cd} = \frac{F}{B(L_s + ke / 2)} \text{ (at end support)} \tag{2.10.10} \]

\[ \sigma_{cd} = \frac{F}{B(L_s + ke)} \text{ (at intermediate support)} \tag{2.10.11} \]
where \( k \) is a distribution parameter, \( e \) is the distance between the centroids of the face layers. Other terms are the same as defined before. Value of \( k \) is determined experimentally. In the absence of experimental value, \( k \) may be taken as 0.5 for rigid plastic foam. For sandwich panels with \( e > 100 \) mm, \( e = 100 \) mm should be used.

### 2.10.5 Failure of the Fasteners

Failure of fasteners in sandwich panels occurs either in tension or shear. Fastenings at supports are loaded by tensile forces caused by wind uplift loads and temperature differences between the faces of the panel. Fastenings are loaded by shear forces caused by self-weight of the panels and by the weight of additional building components on a wall and roof, by the temperature expansion of faces and further, by diaphragm action. Hence, two different types of failure modes have to be considered in dealing with the failure of the fasteners, i.e. failure mode of fastener loaded in shear and failure mode of fastener loaded in tension.

(a) Failure mode of fastener loaded in shear

CIB (2000) recommends that the same failure mode can arise as for fastenings in sheeting as described in the European Recommendations for the design and testing of connections in steel sheeting sections (ECCS, 1983a). The modes of failure include (Figure 2.23): (i) Shear failure of the fastener, (ii) Tilting of the fastener with folding of the inner face of the panel, and (iii) Bending of fastener due to imposed deformation, \( u \).

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**Figure 2.23 Failure Mode of Fastener Loaded in Shear (CIB, 2000)**
(b) Failure mode of fastener loaded in tension

Similar failure modes can arise as for fastenings in sheeting as described in ECCS (1983b). These include (Figure 2.24): (i) Tension failure of the fastener and (ii) Pull-out of the fastener by disturbing the thread in the substrate.

![Failure Mode of Fastener Loaded in Tension](CIB, 2000)

Figure 2.24 Failure Mode of Fastener Loaded in Tension (CIB, 2000)

### 2.10.6 Failure of the Panel at a Point of Connection

The failure mode involved with fastening of sandwich panels systems is often complex involving distortion of the upper face and the core. In fact, failure of fasteners itself in either tension or shear is a relatively remote possibility and is readily checked using values given in the manufacturers’ trade literature. The most common failure associated with fasteners is failure of the panel itself at the point of connection. Design in this instance must be based on test results, as calculation becomes extremely difficult due to the complex failure mode. Indeed, with panels subject to high wind suction forces, this matter can be the most critical part of the design process. Failure modes of the sandwich panels at the point of connection can be grouped into two depending on whether the fastening is loaded in shear or tension.

(a) Failure mode of panel at connection loaded in shear

Two types of failure modes of a panel can be observed at the connection loaded in shear as shown in Figure 2.25. They include: (i) Yield of inner panel sheet only and (ii) Yield of inner panel sheet and/or supporting structure.
Five different types of failure modes of a panel can be observed at the connection loaded in tension as shown in Figure 2.26. They include: (i) Pull-over of the outer face of the panel, (ii) Delamination of the inner face, (iii) Pull-out of the inner face, (iv) Peeling of the inner face, and (v) Failure of the core.
2.10.7 The Attainment of a Specified Limiting Deflection

In calculating the deflections of sandwich panels (e.g. wall or roof), both short-term and long-term shear deformations of the core should be taken into account. In special cases, the deflection caused by local deformations of the core over the supports and by the flexibility of the fastening systems may also be of sufficient importance to be taken into account in calculations. The short-term deflection denotes the combination in which no creep effects are included. It may include deflections caused by both short-term and long-term loads and, therefore, it represents the initial deflection. The long-term deflection consists of the short-term deflection plus the additional deflection caused by the shear creep. It is not correct to associate the short-term deflection with deflection caused by the short-term loads only and the long-term deflection with deflection caused by the long-term loads only.

CIB (2000) recommends the following deflection limits for sandwich panels. For roof panels and ceilings, the deflection caused by the short-term loads should not exceed the value of span/200. Correspondingly, the long-term deflection of roof panels and ceilings including the effects of creep should not exceed the value of span/100. For wall panels, the deflection should not exceed the value of span/100.

2.10.8 Blistering

Blistering is a phenomenon in which local areas of the exposed face of sandwich panel separate from the core. It may affect the outer faces of dark coloured panels exposed to sunlight or other panels subject to relatively high temperatures. Blisters are usually associated with imperfect bond between the core and the faces or voids or imperfection during manufacture. Blistering is of random occurrence and is evidently not caused by the primary stresses resulting from either uniformly distributed or temperature load on the panel. The most likely cause would appear to be a build-up of gas pressure in the cell structure of the foam core due to high temperatures on the outside face. It appears that, at a normal operating temperature (< 80°C), formation of a blister requires some flaw in core or local bond weakness (Davies, 1987a).
It has been claimed by the German firm of Hoesch Siegerlandwerke AG, who have a continuous foaming line and a very high level of quality control, that blistering is often associated with a breakdown in quality control. After stringent quality assurance tests were introduced in the manufacturing process, the blistering problem was completely eliminated (Davies, 1987a).

European Recommendations for sandwich panels (CIB, 2000) recommends that the test to examine the blistering of sandwich panels should be conducted by heating the outer face of the panel up to a uniform temperature of $85^\circ C \pm 3^\circ C$ and then maintaining the panel at that temperature for two hours. The panel should then be carefully examined for visible blisters before it is allowed to cool.

2.11 Durability of Sandwich Panels

Sandwich panels are structural building components and therefore should have an expected life under normal environmental conditions of the order of 20 years. This clearly indicates that durability of sandwich panels becomes one of the important factors in the design process.

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Figure 2.27 Change of Degradation Factor during the Service Life of Sandwich Panels (CIB, 2000)

The assessment of durability takes into account the long-term reduction in the strength of the sandwich element. Therefore, durability is defined in terms of the loss of strength properties as the result of accelerated degradation. The critical factors for durability are considered to be the loss of strength in the core itself and its bond with
the faces. The “degradation factor” is defined as the ratio of the tensile strength of the complete sandwich after a suitable ageing regime to that before ageing (CIB, 2000). The degradation factor thus becomes a design parameter in any structural sandwich element, which suffers significant loss of strength with age. Possible degradation of the strength of sandwich panels with time is given in Figure 2.27. The durability of the strength of sandwich panels is affected by variations in moisture and temperature. Until now, very little is known about the durability of various core materials in current use. Just (1983) investigated the durability problem associated with the core and its bond with metal faces both by observing test panels in situ out of doors over a number of years and by accelerated laboratory tests. The summaries of his results are as follows:

(a) For the polyurethane based rigid foam cores considered, the ageing behaviour with regard to the mechanical properties is adequately stable in a temperate climate provided that the core is relatively protected from direct weathering.

According to Just (1983), there must be an adequate seal at the panel joints so that the core is protected from moisture and humidity. He also considers that increasing degradation of the core can take place when temperatures exceed about 85°C and therefore that careful choice of the surface coating is essential in panels exposed to direct sunlight. His experiment included the measurement of the maximum surface temperature of panels with a variety of surface coatings in Dresden in 1978 and 1980. Most of these panels had aluminium faces and the highest recorded temperature was 101°C on a non-anodised sheet with no surface protection. With colour coatings, grey aluminium reached 78°C and blue steel 72.5°C.

(b) The adhesion between the core and the metal faces is stable, provided that the surface preparation of the metal is adequate. In Table 2.3, Just (1983) summaries his estimate of the resistance of the adhesion joint to climatic loading.

The durability of the adhesion joint depends not only on the method of priming the metal surfaces as claimed by Just but also on its temperature at time of foaming, the method of manufacture, and the quality of the foam (Davies, 1987a). As mentioned before, durability of sandwich panels is taken into account in the design process by
means of “degradation factor” which is determined by ageing of core material. To obtain accelerated ageing of sandwich panels with rigid plastic foam cores, the aging procedure defined in ASTM C 481-62 is recommended. Davies and Heselius (1993), however, have suggested a simple procedure to be sufficient for mineral-wool, which may be extended to other rigid plastic foams. According to them, ageing of the mineral-wool has been found to start when the temperature exceeds about $40^\circ\text{C}$ and, at the same time, the humidity exceeds about 60%. The degradation rate is initially slow but increases as both the temperature and humidity increase. With a temperature of $70^\circ\text{C}$ and 100% humidity, the ageing process is very rapid. In this regime, degradation proceeds quickly and is substantially complete within 24 hours. In tests continued for six months, 60% of the degradation was achieved in the first 24 hours.

<table>
<thead>
<tr>
<th>Surface Skin</th>
<th>Pre-treatment</th>
<th>Adhesion Agent</th>
<th>Supporting Core</th>
<th>Resistance to Climate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminium</td>
<td>Degreasing according to the Na-OH-H-PO process</td>
<td>PVB-washprimer polychloropren adhesive varieties</td>
<td>SH4028 65 kg/m³</td>
<td>Unsatisfactory</td>
</tr>
<tr>
<td></td>
<td>Alkaline or slightly acid degreasing agents</td>
<td>PVB-washprimer</td>
<td>SH4050 50 kg/m³</td>
<td>Unsatisfactory</td>
</tr>
<tr>
<td></td>
<td>Anodising</td>
<td>No adhesive agent HV31+hardener</td>
<td>SH4055 56 kg/m³</td>
<td>Good</td>
</tr>
<tr>
<td></td>
<td>Alkaline degreasing</td>
<td>Epoxy resin</td>
<td>SH4050 50 kg/m³</td>
<td>Good</td>
</tr>
<tr>
<td>Steel</td>
<td>Galvanising</td>
<td>PVB-washprimer</td>
<td>SH4050 50 kg/m³</td>
<td>Unsatisfactory</td>
</tr>
<tr>
<td></td>
<td>Chromatisation</td>
<td>Epoxy resin</td>
<td>SH4050 50 kg/m³</td>
<td>Good</td>
</tr>
<tr>
<td></td>
<td>Phosphatisation</td>
<td>Epoxy resin</td>
<td>SH4050 50 kg/m³</td>
<td>Good</td>
</tr>
<tr>
<td></td>
<td>Galvanising</td>
<td>Epoxy resin</td>
<td>SH4050 50 kg/m³</td>
<td>Good</td>
</tr>
</tbody>
</table>
To determine the degradation factor, a standard tension test as shown in the Figure 2.28 is carried out. Tensile strengths are determined for both aged and unaged specimens cut from production panels with the faces intact. For ageing, the specimens with the face in place are kept for 24 hours above water level in a closed box which is filled to one-third of its height with water maintained at $70^0 \text{C} \pm 3^0 \text{C}$. They should then be tested as soon as possible after ageing.

$$\text{degradation factor} = \frac{\text{ultimate tensile strength of an aged specimen}}{\text{ultimate tensile strength of an unaged specimen}}$$

In relation to the ageing test, there are several factors, which need to be considered in design. Since the test is conducted at severe regime of temperature and humidity, this may not necessarily represent the conditions in service. Also it may be possible that panels may be subject to high level of temperature and humidity causing to consider appropriate design penalty. With the proposed test, it is very difficult to achieve a degradation factor greater than about 0.85 especially with mineral wool panels.

Considering all these factors, Davies and Heselius (1993) suggested that if the degradation factor is greater than 0.7, no reduction in strength need to be applied. If the degradation factor is less than 0.7, all characteristic strength values which are influenced by the core properties should be multiplied by a reduction factor, equal to the degradation factor minus 0.3. CIB (2000) recommends that the tensile bond strength, defined as the characteristic tensile strength value of the unaged core and bond on the total cross-section, should be at least 0.075 N/mm$^2$. 

Figure 2.28 Standard Tension Test (Davies and Heselius, 1993)
2.12 Influence of Temperature on Sandwich Panels

One of the important characteristics of sandwich panels is the high degree of thermal insulation provided by the plastic foam cores. It is not unusual to experience high temperature difference between the inside and outside wall surfaces in extreme climatic conditions. Thermal bowing of the panel occurs due to the temperature differences between the steel faces. In summer, the outside face is exposed to extreme sunlight creating a high temperature difference between the faces of the panels. Similarly a temperature difference arises in winter when the outside face is exposed to cold air and the inside face remains at the ambient temperature of the inside of the building. As a result of the flexural rigidity of steel facings, thermal stresses are present even for a simple span condition (Chong and Hartsock, 1993).

CIB (2000) recommends that if the “National Standards” do not give values for the temperatures of the faces of the element, the following values for the temperature of the outside face may be used: the temperature $T_1$ of the outside face has a minimum winter value of $-10^\circ C$ in maritime climate (UK), $-20^\circ C$ in Central Europe and $-30^\circ C$ in the Nordic countries. The temperature of outside face of a roof panel with an overlying snow load is $0^\circ C$.

The temperature $T_1$ of the outside face has a maximum summer value, which depends on the colour and reflectivity of its surface. For ultimate limit state calculations, $T_1 = 80^\circ C$ for all colours. For serviceability calculations, $T_1$ may be taken as follows:

(i) very light colours, $R_G = 75 - 90$, $T_1 = +55^\circ C$
(ii) light colours, $R_G = 40 - 74$, $T_1 = +65^\circ C$
(iii) dark colours, $R_G = 8 - 39$, $T_1 = +80^\circ C$

where $R_G =$ degree of reflection relative to magnesium oxide = 100%. The value of $R_G$ can be obtained from data provided by the manufacturer of coating. In special cases, the value of $R_G$ may be determined by testing. The appropriate testing method and interpretation of test data can be found in ECCA-T3 1985 and ASTM D2244-93.

The surface temperature $T_1$ for the specific coating may be interpolated using:
\[ T_1 = 55^\circ C \quad \text{for} \quad R_G \geq 75 \]
\[ T_1 = 65^\circ C - \frac{R_G - 40}{35} \times 10^\circ C \quad \text{for} \quad 40 \leq R_G < 75 \]
\[ T_1 = 80^\circ C - \frac{R_G - 15}{25} \times 15^\circ C \quad \text{for} \quad 15 \leq R_G < 40 \]
\[ T_1 = 80^\circ C \quad \text{for} \quad R_G < 15 \]

In general, the temperature $T_2$ of the inside face may be taken as $+20^\circ C$ in winter and $+25^\circ C$ in summer for both ultimate limit state and serviceability limit state calculations. For buildings with air conditioning, cold stores, etc. where a defined ambient temperature is maintained, $T_2$ is ambient operating temperature. In Australia, it is recommended that dark coloured panels are avoided (McAndrew, 1999).

As stresses and deflections arising from temperature loading are readily calculable, the temperature load must be considered in design in conjunction with other relevant load cases. Davies (1987a) has pointed out that there are three distinct cases to consider from the point of view of structural design. They are summarised next:

- For simply supported panels with flat or lightly profiled faces, there is significant deflection but no significant stresses due to temperature gradient. The panel bends into the arc of a circle.

- For simply supported panels with profiled faces, an internal stress system is set up with equal and opposite bending moments in the sandwich part and flange part of the cross-section. There are also significant shear forces near the end of the panel. The panel bends into an approximate circular arc.

- For panels that are continuous over two or more spans, the tendency to bend into a circular arc is prevented by the internal supports with the result that large support forces occur. These give rise to bending moments and shear forces, which are much larger than those arising in simply supported panels. In extreme cases, it is possible for failure to occur because of thermal effects alone.
2.13 Influence of Creep on Sandwich Panels

As the expected service life of the usual building components varies between 20 and 60 years, long-term behaviour and strength are very important design criteria for building structures. The plastic foams tend to creep under long-term loads, i.e., the deformations increase due to the time even if the foam is loaded by constant stresses. Although creep has no influence on wall panels, special attention has to be paid to long-term problems in the design of roof panels. Creep causes both increased deflections and a redistribution of internal stress resultants. A full theoretical treatment is both complicated and, from the practical point of view, hardly necessary (Davies, 1987a). Therefore, in design it is sufficient to calculate creep for long-term loads using a reduced value of the shear modulus $G_C$ of the core.

Effect of creep in sandwich panels can be demonstrated by an example of simply supported panel loaded by a uniformly distributed load (Kilpelainen and Hassinen, 1995b). Mid-span deflection of such a panel can be given as the sum of the bending deflection and the shear deflection as shown next.

$$w(t) = w_b + w_s(t)$$
$$w(t) = w_b + w_{s0}[1 + \phi(t)]$$
$$w(t) = \frac{5}{384} \frac{qL^4}{B_i} + \frac{1}{8} \frac{qL^2}{S}$$

where $q$ is the load, $L$ is the span, $B_i$ is the bending stiffness, $S$ is the shear stiffness, and $\phi(t)$ is the creep coefficient. The shear stiffness $S$ depends on the shear modulus of the core layer and on the cross-sectional dimensions of the sandwich beam and is given by

$$S = \frac{e^2 b G_c}{h_c}$$

where $e$ is the distance between centroid of the face layers, $b$ is the width and $h_c$ is the depth of the core of the sandwich beam. As shown in Equation 2.13.1, the
deflection caused by the bending moment does not depend on time as the steel faces can be assumed to be elastic. However, the second part depends on the shear deformations and tends to increase in the course of time because of the shear creep in the core layer. Therefore, the shear creep has to be precisely known in the design of sandwich panels with plastic foam cores. The shear modulus of the core for long-term loading is a function of the creep coefficient $\phi(t)$ and is expressed as

$$G_{Ci} = \frac{G_c}{1 + \phi_t}$$

(2.13.3)

To define the shear creep properties of the plastic foam core used in sandwich panels, many researchers have conducted long-term experiments in the laboratories. Just (1983) conducted long-term creep tests on sandwich beams with plain metal faces and a variety of polyurethane foam cores. These tests were sustained for about 10 years. Some of the important conclusions drawn from his work are (Davies, 1987a):

- The foam was still creeping after 10 years.

- Although there was some scatter of results, the creep function was approximately linear on the double logarithmic scale and all results fall within a relatively narrow band.

- A reasonable upper bound formula for the creep coefficient is given by:

$$\phi_t = 0.12t^{0.36}$$

(2.13.4)

- Just found that, under normal conditions, at least 50% of creep is reversible but that the speed of recovery is smaller than the creeping speed.

Based on his work, it is proposed that the following conservative values of the creep function $\phi_t$, which gives significantly increased deflections, may be used in the design of panels with polyurethane foam cores.
Kilpelainen and Hassinen (1995b) have reported in their paper that an extensive series of long-term tests was started in 1981 in several laboratories in Finland to determine shear creep coefficient for the expanded polystyrene foam (EPS) core. The experimental study consists of two series of long-term test. The first series included 35 full-scale one- and two-span panels with different panel depths, cross-sections, span lengths and loads. The test was completed in 1993 by loading the panel up to the ultimate failure state. The duration of the test was more than 100,000 hours, which is more than 11 years. The second test series consisted of EPS foam sandwich panels loaded by a high shear stress for a period of 8000 hours, which is about one year. The density of EPS core in the test panels was approximately 20 kg/m$^3$. From that study, the following conclusions were drawn (Kilpelainen and Hassinen, 1995b).

- Mechanical properties of EPS foams depend mainly on the density, which should always be taken into account when studying the property tables.

- The shear creep coefficient recommended to be used in the structural design of EPS foam with a density of 20 kg/m$^3$ can be expressed as

$$\phi_t = 0.195t^{0.177} \quad (2.13.5)$$

- The recommended creep coefficient for the EPS foam (20 kg/m$^3$) sandwich panels to be used in the practical design work is

Permanent loads \hspace{1cm} t = 100,000h, \hspace{0.5cm} \phi_t = 1.5

Quasi-permanent (snow) loads \hspace{1cm} t = 2,000h, \hspace{0.5cm} \phi_t = 0.75

- The long-term behaviour of multi-span sandwich beams can be evaluated at the serviceability limit state on the basis of the creep coefficient defined from the single span panel tests.
If the shear stress in the core is lower than 40% of the initial shear strength, creep is linear. For shear stresses higher than 40% of the initial shear strength, experimental creep is found to become non-linear.

CIB (2000) recommends that the reduced value of shear modulus $G_{Ct}$ should be determined for a time period of 2000 hours for snow load in Central Europe and 100000 hours for permanent actions (dead load). In the absence of test results, creep coefficient can be taken as

For rigid plastic foams (PUR, EPS, XPS): $\phi_t = 2.4$ for $t = 2000$ hours

$\phi_t = 7.0$ for $t = 100,000$ hours

For mineral wool:

$\phi_t = 1.0$ for $t = 2000$ hours

$\phi_t = 2.0$ for $t = 100,000$ hours

If the creep coefficient $\phi_t$ is less than 0.5, creep effects can be neglected in thin faced sandwich panels, i.e., in panels with flat or lightly profiled faces.

### 2.14 Complicated Behaviour of Core Material

It is explained earlier that a sandwich panel represents an excellent example of two dissimilar materials acting together. However, design of sandwich panels becomes complicated because of their different material properties. The face materials are well known, and their use in sandwich construction does not introduce any new material problems. However, core materials are chemically complex and incorporate additives such as expanding agents and fire retardants, which modify the cell structure and therefore the physical behaviour. Davies (1987a) has pointed out that the fundamental structural parameters of foam relate approximately to the density. Typical values of strength and modulus of elasticity of a polyurethane foam core as a function of density are given in Figures 2.29 and 2.30, respectively.
Although the properties shown in the above graphs may be adequate for preliminary design, there are a number of factors that complicate the situation. In particular, the density is not usually constant over the cross-sectional area of the panel and the properties are not necessarily the same in all directions because they are dependent on the orientation of the cell structure. The precise foam properties are dependent on the particular machinery used and the operating conditions (Davies, 1987a).

The mechanical properties of rigid foam cores are both temperature and humidity dependent. One consequence of this is that tests on core material should be carried out under controlled environmental conditions. Various standards generally specify that specimens should be stored and tested with temperature $23 \pm 1^\circ C$ and relative humidity $50\% \pm 2\%$. 

Figure 2.29 A Typical Relationship between Strength and Density of Rigid Polyurethane Foam (Davies, 1987a)

Figure 2.30 A Typical Relationship between Modulus of Elasticity and Density of Rigid Polyurethane Foam (Davies, 1987a)
2.15 Fire Resistance Behaviour

Sandwich panels with polystyrene, polyurethane, and polyisocyanurate foam cores have been widely used to cover facade walls, roofs, internal ceiling, and partition walls. But the behaviour and the resistance of these foam cores in a fire are seen to be problematic for the effective use of panels. Sandwich panels with mineral wool as a core material, which is non-combustible, are considered suitable in situations where performance in fire is a design criterion (Davies, 1993). However, mineral wool is less durable than the rigid plastic foams and may be unsuitable for use as a structural sandwich panel. The required fire resistance level for the structure depends on the activities inside the building and its locations. Kilpelainen and Hassinen (1995a) have reported that, for a polystyrene core (EPS), the highest service temperature is around 80 – 90°C. At a temperature of 100°C EPS foam starts to shrink, which soon leads to loss of strength of the panel. The EPS-foam core meets fire requirements given for class B1 in the German standard DIN 4102, Part 1 (Kilpelainen and Hassinen, 1995a). In the past, few tests have been conducted to investigate the fire resistance behaviour of sandwich panels. Some of these tests include:

- Small-scale laboratory tests carried out in Germany and reported by Berner (1978) and Jungbluth and Berner (1986).

- A large-scale demonstration test carried out by the German firm, Hoesch Siegerlandwerke AG. This test was filmed and also has records of several actual fires in buildings clad with sandwich panels (Davies, 1987a).

- Tests on four-layer EPS-foam sandwich panels conducted in different parts of Europe and reported by Kilpelainen and Hassinen (1995a).

Davies (1987a) derived some important conclusions based on the tests conducted by various past researchers. These conclusions are as follows:

- Metal faces prevent the fire from directly reaching the core material and also prevent this material from obtaining sufficient oxygen for combustion. There is,
therefore, no spread of fire within the panels. Nevertheless, it has been recommended that junctions between sandwich panels and fire-separating walls should be fire stopped and that it is desirable that, where possible, joint details between individual panels should be detailed so that they also include fire stops.

- When the surface temperature of the panel reaches about 200°C, adhesion between the core and the face is lost. The core material starts to carbonise from about 300°C. The integrity of the core is therefore quickly lost but only in panels that are in direct contact with the fire.

- The breakdown of the core is accompanied by the production of the pungent grey smoke that is characteristic of polymer foam materials. In the case of polystyrene, this smoke is flammable but in the case of polyurethane and polyisocyanurate it is relatively non-flammable.

- In comparison with the contents of typical buildings, the additional fire load from the foam core material is small.

Several fire research laboratories in Europe have investigated expanded polystyrene foam (EPS) sandwich panels using full-scale wall, roof and room tests. In their paper, Kilpelainen and Hassinen (1995a) have reported some of these tests and findings in detail. These tests are described next.

The fire laboratory of the technical research centre of Finland had performed fire test for the usual three-layer EPS-foam panel. The faces of the specimen were made of steel sheets with a thickness of 0.5 mm and the core layer of EPS foam with a density of 20 kg/m³. The dimensions of the test specimen were 2430 × 2510 mm and the thickness 200 mm. After 8 minutes in a fire, the opposite face of the specimen reached locally a temperature of 180°C. At the same time the mean temperature of the opposite face had increased by 100°C compared with the initial temperature.

The Technical University of Braunschweig in Germany conducted an experiment on the partition test walls made of EPS-foam panels (Kilpelainen and Hassinen, 1995a).
In the specimens, additional fourth layers, chipboards, had been glued between the core and steel face. Thickness of the steel faces and the fourth layers were 0.5 mm and 28 mm. In the tests all the specimens fulfilled the requirements for integrity, insulation and strength for more than 30 minutes. The university also conducted a similar type of experiment on two EPS-foam roof panels in fire. The fourth layers made of chipboards were glued between the core and the lower steel face. In the fire tests the mean deflection of the specimens reached the value of 28 mm and the highest temperature of the upper face a value of 92°C after 40 minutes from the beginning of the tests. The specimens passed the German F30 requirement.

Based on these experiments, Kilpelainen and Hassinen (1995a) have pointed out that the resistance of usual three-layer EPS-foam sandwich panels in a fire is low because of the strong heat flow through the steel face to the core, and the loss of resistance of the panel because of the shrinking of the EPS foam. The fourth layer made of chipboard or gypsum board and glued between face and core delays the temperature increase in the core and shrinking of it and also prevents dropping of the core. Further information regarding the fire behaviour of sandwich panels incorporating a variety of core materials with both metallic and non-metallic facings is given by Dowling and Martin (1981).

2.16 Spanning Capability and Some Technical Aspects of Sandwich Panels

Combining the foam core with relatively thin steel faces significantly improves the bending capacity of the composite sandwich panels. This allows the sandwich panel to span greater distances between the supports. Profiled steel sheeting with a 0.42 mm trapezoidal profile (BHP’s Trimdeck profile) can span up to 2000 mm in a lower wind category region. However, a sandwich panel made of similar profile with a 0.42 mm top steel face combined with a 0.6 mm bottom flat steel face and a 50 mm polystyrene foam core can span up to 4900 mm. When a 100 mm thick polystyrene core is used, the span increases considerably to 7500 mm. This example indicates that the spanning ability of sandwich panels is significantly higher than that of traditional profiled steel roof and wall sheeting systems.
During the construction process, structural sandwich panels are installed using either a simple tongue and groove type connection or by overlapping the still faces. Side lap fixing is necessary during the overlapping to maintain a weather proof seal and secure the overlap especially when the roof is walked on occasionally. Care should be taken not to over-tighten roofing or walling screws as this may deform the sides of the ribs. Over tightening can also damage seals on roof screws creating a potential for leak to occur.

As sandwich panels are extremely versatile elements, their use in construction industry can sometimes cause a greater impact on the environment. To reduce their adverse impact on the environment, it is important that sandwich panels provide long term resistance to weather, corrosion of the faces, and degradation of the core material and adhesive. Currently very little information is known about such environmental issues and further investigation should be undertaken to improve and expand the present knowledge.

### 2.17 Summary of Literature Review

Extensive literature reviews as described in the previous sections have enabled the accumulation of the required knowledge in many aspects that are useful to conduct this proposed research. The following conclusions can be drawn from this study:

- Research and development of sandwich panels began a long time ago with the major applications related to the aircraft industry.

- In recent years, sandwich panels, as composite structural elements, have been widely used in the construction of buildings and other structural systems due to their wide range of advantages.

- Sandwich panels have been well researched in Europe and the USA, however, in Australia little research has been conducted. The increasing use of sandwich panels in buildings has accelerated the research programs in Australia.
Under the action of different loadings such as gravity, wind, snow, and others, sandwich panels experience various types of buckling failures. The buckling failure modes are local buckling of plate elements of fully profiled faces, flexural wrinkling of flat and lightly profiled faces, and mixed mode buckling of lightly profiled faces with interactions between flexural wrinkling and local buckling.

The presence of a flexible plastic foam core has a considerable influence on the strength of a steel compression member and its various buckling modes. They are significantly improved due to composite action.

Theoretical formulation for the various buckling strengths of sandwich panels are determined based on widely accepted energy principles, in which the sandwich panel is considered as a steel plate element on an elastic foundation.

For local buckling of profiled sandwich panels, the method presented by Davies and Hakmi (1990) models the composite action of the structural member using an enhanced buckling coefficient $K$ and a dimensionless stiffness parameter $R$.

Theoretical equations for lightly profiled panels show that rib depths and spacings of ribs can have a very large effect on the wrinkling strength. Manufacturers should ensure that the required rib depth is achieved during the production process.

In addition to various types of buckling failures, there are a number of alternative modes of failure, which must be considered in the design of sandwich panels. Some of these failure modes include yielding of the metal face, shear failure of the core and face layer, crushing of the panel, failure of the fastener, and so on.

Although rigid plastic foams are classified as combustible materials, when they act compositely with metal faces they do not make a significant contribution to the fire risk. In particular, they do not contribute to either the continuation or spread of an existing fire. The resistance of plastic foam sandwich panels in fire
can be improved by introducing an additional fourth layer made of chipboard or gypsum board glued between the core and the face.

- Several numerical methods such as finite strip, finite layer, finite prism, finite difference, and finite element analysis approaches are available for the analysis and design of sandwich panels. However, for more general applications, the conventional finite element method offers the best approach.

**Gap in the current knowledge:**

- For practical design purposes, many researchers have proposed explicit mathematical formulae (e.g., Davies et al., 1991; Davies and Hakmi, 1990; Mahendran and Jeevaharan, 1999) to determine the enhanced buckling coefficient $K$ in sandwich panels with profiled faces. However, all of these formulae are not applicable for the wider range of $b/t$ ratios of plate elements. For a relatively low plate slenderness, these design expressions can be used to predict effective width, but for plates with a high slenderness, they are inadequate and need suitable modifications.

- Extensive research into the wrinkling of flat sandwich panels has been undertaken and the analytical solutions agree well. But the analytical solution for the wrinkling of lightly profiled sandwich panels has been less well developed.

- Currently used flexural wrinkling formula is inadequate for lightly profiled panels with increasing rib/ridge heights and spacing between ribs. For such panels, failure occurs due to the interaction of the two buckling modes, namely local buckling and flexural wrinkling. An extensive series of experimental tests and numerical analysis are needed to modify and/or develop the flexural wrinkling formula for such panels.
3.1 General

Buckling failure modes of sandwich panels can be investigated by using an extensive series of tests or numerical methods including finite element analysis. The overall aim of this project was to investigate various buckling failures such as local buckling, flexural wrinkling and their interactions of sandwich panels with fully profiled and lightly profiled faces. Fully profiled sandwich panels are generally subjected to local buckling failures whereas lightly profiled sandwich panels are subjected to either flexural wrinkling failures or mixed mode type buckling failures due to the interaction of local buckling and flexural wrinkling. In this project, investigations of the buckling behaviour of fully profiled sandwich panels and lightly profiled sandwich panels were conducted in two different stages. In the first stage, an extensive series of tests to investigate local buckling behaviour of foam supported thin steel plate elements as used in fully profiled sandwich panels was conducted followed by corresponding finite element modelling and analyses. In the second stage, flexural wrinkling behaviour and mixed mode type failure behaviour of lightly profiled sandwich panels with increasing depths and spacings of the ribs were investigated using an extensive series of finite element analyses and corresponding experimental studies.

This chapter presents the details of the first stage of experimental program undertaken to investigate the local buckling and postbuckling behaviour of steel plate elements supported by polystyrene foam core as used in a fully profiled sandwich panels. The test method used in this investigation was compression test using a universal testing machine in the Structural Laboratory at QUT. The first stage experimental results helped to understand the local buckling phenomenon of the plate elements and provided benchmark data to calibrate the finite element model presented in the next chapter. Mechanical properties of the foam core and the steel faces of sandwich panels are very important parameters required in the design of sandwich panels for various buckling failures including local buckling. Therefore, in
this chapter, the experimental method of determining these mechanical properties and the experimental values are presented first. The detailed experimental programs for the local buckling tests on foam supported steel plate elements are outlined next. A specially constructed test rig to hold the steel plate specimens for the compression tests are discussed in detail. Experimental results including local buckling and ultimate strengths along with relevant parameters are presented. These experimental results were used to review the current design method for sandwich panels subjected to local buckling effects. Comparisons of experimental results with the predictions made by the current design rule (CIB, 2000) along with the relevant experimental results from past researchers are presented. Finally, an interim design equation developed based on experimental results for the fully profiled sandwich panels subjected to local buckling effects is described in detail.

3.2 Mechanical Properties

3.2.1 Foam Core

The core used in sandwich panels is a thicker and relatively low density material bonded to inner and outer metal faces to provide a composite load bearing panel. This is a principal element of a sandwich panel as it transfers the shear loads between the faces and provides a high bending stiffness. There are various types of core materials that can be used in sandwich construction. They are chemically formulated rigid plastic foams such as polyurethane, polyisocyanurate, expanded polystyrene, extruded polystyrene, phenolic resin or mineral wool. Core material that can be used in sandwich panels must have sufficient strength and stiffness to contribute to the composite action and to enable the panel to adequately carry the design loads. The mineral wool as a core material has been increasingly used in the areas where performance of fire is a design criterion as it is almost non-combustible. However, this is a relatively new material and limited information is available on its performance. Polyurethane and polyisocyanurate core materials are very popular in Europe and have been used for many years. However, Australian sandwich construction uses polystyrene foams as the core material as they are very good insulation materials, easily available and relatively economical. This type of foam
can be produced anywhere and in any shape. Also, it adheres well to the metal surface during the gluing process. As the polystyrene foam was used as the core material in this project, further discussion is limited to polystyrene foam core.

Although the polystyrene foam is commonly used in Australian sandwich panels, its mechanical properties are not readily available. Mechanical properties such as density, Young’s modulus $E_c$ and shear modulus $G_c$ of polystyrene foam are very important for design purposes and for the quality control of the material during the production stage. However, this foam material exhibits very complicated behaviour. The density of the foam is not usually constant over the cross-sectional area of the panel and the properties are not necessarily the same in all directions, because they are dependent on the orientation of the cell structure. Furthermore, foam properties depend on the particular testing method used and the operating conditions. Hence for the design of sandwich panels, foam properties should be determined by testing using recommended testing procedures. European recommendations for sandwich panels (CIB 2000) states that Young’s modulus $E_c$ of the polystyrene foam shall be determined using standard tensile or compression tests. For the shear test to determine the shear modulus $G_c$ of the foam, CIB (2000) recommends the four point bending test method or any alternative methods except the lap test method. Jeevaharan (1997) conducted an extensive series of tests based on the CIB recommendations to determine Young’s modulus $E_c$ and shear modulus $G_c$ of polystyrene foam. In this study, the values of $E_c$ and $G_c$ are adopted directly from the experimental results of Jeevaharan (1997). Brief details of his experimental methods and the results are reproduced here for the sake of completion.

### 3.2.1.1 Young’s Modulus

Young’s modulus $E_c$ of the foam was determined using a standard compression test on cubic specimens with dimensions recommended by CIB documents. European recommendations for sandwich panels (ECCS 1991 and CIB 2000) recommend that specimens of square cross-section be used for the compression tests with the width $b$ no less than 0.5 times the height and no more than 1.5 times the height of the specimen. In any case, width $b$ should not be less than 50 mm to get better results.
Considering these limitations, Jeevaharan (1997) conducted tests on 100 to 150 mm cubic specimens of the polystyrene foam. The foam used was SL grade with a nominal density of 13.5 kg/m$^3$. The schematic diagram of the compression test on foam core is given in Figure 3.1. Since polystyrene foam is not an isotropic material, specimens were tested in all three perpendicular directions (x, y and z). The modulus $E_c$ of the foam was determined using the slope of the compressive stress $\sigma_c$ versus compressive strain $\varepsilon_c$ curve, where $\sigma_c = F_c/b^2$ and $\varepsilon_c = \Delta H/H$. Figure 3.1 shows $F_c$, $b$ and $H$ and $\Delta H$ is the shortening.

Since the variation in Young’s modulus values in three perpendicular directions was found to be very small, the final value was taken as the overall average of all the values. Hence the experimentally measured average value of Young’s modulus $E_c$ of the polystyrene foam core was 3.80 MPa. Table 3.1 shows the details of the test results in all three perpendicular directions for both 100 and 150 mm cubes.

<table>
<thead>
<tr>
<th>Table 3.1 Young’s Modulus Results from Compression Tests</th>
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<tbody>
<tr>
<td>Young’s Modulus</td>
</tr>
<tr>
<td>100 mm Cube</td>
</tr>
<tr>
<td>150 mm Cube</td>
</tr>
<tr>
<td>Average</td>
</tr>
<tr>
<td>Overall Average</td>
</tr>
</tbody>
</table>
There are several methods to determine the shear modulus of the foam core material. Jeevaharan (1997) determined the shear modulus of the polystyrene foam core by using a two point loading test (single shear test) as shown in Figure 3.2. Eight tests were carried out on approximately 100 × 100 × 300 mm or 100 × 100 × 400 mm specimens of foam core. Width $b$ and height $h$ were taken as 100 mm and the length was taken as 300 mm or 400 mm. Four tests were carried out by applying the shear load in the length direction and the other four tests by applying load in the width direction. This provided the shear modulus values in two perpendicular directions. For each increment of the shear load, vertical and horizontal displacements were measured using dial gauges. Vertical displacement was found approximately zero in all of these experiments.

The shear modulus $G_c$ of the foam core material was determined using the slope of shear stress $\tau$ versus shear strain $\gamma$ curve, where $\tau = V/ab$ and $\gamma = d/h$. All of these symbols are shown in Figure 3.2.

The variation in shear modulus values in the two perpendicular directions ($G_{cxy}$, $G_{cxz}$) was found to be very small. Therefore, the foam can be considered as isotropic and the final shear modulus value was taken as the overall average of all the values. Hence the experimentally measured average value of shear modulus $G_c$ of the polystyrene foam was 1.76 MPa. Table 3.2 shows the required shear modulus results from shear tests for both 100 × 100 × 300 mm and 100 × 100 × 400 mm specimens.
Table 3.2 Shear Modulus Results from Shear Tests

<table>
<thead>
<tr>
<th>Shear Modulus</th>
<th>$G_{czz}$ (MPa)</th>
<th>$G_{cxy}$ (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Foam size 100 × 100 × 300 mm</td>
<td>1.79</td>
<td>1.72</td>
</tr>
<tr>
<td>Foam size 100 × 100 × 400 mm</td>
<td>1.79</td>
<td>1.77</td>
</tr>
<tr>
<td>Average</td>
<td>1.79</td>
<td>1.745</td>
</tr>
<tr>
<td>Overall Average</td>
<td></td>
<td>1.76</td>
</tr>
</tbody>
</table>

With the measured Young’s modulus $E_c = 3.8$ MPa and shear modulus $G_c = 1.76$ MPa, Poisson’s ratio was calculated as $\nu_c = 0.08$ from the following equation.

$$G_c = \frac{E_c}{2(1+\nu_c)}$$

(3.2.1)

3.2.2 Steel Faces

Unlike foam core material, faces used in sandwich panels are well known materials and do not introduce any material problems. Commonly used face materials of the sandwich panels are cold-formed steel plates, aluminium sheets, hardboard and gypsum. In Australia, the faces of sandwich panels are generally made of thinner (0.42 mm) and high strength (minimum guaranteed yield stress of G550 MPa and reduced ductility) cold-formed steel plates. In this study, cold-formed steel plates with different grades (G550 and G250) and thicknesses ranging from 0.4 to 1.0 mm were used as the face materials. Hence, further discussion here is limited to the material properties of G250 and G550 grades and different thicknesses of cold-formed steel faces only.

In practice, $E_f$ for steel is generally taken as 200 GPa and the minimum value of $f_y$ is taken as the grade of the steel itself irrespective of different thicknesses. However, past research (Mahendran, 1996) has shown that the Young’s modulus $E_f$ and yield stress $f_y$ values change with the change in steel grades and thicknesses. Jeevaharan (1997) conducted a series of experiments and determined the values of Young’s modulus $E_f$ and yield stress $f_y$ for different grades and thickness of steel. These experimentally measured values were used in this study.
3.2.2.1 Young’s Modulus

Young’s modulus $E_f$ of the steel faces was determined using standard tensile tests on steel plate specimens shown in Figure 3.3. Although standard coupons according to AS1391 – 1991 could be used to determine the Young’s modulus of steel, Jeevaharan (1997) used strips of equal length with strain gauges attached to the centre. This avoided the need to make the special tensile coupons in the tests that were aimed at only determining the Young’s modulus. All the tested specimens were cut longitudinally. 10 mm strain gauges were placed in the middle and both sides of the specimen. The specimens were loaded using Tinius Olsen Testing Machine to approximately 70% of the minimum yield stress.

![Figure 3.3 A Typical Test Specimen to determine Young’s Modulus of Steel](image)

The tensile strain $\varepsilon$ of the steel face material was taken as the average of two strain gauge readings. The tensile stress $\sigma$ versus tensile strain $\varepsilon$ curves were plotted for different grades and thicknesses of steel faces. Although coated specimens were used in the experiments, base metal thickness was used to calculate the cross-sectional area $A$, and hence the tensile stress ($\sigma = F/A$). Young’s modulus $E_f$ for each specimen was calculated from the slope of the corresponding stress-strain curve. The experimentally measured average values of Young’s modulus $E_f$ for different grades (G550 and G250) and thicknesses of the steel faces are summarised in Table 3.3.

3.2.2.2 Yield Stress

Another series of tensile tests were conducted by Jeevaharan (1997) to determine the yield stress $f_y$ of the steel face. For this purpose, standard tensile test specimens, which were prepared according to AS1391 – 1991, were used. The shape and
dimensions of the standard test specimen used by Jeevaharan (1997) is shown in Figure 3.4.

**Table 3.3 Yield Stress and Young’s Modulus Results for different Grades and Thicknesses of Steel**

<table>
<thead>
<tr>
<th>Steel Grades</th>
<th>Thickness (mm)</th>
<th>Young’s Modulus $E_f$(GPa)</th>
<th>Average Yield Stress $f_y$ (MPa)</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>spec.</td>
<td>tct.</td>
<td>bmt.</td>
</tr>
<tr>
<td>G550</td>
<td>0.42</td>
<td>0.46</td>
<td>0.42</td>
</tr>
<tr>
<td>G550</td>
<td>0.60</td>
<td>0.64</td>
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</tr>
<tr>
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<td>0.80</td>
<td>0.84</td>
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<tr>
<td>G550</td>
<td>0.95</td>
<td>0.99</td>
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<tr>
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<tr>
<td>G250</td>
<td>1.00</td>
<td>0.98</td>
<td>0.93</td>
</tr>
</tbody>
</table>

Note: spec. = specified thickness  
tct. = measured total thickness with zinc coating  
bmt = estimated base metal thickness

**Figure 3.4 A Typical Tensile Test Specimen to determine Yield Stress of Steel**

Measured average yield stress $f_y$ for different grades and thicknesses of steel faces are summarised in Table 3.3. As seen from the table it can be observed that the actual yield stress is always higher than the specified minimum guaranteed yield stress.
3.3 Experimental Program to Study Local Buckling Behaviour

3.3.1 Test Specimens

In order to investigate experimentally the local buckling behaviour of sandwich panels, a series of laboratory tests was conducted on the slender plate elements supported by polystyrene foam as used in the profiled sandwich panels. The experiments were essentially compression tests on flat cold-formed steel plate elements with varying grades, thicknesses and $b/t$ ratios.

To prepare the foam supported steel plate test specimens, steel plates of required width $b$ and length $L$ were cut longitudinally from the flat cold-formed steel sheets of known grade and thickness. The length $L$ of the specimen was taken as three times the width $b$ plus additional 5 mm at each end for clamping. The length $3b$ was chosen because it is short enough to eliminate any global or overall buckling of the plates while allowing sufficient number of elastic buckles to occur within such a length. All the cut steel plates were cleaned and dried. Polystyrene foam cores (SL grade with a density of 13.5 kg/m$^3$) were then cut into the required sizes (width $b$ and length $3b$) from the big panels using a hot-wire machine in the laboratory. Flat steel plates and corresponding polystyrene foam cores were glued to each other using a separate adhesive. With the foam manufacturer’s recommendation, the adhesive used in this process was Bostik, a blue colour adhesive. The foam attached steel plates were then kept over a flat surface and pressed for more than 12 hours with a heavy steel plate. This enabled the steel plate and polystyrene foam core to be fully attached without any gaps between them. These specimens were cured for at least 48 hours in room temperature before testing to ensure that the adhesive was set and the steel plate and foam core were joined properly.

In this study two different grades of steel were considered for the local buckling investigation. These two steel grades included a low strength steel grade G250 and a high strength steel grade G550 with minimum guaranteed yield stresses of 250 and 550 MPa, respectively. These two grades were chosen in order to study the applicability of the design formula to be developed to any available grades of steel.
Local buckling failure behaviour of sandwich panels depends on the width to thickness ($b/t$) ratio of the steel plate elements used in the panels. As the $b/t$ ratio increases, the critical local buckling load and ultimate load carrying capacity of the plate element continually decrease. To study this behaviour, foam supported steel plates with varying $b/t$ ratios were used in this experimental investigation. The width to thickness ($b/t$) ratios were varied using different widths and thicknesses of steel plates. The widths $b$ of the steel plates chosen were 50, 80, 100, 120, 150, 180 and 200 mm. Different nominal (specified) thicknesses used for G550 grade steel were 0.42, 0.6, 0.8 and 0.95 mm. Similarly, for G250 grade steel, 0.4, 0.6, 0.8 and 1.0 mm thicknesses were used. The use of wide ranges of thicknesses and widths provided a large range of $b/t$ ratios, thus enabling the investigation of local buckling behaviour of very compact plates to very high slender (very thin) plates. The $b/t$ ratios for G550 grade steel ranged from 52 to 476 while for G250 grade steel they ranged from 53 to 512.

Mahendran and Jeevaharan (1999) and Mahendran and McAndrew (2000) conducted experiments and numerical analyses on foam-supported steel plate elements with different foam thicknesses equal to the plate width $b$, $b/2$, and $b/4$ in order to study the effect of foam thickness on the strength of the panel. They found that foam core thickness has a negligible effect on the strength results provided that the depth of foam core is more than 50mm. Therefore, in this study, a constant foam thickness of 100 mm was chosen in all the experiments irrespective of the plate widths used. The initial imperfections of test specimens relating to the flatness of steel plates were found to be minimal as was the case in most of the fully profiled sandwich panels.

Tables 3.4 and 3.5 show the details of the experimental program and geometrical and mechanical properties of test specimens for G550 and G250 grades of steel, respectively. The mechanical properties were determined by Jeevaharan (1997) and were adopted in this research project as all the steel plates in this research were taken from the same batch of steel. As seen from the tables, 25 foam supported steel plates were tested for each grade of steel, giving a total of 50 tests.
<table>
<thead>
<tr>
<th>Test No.</th>
<th>Plate Width $b$ (mm)</th>
<th>Thickness $t$ (mm)</th>
<th>Spec.</th>
<th>bmt</th>
<th>$f_y$ (MPa)</th>
<th>$E_f$ (GPa)</th>
<th>$b/t$ Ratio</th>
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$f_y$ – measured yield stress of steel, $E_f$ – measured Young’s modulus

$b/t$ ratio – plate width $b/b_{mt}$, Spec. – specified thickness

bmt – estimated base metal thickness based on measured total coated thickness
Table 3.5 Test Program and Specimens for G250 Steel Plates

<table>
<thead>
<tr>
<th>Test No.</th>
<th>Plate Width ( b ) (mm)</th>
<th>Thickness ( t ) (mm)</th>
<th>Measured</th>
<th>( b/t ) Ratio</th>
</tr>
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<tr>
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<td>( f_y ) (MPa)</td>
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<td>326</td>
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<td>326</td>
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</table>

\( f_y \) – measured yield stress of steel, \( E_f \) – measured Young’s modulus

\( b/t \) ratio – plate width \( b/b_{\text{mt}} \), Spec. – specified thickness

\( b_{\text{mt}} \) – estimated base metal thickness based on measured total coated thickness
3.3.2 Test Rig

The sandwich panel can be represented as a simply supported plate subjected to an applied pressure $p$ along the two transverse edges. In other words, the longitudinal edges of sandwich panels are assumed to be simply supported. In order to model the required simply supported boundary conditions along the longitudinal edges, a specially constructed test rig was used to hold the test specimen during the compression test. Furthermore, these simply supported boundary conditions along the longitudinal edges of the plate were designed to simulate the real conditions present on the plate elements of the profiled faces supported by adjoining plates.

It must be noted that the test rig used in this study was specially constructed to test the foam-supported steel plate elements under compression. A steel plate element when subjected to compression load is susceptible to local buckling failure. However, sandwich panels in many structural and building systems are not subjected to compression only but also to bending due to lateral wind pressure loading. It is obvious that when the panel is loaded under bending, one steel face will be in tension and the other will be in compression. Generally, failure does not occur in the tension face as steel plates are strong enough to resist the tensile force. More likely, the failure in sandwich panel occurs due to the local buckling of compression face. This indicates that the sandwich panel always fails by local buckling of the compression face irrespective of whether it is subjected to bending or direct compression. Therefore compression tests using the test rig developed in this research project can be used to represent any steel faces under compression either due to direct compression, bending or a combination of these two. A direct compression test is the simplest way to study the local buckling behaviour of foam-supported plates and hence this method was used in this study.

A complete schematic diagram of the test rig is shown in Figure 3.5 (a). Figure 3.5 (b) shows the photograph of the test rig used in the experiment. The test rig consisted of a base plate and two vertical supports. The two vertical clamps used to hold the steel plates were attached to vertical supports. The vertical supports were adjustable both in horizontal and vertical directions to accommodate the required plate width.
and length, respectively. Plate lengths up to 600 mm can be held between these vertical support edges. Similarly, these vertical support edges can hold the required plate widths ranging from 50 mm to 200 mm. The vertical clamps allowed shortening of the plates and rotation about the vertical edges to occur freely, hence well representing the simply supported conditions of longitudinal edges.

(a) A Schematic Diagram of Test Rig

(b) Photograph of Test Rig

Figure 3.5 Test Rig
Test specimens were placed in the test rig between the two top and bottom loading blocks. These loading blocks facilitated the inclusion of plates with varying widths between vertical supports. Seven different sets of loading blocks as shown in Figure 3.6 were made to satisfy the plate widths of 50, 80, 100, 120, 150, 180 and 200 mm. Each loading block had a groove in the middle to hold the plate specimens firmly.

### 3.3.3 Test Set-Up and Procedure

The compression tests of steel plates were carried out using a Tinius Olsen Testing Machine in the Structural Laboratory at QUT. The foam supported test specimen was first fixed in the test rig between the two loading blocks and placed in the testing machine. The axial compression load was applied to the steel plates via the loading block. Arrangements were made to measure the axial compression load, axial shortening and out-of-plane deflection. The complete arrangement of the test set-up is shown in Figure 3.7.
Out-of-plane deflections were measured by using two linear variable displacement transducers (LVDTs), which were placed on the steel plates in appropriate positions as shown in Figure 3.7. To accurately measure the out-of-plane deflection of the steel plates under local buckling, the placement of the LVDT was very significant. It was very difficult to measure the out-of-plane deflection by these LVDTs in their original form as the point of LVDT may not be placed in a position where the maximum buckling occurs in the steel face. To overcome this problem, a long straight edge made of steel was attached to the pointing edge of the LVDT as shown in Figure 3.7. This 100 mm long straight edge enabled the measurement of the maximum out-of-plane deflection more accurately.

For the axial shortening, Tinius Olsen Testing Machine had an automatic arrangement, which directly measured the resulting displacement. Displacement transducers and load cells were connected to a calibrated Labteck computer data acquisition system. All the measured values of axial shortening, out-of-plane
deflection and axial compression load were continuously recorded by a computer at two second intervals.

A compression load was applied to the foam supported steel plates via the top loading block at a constant rate of 0.5 mm/min until failure of the specimen occurred. It is to be noted that the compression load was applied to the steel plate element only and not to the foam core. For every test, the buckling and ultimate loads carried by the test specimens were recorded. The buckling load was based on the physical observation of plate buckling. The ultimate load was the maximum load carried by the specimen as recorded by the testing machine. Hence the buckling load was approximate, but the ultimate load could be considered exact. No specimen was observed during the test to fail by the delamination of the steel plate and foam core. This confirmed the adequacy of the adhesive agent (Bostik) used in the fabrication of foam-supported plates. Some of the tested specimens are shown in Figure 3.8.

![Some Test Specimens after Failure](image)

**Figure 3.8 Some Test Specimens after Failure**

The test set-up used in this investigation is similar to that used by Davies et al. (1991) at the Technical Research Centre of Finland (VTT) for the investigation of the ultimate strength of compressed steel plates with and without core support. Researchers consider this test set-up using a simply supported plate element as a
simplified model to study the local buckling problem of the faces of sandwich panels (Hassinen, 2003). A similar compression test set-up was also used by Kech (1991) in his investigation to verify the improved equation developed for lightly profiled faces subject to local buckling and wrinkling effects.

### 3.3.4 Experimental Results

As stated earlier, a total of 50 (25 for G550 and 25 for G250) compression tests on foam supported steel plate elements was conducted in this local buckling investigation. This section presents the test results and observations obtained from the tests. It was observed during the tests that all the test specimens failed in a similar manner with the continuous application of compression load. They first buckled locally as shown in Figure 3.9 (a), then developed postbuckling strength, reached the ultimate load and collapsed through the formation of a local plastic mechanism as shown in Figure 3.9 (b). Figures 3.10 (a) and (b) show the typical compressive load versus axial shortening and out-of-plane deflection curves, respectively. Other load versus deflection curves for large \( b/t \) ratios are given in Chapter 4 as Figures 4.10 (a) to (h). After the tests, steel plate and foam core of some specimens were separated to check the bond between them. It was found that a thin layer of foam was continuously attached to the steel face, and no glue was seen at the interface confirming the perfect bond between them.

![Local Buckle](image1.png)  
(a) Local Buckle

![Local Plastic Mechanism](image2.png)  
(b) Local Plastic Mechanism

**Figure 3.9 Typical Local Buckling and Collapse Modes**
(a) Compressive Load versus Axial Displacement

(G550, $b = 180$ mm, $t = 0.60$ mm)

\[ \sigma_{cr} = 8.50 \text{ kN} \]

(b) Compressive Load versus Out-of-Plane Displacement

(G550, $b = 180$ mm, $t = 0.60$ mm)

\[ \sigma_{cr} = 8.50 \text{ kN} \]

Figure 3.10 Typical Load versus Deflection Curves
Table 3.6 Experimental Buckling and Ultimate Strengths Results for G550 Steel Plates

<table>
<thead>
<tr>
<th>Test No.</th>
<th>b/t Ratio</th>
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<th>Local Buckling</th>
<th>Ultimate</th>
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<td></td>
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Table 3.7 Experimental Buckling and Ultimate Strengths Results for G250 Steel Plates

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<th>Stress (MPa)</th>
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Figure 3.9 (a) shows that many half wave buckles were formed within the length of test specimen thus increasing the buckling load. Similar test conducted by Mahendran and Jeevaharan (1999) on steel plates (length = 3b) without foam core showed that approximately three half wave buckles were developed confirming that the half wave buckle length is approximately equal to the plate width. The present investigation on foam supported steel plate elements confirmed that the presence of foam reduced the half wave buckle length (a < b) and produced many half wave buckles within the test specimen. This helped to increase the buckling strength considerably.

The test results of local buckling and ultimate strengths are shown in Table 3.6 for G550 steel plate elements and in Table 3.7 for G250 steel plate elements. These tables include the local buckling load, the local buckling stress, the ultimate load and the ultimate stress for all the foam supported steel plate elements considered in this investigation.

### 3.4 Buckling Coefficient for Foam-Supported Steel Plates

As discussed in Chapter 2 (Literature Review – Section 2.4), sandwich panels subjected to local buckling effects are designed using the concept of effective width. The effective width approach originally developed for plain plate elements is extended to the profiled faces of sandwich panels by modifying the buckling coefficient $K$ to take into account of the core support. In cold-formed steel design, $K$ depends only on the edge boundary conditions. However, for sandwich panels, $K$ depends not only on the edge boundary conditions, but also on the properties of foam core and $b/t$ ratio of the steel plate element. Hence it is obvious that the buckling coefficient $K$ will increase due to the plate stiffening effects of foam core material. $K$ value for sandwich panels can be evaluated by using various buckling formulae. Theoretically $K$ can be determined for profiled sandwich panels by using Equations 2.4.14 and 2.4.18. However, the theoretical evaluation of $K$ is a complicated process and hence many researchers have proposed explicit mathematical formulae to determine $K$ for sandwich panels with profiled faces. These include Equation 2.4.21 proposed by Davies and Hakmi (1990), Equation 2.4.23 proposed by Mahendran and
Jeevaharan (1999), and Equation 2.4.24 included in the current European Recommendations for Sandwich Panels, Part 1: Design (CIB 2000). These equations are reproduced next to facilitate easier understanding of the following discussions.

Davies and Hakmi (1990):

\[ K = [16 + 11.8R + 0.055R^2]^{1/2} \quad \text{with} \quad R = \frac{12(1-v_f^2)\sqrt{E_c G_c}}{\pi^3 E_f} \left[ \frac{b}{t} \right]^3 \quad (3.4.1) \]

Mahendran and Jeevaharan (1999):

\[ K = [16 + 4.76R^{1.29}]^{1/2} \quad \text{with} \quad R = \frac{12(1-v_f^2)\sqrt{E_c G_c}}{\pi^3 E_f} \left[ \frac{b}{t} \right]^3 \quad (3.4.2) \]

CIB (2000):

\[ K = [16 + 7R + 0.02R^2]^{1/2} \quad \text{with} \quad R = \frac{0.35\sqrt{E_c G_c}}{E_f} \left[ \frac{b}{t} \right] \quad (3.4.3) \]

\( R \) is a stiffness parameter which models the influence of composite action between steel faces and foam core. Davies and Hakmi (1990) indicated that Equation 3.4.1 is accurate for a range of \( R \) values from 0 to 200. However, Mahendran and Jeevaharan (1999) showed that this can be extended to higher values of \( R \) up to 600. Hence they proposed Equation 3.4.2 which is applicable for \( R \) values from 0 to 600.

For a comparison study, the \( K \) values evaluated from Equations 3.4.1, 3.4.2 and 3.4.3 proposed by Davies and Hakmi (1990), Mahendran and Jeevaharan (1999), and CIB (2000), respectively, and theory (Equations 2.4.14 and 2.4.18) are given in Tables 3.8 and 3.9 for G550 and G250 steel plates, respectively. In calculating \( R \) and \( K \) values, the experimentally measured material properties such as Young’s modulus \( E_c \) and shear modulus \( G_c \) of polystyrene foam core and Young’s modulus \( E_f \) of steel faces were used (see Section 3.2). These experimental values for foam core are \( E_c = 3.8 \) MPa, \( G_c = 1.76 \) MPa, and \( v_c = 0.08 \). Similarly, the experimental values of Young’s modulus and yield stress for both G550 and G250 grades of steel with different thicknesses were taken from Tables 3.4 and 3.5. The Poisson’s ratio \( v \) for steel plate was taken as 0.3.
### Table 3.8 Comparison of Buckling Coefficients for G550 Steel Plates

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### Table 3.9 Comparison of Buckling Coefficients for G250 Steel Plates

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From the buckling coefficient results it can be observed that, for low $b/t$ ratios, the $K$ values predicted by all three formulae (Equations 3.4.1, 3.4.2 and 3.4.3) are close to the theoretical predictions. But, for higher $b/t$ ratios, Equation 3.4.1 overestimated the $K$ values whereas Equation 3.4.3 underestimated it. On the other hand, $K$ values predicted by Equation 3.4.2 (proposed by Mahendran and Jeevaharan, 1999), are closer to the theoretical values for plate elements with any $b/t$ ratios. From the results it can be concluded that, for low $b/t$ ratios, both Equations 3.4.1 and 3.4.2 can be used to determine the buckling coefficient $K$. However, for slender plates (high $b/t$ ratio), Equation 3.4.2 is more suitable in comparison to Equation 3.4.1.

### 3.5 Comparison of Effective Width Results and Discussions

The widely accepted effective width principle originally developed for plain plate elements and extended to the profiled sandwich panels by modifying the buckling coefficient $K$ as given in Equation 2.4.17 is reproduced here,

\[
\begin{align*}
\text{beff} &= \rho b \\
\rho &= \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] \quad \text{for } \lambda > 0.673 \\
\rho &= 1.0 \quad \text{for } \lambda \leq 0.673 \\
\lambda &= 1.052 \left( \frac{b}{t} \right) \left( \frac{f_y}{E_f K} \right)
\end{align*}
\)

(3.5.1)

where $b_{\text{eff}}$ = effective width of the plate, $f_y$ = yield stress of steel, $E_f$ = Young’s modulus of steel, $t$ = thickness of the steel plate, $K$ = enhanced buckling coefficient.

The values of the enhanced buckling coefficient $K$ obtained from different design formulae as given in Tables 3.8 and 3.9 were utilised to determine the slenderness parameter $\lambda$ using Equation 3.5.1. In this manner, the effect of foam was included in the $\lambda$ expression of Equation 3.5.1. Following this, the effective widths of foam supported steel plate elements were determined using Equation 3.5.1 as for steel plate elements without any foam core support. This effective width approach is the current design rule included in the “European Recommendations for Sandwich
Panels, Part 1: Design” (CIB, 2000) for the design of profiled sandwich panels subjected to local buckling effects. In this approach, no distinction is made between the ductile low strength steels (G250) and the less ductile high strength steels (G550). Therefore one could use Equation 3.5.1 for both G250 and G550 steels of all thicknesses.

However, Clause 1.5.1.5 (b) of AS/NZS 4600 (SA, 1996) recommends that the yield stress of G550 grade steels with a thickness less than 0.9 mm should be multiplied by a reduction factor of 0.75 for design purposes as these steels do not satisfy the ductility criteria needed to ensure satisfactory performance of thin steel members under essentially static load. A recent study conducted by Yang and Hancock (2002) showed that the predicted strength results for G550 steel columns based on 75% of yield stress (0.75f_y) are too conservative. They therefore recommended that a high reduction factor of 0.90 is used for G550 steels with a thickness less than 0.9 mm. Since Yang and Hancock’s (2002) research was not for foam-supported steel plates, the accuracy of their recommendation to foam supported steel plate elements is not known. However, it was considered appropriate to use a reduction factor to allow for possible strength reduction in thinner G550 steel plates. In this study, the modified slenderness parameter λ in Equation 3.5.1 was calculated based on 0.9f_y for G550 grade steels with a thickness less than 0.9 mm to investigate this further. For G550 grade steels with a thickness more than 0.9 mm and for all thicknesses of G250 grade steel actual f_y values were used.

On the other hand, the ultimate stress results obtained from the experiments on foam-supported steel plates as given in Tables 3.6 and 3.7 can be converted to ratios of effective width b_{eff} to plate width b. This ratio was taken as the ultimate stress of foam-supported steel plates divided by the yield stress f_y as given in Equation 3.5.2. Here also a reduced yield stress, 0.9f_y, was used for G550 grade steels with a thickness less than 0.9 mm.

\[
\frac{b_{eff}}{b} = \frac{\text{Ultimate stress of steel plate stiffened by foam}}{\text{Yield stress } f_y \text{ of steel}}
\]  

(3.5.2)
Table 3.10 Comparison of Effective Widths for G550 Steel Plates

<table>
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<tr>
<th>Test No.</th>
<th>$b/t$ Ratio</th>
<th>Ultimate Stress (MPa)</th>
<th>Effective Widths $b_{eff}/b$</th>
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Davies and Hakmi (1990) - Equations 3.5.1 and 3.4.1
Mahendran and Jeevaharan (1999) - Equations 3.5.1 and 3.4.2
CIB (2000) - Equations 3.5.1 and 3.4.3
### Table 3.11 Comparison of Effective Widths for G250 Steel Plates

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Davies and Hakmi (1990) - Equations 3.5.1 and 3.4.1
Mahendran and Jeevaharan (1999) - Equations 3.5.1 and 3.4.2
CIB (2000) - Equations 3.5.1 and 3.4.3
Figure 3.11 Effective Widths of Steel Plates supported by Foam Core
Effective widths $b_{eff}$ evaluated from Equation 3.5.1 using $K$ values determined from various buckling formulae together with corresponding experimentally determined effective widths using Equation 3.5.2 are given in Table 3.10 for G550 steel plates and Table 3.11 for G250 steel plates, respectively. The results show that this study covered a wide range of width to thickness ($b/t$) ratios of steel plate elements to represent every possible situation of sandwich panel construction. This helped to include slenderness parameter $\lambda$ starting from a reasonably smaller value to the largest practical value.

Effective widths evaluated from different design formulae together with the experimental results are plotted against the $b/t$ ratios in Figure 3.11 (a) for G550 steel plates and Figure 3.11 (b) for G250 steel plates, respectively. It is to be noted that foam properties ($E_c$ and $G_c$) were the same for all the tests used in Figures 3.11 (a) and (b).

As seen from Tables 3.10 and 3.11 and Figures 3.11 (a) and (b) for G550 and G250 steel plates, the effective width $b_{eff}$ obtained from experimental results and those evaluated from the effective width approach (Equation 3.5.1) using $K$ values predicted by theory and different buckling formulae vary considerably with the $b/t$ ratios. It can be observed that, for low $b/t$ ratios, the effective width values evaluated from various design equations agreed reasonably well with the experimental results. The effective width approach predicted realistic and acceptable values of effective widths in comparison with the experimental values when $b/t$ ratio was less than 100. However, for higher $b/t$ ratios, all the formulae predicted very high effective width values compared with the experimental results, i.e. unconservative. So for slender plates, none of the current formulae could estimate reasonable values of effective width $b_{eff}$. The use of a reduced yield stress for thinner G550 steel plates only allows for possible strength reductions due to lack of ductility in those plates, and the above observations in relation to slender plates are equally valid for both G250 and G550 steel plates.

This outcome is similar to that of Davies and Hakmi (1992) who used a series of bending tests on foam-filled C-section beams. Figure 3.12 shows the comparison of their test results for effective width with predictions based on design rules given in
CIB (2000). As seen in Figure 3.12, the effective width predictions from the current design formula were higher than those from tests for $b/t$ ratios greater than 150. This clearly demonstrates that the current design rule is adequate for plates with low $b/t$ ratios, but is inadequate for slender plates. In Davies and Hakmi’s tests, the foam properties ($E_c$ and $G_c$) were varied for each $b/t$ ratio and hence the predictions from CIB (2000) (Equations 3.5.1 and 3.4.3) are not shown as a continuous curve as in Figures 3.11 (a) and (b).

As the compression test used in this investigation is a simplified model to study the local buckling behaviour of foam supported plates, it has some drawbacks when compared with the bending test. Simply supported end conditions were used along the longitudinal edges of the foam supported plates in the compression tests. However, the plate elements of profiled faces of sandwich panels receive some rotational restraint from the adjoining plate elements. This will increase the effective width of the plate element thus increasing the ultimate strength of the sandwich panels. Therefore, the results obtained from the simplified compression tests are likely to be slightly conservative.

This shortcoming may be eliminated by conducting bending tests like those undertaken by Davies and Hakmi (1992). However, the trends of the results of the
bending tests of Davies and Hakmi (1992) and the present compression tests are very similar. For example, for a steel plate with $f_y = 281$ MPa and $b/t$ ratio 260 in Davies and Hakmi’s test, the ratio of effective width to actual width ($b_{eff}/b$) was about 0.32 from experiments and 0.42 from Equations 3.5.1 and 3.4.3 (see Figure 3.12). In the present compression test, $b_{eff}/b$ from experiments and Equations 3.5.1 and 3.4.3 were 0.30 and 0.40, respectively, for a steel plate with $f_y = 368$ MPa and $b/t$ ratio 256 (Figure 3.11 (b)). Both test methods gave similar results with the present compression tests being slightly more conservative. Detailed test data of Davies and Hakmi’s experiments (1992) are given in Table 2.1 of Chapter 2 (Literature Review).

It is worth noting here that Equation 3.4.3 included in CIB (2000) to evaluate the buckling coefficient $K$ was a simple modification of Equation 3.4.1 proposed by Davies and Hakmi (1990). To account for non-linear behaviour of foam properties and other material uncertainties, Davies and Hakmi (1990) recommended the use of an empirical reduction factor of 0.6 for $R (0.6R)$ to obtain the modified Equation 3.4.3. However, experimental results reported in this study revealed that this equation is also unable to predict the effective widths accurately for slender plates.

This detailed experimental investigation showed and confirmed that the current design formulae based on effective width principles are inadequate for the profiled sandwich panels with slender plates. However, it must be noted that the original effective width formulae (Equation 3.5.1) for the plate elements were developed by Winter (1947) based on many tests and extensive studies of postbuckling strength on cold-formed steel plates and sections. These steel plates buckled locally and developed considerable postbuckling strength before collapsing at their ultimate loads. This implies that this method can be applied for sandwich panels that have plate elements with low $b/t$ ratios as they exhibit considerable postbuckling strength. With the increasing $b/t$ ratio of the steel plates supported by foam core, there is either very little or no postbuckling strength. Therefore the extension of the conventional effective width method to sandwich panels with slender plates may not represent the true ultimate strength behaviour.

For the plates with very high $b/t$ ratios, the strength will be governed by wrinkling failure and can be evaluated using the well established wrinkling formula (CIB,
The main problem is the intermediate range of $b/t$ ratios between the Winter and wrinkling regions. In this case there is no wrinkling failure, instead local buckling occurs, but with little or no postbuckling strength. Many fully profiled sandwich panels fall in this region. Therefore the current effective width design formula in its current form can not predict the true strength of the slender plates in this intermediate region. However, it is considered that a further modification to the effective width design formula will enable accurate strength predictions for these sandwich panels. Further investigation of the local buckling behaviour of foam supported steel plate elements that fall into this intermediate region has been conducted using experimental and finite element analysis in Chapter 4.

However, it is useful to note that preliminary finite element analyses have confirmed the above observations. These analyses indicated that wrinkling failure is more dominant for the plates with $b/t$ ratio more than 1000. Most practical profiled sandwich panels have a $b/t$ ratio less than 600. Hence the wrinkling formula can not be applied to the plate elements in profiled sandwich panels as it will underestimate the strength. Kech (1991) had developed a buckling formula for sandwich panels subjected to wrinkling and local buckling effects. But his method is valid only for the case of profile depth to thickness ratio of less than 10 and can not be applied to fully profiled sandwich panels as the profile depth to thickness ratio is very high compared with the applicable limits of Kech’s formula. All of these indicate the need for a new or improved design rule for fully profiled sandwich panels with slender plates.

Since the buckling coefficient $K$ for the plate element without foam support is constant for a particular type of boundary conditions (eg. $K = 4$ for simply supported conditions), a constant value of $K$ was used in developing Equation 3.5.1. It was found that by simply changing the $K$ value, the formulae could be extended to other types of boundary conditions. However, for sandwich panels, the buckling coefficient $K$ changes with $b/t$ ratios and properties of foam and steel plates. Hence, by extending this formula to sandwich panels by simply considering a modified $K$ value may not be sufficient. To make the basis of formulation valid and accurate, it is important to consider variable $K$ while formulating the effective width formula for sandwich panels. This will make the design rule more effective and reliable.
3.6 An Interim Design Equation for Safe Solution

To improve the understanding of local buckling behaviour of profiled sandwich panels further and develop the new design rule, finite element analyses of sandwich panels were undertaken. Details of the finite element analyses and the results have been presented in the next chapter. However, based on experimental findings on foam supported steel plates, it can be concluded that currently used effective width approach is unconservative to sandwich panels with slender plates in its present form and new improved design formulae have to be developed to estimate accurate effective widths that can be used for design purposes.

As an interim design solution, $R$ in Equation 3.4.1 was reduced by an empirical reduction factor of 0.1 to determine the buckling coefficient $K$. The modified equation for $K$ can be written as:

$$K = [16 + 1.18R + 0.00055R^2]^{1/2}$$

with $R = \frac{12(1-\nu_f^2)^{\frac{1}{2}}E_cG_c}{\pi^3E_f} \left(\frac{b}{t}\right)^3$ (3.6.1)

The new $K$ values from Equation 3.6.1 were used in Equation 3.5.1 to determine the effective widths of the foam supported steel plates. The predicted effective widths were compared with experimental results and CIB (2000) recommendations (Equations 3.5.1 and 3.4.3) as shown in Figures 3.13 (a) and (b) for G550 steel plates and G250 steel plates, respectively. From these figures it can be observed that effective widths based on Equations 3.5.1 and 3.6.1 are in better agreement with the experimental results for a wider range of $b/t$ ratios from 0 to 500. This shows that effective width based on $0.1R$ provides a safe solution for all practical plate slenderness values. Hence Equation 3.6.1 combined with the current effective width Equation 3.5.1 is recommended as an interim design solution for the safe design of profiled sandwich panels subject to local buckling effects. It should be noted here that a reduced yield stress, $0.9f_y$, must be used in the design calculations of G550 grade steel plates with thickness less than 0.9 mm as recommended by Yang and Hancock (2002). This is to allow for possible strength reductions caused by lack of ductility in these plates.
Figure 3.13 Validation of Modified Design Rule with Experimental Results for Steel Plates Supported by Foam Core
3.7 Summary

Local buckling behaviour of profiled sandwich panels was investigated using an extensive series of laboratory experiments on foam supported steel plate elements. Experimental results were compared with predictions from current effective width design formulae. The results indicated that these design formulae are adequate for sandwich panels with plate elements that have low $b/t$ ratios, but not for panels with slender plate elements. Improved buckling and ultimate strength formulae have to be developed for these panels based on experimental and finite element analyses. Based on the results from experimental investigation, an interim design formula using an empirical reduction factor was recommended for the safe design of profiled sandwich panels subject to local buckling effects.
CHAPTER 4.0  NUMERICAL ANALYSIS OF FOAM SUPPORTED FLAT PLATES SUBJECT TO LOCAL BUCKLING

4.1 General

In this research project on sandwich panels subjected to local buckling effects, the behaviour of foam supported cold-formed steel plate elements used in profiled sandwich panels was investigated thoroughly using finite element analysis (FEA). Currently there are several finite element programs that can be used in the analysis and design of structures. They are SAP, NASTRAN, NISA, ANSYS and ABAQUS, which are capable of treating very complex problems. In this study, the finite element program ABAQUS was used to model and analyse sandwich panels subjected to local buckling effects because of its extensive capabilities and availability. HKS/Abaqus Standard Versions 5.8 and 6.3.1 were the finite element codes utilised in the numerical analysis. All the numerical computations were performed on Silicon Graphics Origin 3000 Series super computer that have 60 R14000 MTPS processors, with 26 GB memory using the IRIX operating system at QUT’s High Performance Computing Facility. MSC/PATRAN was used for pre-processing (model generation) and post-processing (visualisation of results) phases of modelling.

This chapter describes mainly the finite element analysis carried out on foam supported steel plate elements of profiled sandwich panels to investigate their local buckling, postbuckling and ultimate strength behaviour. The complete description of the models including the types of elements, the mesh density, the boundary conditions, and analysis methods used is presented first. Details of half-length models along with full-length models are outlined next. Detailed descriptions of the half-wave buckle length model including model geometry, validation with theoretical results are also discussed. The Chapter also includes details of an experimental study on slender plate elements conducted to identify the reasonable limit of $b/t$ ratio that should be included in developing the new design rule for local buckling. Finally, the review of the existing design rules and the formulation of new improved design formulae are presented based on FEA results from half-wave buckle length models.
4.2 Finite Element Model

4.2.1 Types of Model

In order to develop an accurate and reliable finite element model that simulates true behaviour of sandwich panels, various types of numerical models can be used and analysed in a finite element investigation. The full-scale model may be the easiest way to develop and use in the analysis as it uses the actual dimensions of the structure and does not require any parametric study to scale down the sizes of the model. However, the disadvantage associated with this model is the poor level of accuracy obtained due to the smaller number of elements that can be included in the analysis. Also, it is very uneconomical as it needs large computational time.

To eliminate such difficulties, a reduced model with appropriately determined member dimensions can be used for the analysis. One such reduced model is the half-length model. In this model, only half the length \((L/2)\) of the panel is used to create and analyse the model using appropriate boundary conditions. Also, by using half the width \((b/2)\), the half-length model can be reduced to the quarter size of the full panel. As the full panel is reduced to quarter size, a large number of elements with smaller sizes can be used in finite element meshing that will ultimately increase the level of accuracy of numerical results. Figure 4.1 shows the actual dimensions used in the half-length model.

![Figure 4.1 Concept of Half-Length Model](image-url)

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*Behaviour and Design of Sandwich Panels Subject to Local Buckling and Flexural Wrinkling Effects*  
4-2
Another reduced model generally used in the numerical analysis of sandwich panels is the half-wave buckle length model. In this model, half of the half-wave buckle length is used to create the geometry of the model. In sandwich panels, the half-wave buckle length is very small due to the stiffening effect of the foam core. Therefore the half-wave buckle length model is significantly smaller than half-length model. The half-wave buckle model is very similar to the theoretical model based on the elastic half-space method. Figure 4.2 shows the typical dimensions used for the half-wave buckle length model.

![Figure 4.2 Concept of Half-Wave Buckle Length Model](image)

In this investigation, a half-length model was used to compare with the experimental results, as this model closely represents the foam-supported steel plates tested in the laboratory. A full-length model was also used to validate the use of the half-length model. The half-wave buckle length model was used to review the current design rule and develop the new design rule for local buckling effects. A detailed description of these three types of finite element models is presented in the relevant sections of this chapter. All the finite element models used in this investigation were based on the application of compressive load to one end of the steel plate with two longitudinal edges of the plate being simply supported. This condition simulates the theoretical approach based on the energy method. The foam core was extended sufficiently deep so that the theoretical approach of elastic half space (i.e. the core) to extend infinitely in this direction was simulated. To achieve this, a constant depth of 100 mm, same as in the experiments, was used in all the models.
4.2.2 Elements

To simulate the actual structural behaviour of profiled sandwich panels, it is necessary to give attention to several considerations. Since the thin steel plate elements used in fully profiled sandwich panels are subjected to local buckling effects, the chosen element must be capable of modelling local buckling deformations and associated behaviour. For the plate element with a low $b/t$ ratio, the ultimate capacity is governed by postbuckling behaviour rather than local buckling behaviour. The postbuckling phenomenon is a complicated behaviour and occurs beyond the elastic region. So the element must be capable of modelling structural behaviour both in linear and non-linear regions involving large displacements, elastoplastic deformations and associated plasticity effects. In the ABAQUS element library (HKS, 1998), the shell elements generally satisfy these criteria and can be used to model the steel plate elements of profiled sandwich panels.

Although there are different types of shell elements available in the ABAQUS element library (HKS, 1998), the S4R5 three dimensional (3D) thin shell element (Quad4) with four nodes and five degrees of freedom per node was used to model the steel plate element as it is considered the most suitable for the proposed finite element analysis. It is a small-strain thin shell element and can model large rotations accurately. This element can be more economical as it uses only five degrees of freedom (three displacement components and two in-surface rotation components). The thin shell element is used in cases where transverse shear flexibility is negligible and the thickness of the shell is less than about 1/15 of a characteristic length on the surface of the shell, such as the distance between the supports or the wave length of a significant eigenmode (HKS, 1998). Steel faces used in the sandwich panels fall well within this category and hence, 3D thin shell element S4R5 with reduced integration were used successfully to model the steel plate elements.

The foam core was modelled using C3D8 three dimensional solid (continuum) elements (Hex8) with eight nodes and three degrees of freedom per node. These elements, which have no rotational degrees of freedom, are also called 8-node linear bricks. They are hexahedra and isoparametric in form. Compared with triangular and
tetrahedra elements, quadrilateral shell element used to model the steel faces and brick elements used to model the foam core have a better rate of convergence (HKS, 1998). Since there was no relative movement between the steel faces and foam core, they were modelled as a single unit.

### 4.2.3 Mechanical Properties

A wide variety of materials is encountered in structural analysis problems, and a range of constitutive models is available to describe the material behaviour for these different varieties of materials. The constitutive library in ABAQUS provides a comprehensive coverage of linear and non-linear, isotropic and anisotropic material models for most of the engineering materials. For the more commonly encountered materials (eg. metals), several ways of modelling the material are provided, each suitable to a particular type of analysis application. The user can select any constitutive model required for the simulation. Different regions in a model can be associated with different material definitions through the assignment of section properties that refer to the material name.

For all the analyses in this study, an isotropic material model was used for both steel plates and polystyrene foam core as previous studies (Jeevaharan, 1997 and McAndrew 1999) have indicated that the differences in mechanical properties in longitudinal and transverse direction are not significant. For buckling analysis, a linear elastic material model was used. For the non-linear analysis, the ABAQUS classical plasticity model with elastic perfectly plastic bi-linear stress-strain model was used. Perfect plasticity assumes no strain hardening (i.e., the yield stress does not change with increasing plastic strain).

To simulate the observed experimental behaviour, measured mechanical properties of polystyrene foam and steel plates as used in the experimental investigation were used in the analysis. These material properties are $E_c = 3.8$ MPa, $G_c = 1.76$ MPa and $\nu_c = 0.08$ for foam whereas the values for the G550 and G250 grades of steels were taken from Tables 3.4 and 3.5, respectively. Poisson’s ratio $\nu$ of steel was assumed to 0.3.
4.2.4 Loads and Boundary Conditions

Loads were represented as concentrated nodal forces. Axial compressive load was applied to one end of the steel face by distributing equally into the individual nodes.

The accuracy of the results obtained from the finite element modelling largely depends on the appropriate selection of the boundary conditions. MSC/PATRAN uses the following notations for the constraint conditions to be used in the numerical analysis.

a) 1 = X axis translation
b) 2 = Y axis translation
c) 3 = Z axis translation
d) 4 = X axis rotation
e) 5 = Y axis rotation
f) 6 = Z axis rotation

The choice of the boundary conditions depends on the type of model selected i.e. full model or symmetric model. The size of the finite element model can be reduced significantly by using symmetry in the structure being analysed. The loads and boundary conditions used for the three different models such as half-length model, full-length model and half-wave buckle length model are described in the respective sections of this chapter.

4.2.5 Analysis Methods

The methods of analysis used for the investigation of local buckling behaviour of the foam-supported steel plate elements were elastic buckling analysis and non-linear analysis. Elastic buckling analysis is a linear perturbation analysis used to obtain eigenvalue-buckling estimates. The critical buckling stress, buckling shape and half-wave buckle length required to create the half-wave buckle length model were obtained from elastic buckling analysis. Elastic buckling analysis was also used to obtain the eigen modes to represent the geometric imperfection distribution shape.
required for non-linear analysis. Ultimate strength of the foam supported steel plate elements was determined from a non-linear analysis. ABAQUS allows for both geometric and material nonlinearities to be included in the non-linear analysis. In this study, a plastic constitutive model with a Mises/Hill yield criteria and a perfect plasticity hardening rule was used in the non-linear analysis. Both RIKS and Newton-Raphson solution techniques with a default convergence tolerance were used in the non-linear analysis. The ultimate strength results obtained using both techniques were found to be numerically very close. In this study, the results obtained from RIKS method were used in all the evaluations and validations.

4.2.6 Geometric Imperfections and Residual Stresses

It is unlikely to find any cold-formed steel members in a shape of their original perfect geometry. The actual members always deviate from their original shape to some extent. Hence, it is important that appropriate geometric imperfections along with residual stresses are introduced in a finite element model while undertaking non-linear analysis to simulate the true shape and structural behaviour of the specimens. However, residual stresses were not considered in the analysis of foam supported steel plate elements considered in this study as they did not involve cold-forming or welding of a section or similar fabrication/manufacturing process capable of producing higher residual stresses. Past research has also shown that the effect of residual stresses on the ultimate strength of thin cold-formed steel structures is not significant (Mahaarachchi, 2003; Telue, 2001).

The presence of geometric imperfections seriously affects the strength behaviour of compressed plate elements and ignoring them in the numerical analysis results in unrealistic strength predictions. It is generally considered that the most detrimental type of imperfection is one which has the same shape as the critical local buckling mode. The strength of a cold-formed steel member is always sensitive to the imperfection in the shape of its eigenmodes. In most cases, the mode shape based on the lowest eigenmode is sufficient to adequately characterize the most influential geometric imperfections, and this is considered an acceptable conservative approach (Schafer and Pekoz, 1998). Therefore, in the non-linear analyses of this study, the
mode shape of first buckling mode obtained from the elastic buckling analysis was used to introduce the critical geometric imperfection distribution shape.

Although, the shape of geometric imperfection can be based on the eigenmodes from the elastic buckling analysis, determining the appropriate magnitude of geometric imperfection required to scale the imperfection distribution shape is a very difficult task. In the past, several researchers have investigated geometric imperfections of cold-formed steel members, but no attempt has been made to date to find any general characterization of geometric imperfections (Schafer and Pekoz, 1998). However, in practice, maximum imperfection magnitudes are specified in terms of either the thickness or the width of the plates.

![This figure is not available online. Please consult the hardcopy thesis available from the QUT Library](image)

**Figure 4.3 Geometric Imperfections of Cold-Formed Steel Members**

(Schafer and Pekoz, 1998)

Schafer and Pekoz (1998) recommended that the maximum value of geometric imperfection \(d_1, d_2\) can be expressed in terms of plate width \(b\) in the case of local buckling of the web \(d_1 = 0.006b\) and in terms of thickness in the case of local buckling of the flange or distortional buckling \(d_2 = t\). Definitions of local imperfections \(d_1\) and \(d_2\) are shown in Figure 4.3. These recommendations are for cold-formed steel members without any foam core support. However, for sandwich panels, in which cold-formed steel plates are supported by a foam core, no data is available on the maximum value and the distribution of appropriate imperfections to be used in the numerical analyses. Experimental observations showed that the actual magnitudes of geometric imperfections in sandwich panels are quite small. This was expected as the imperfection magnitude of thin steel faces supported by a foam core
will not be as high as the values of flat plates without a foam core. However, it was
considered important to understand how the imperfections are produced in the
manufacturing process (Schafer, 2002). Through visits to sandwich panel
manufacturing plants and extensive consultations with sandwich panel
manufacturers, it was found that the width of the steel face has limited contribution
to the geometric imperfection magnitude as the steel face is fully supported by foam
core. However, it was observed that some imperfections might arise in the steel plate
due to the uneven surfaces of the polystyrene foam core, and therefore their
magnitudes would depend on the thickness of the steel plate used rather than on the
plate width. Although this imperfection was very small (considerably less than plate
thickness), it might still cause some reduction to the ultimate strength. Hence, in this
study, the maximum imperfection value used in the FEA was expressed in terms of
the thickness \( t \) of foam supported plate elements.

It is important to note that larger imperfections always result in lower estimates of
the failure load or ultimate strength while smaller imperfections overestimate the
failure load. A realistic value of imperfection will yield true post buckling and
ultimate strength behaviour of foam supported cold-formed steel plates. Schafer
(2002) suggests that a series of analyses will provide ultimate strength as a function
of imperfection magnitude and this will give an insight as to how sensitive sandwich
panels actually are to the geometric imperfection magnitude. Multiple analyses on at
least a few sandwich panels should be conducted to characterize the imperfection
sensitivity. Therefore, a sensitivity analysis was conducted using FEA on a few
sandwich panels with different plate thicknesses to determine the effects of
maximum geometric imperfections. Figures 4.4 (a) and (b) present the relationship
between the ultimate strength and the imperfection magnitudes expressed in terms of
plate thickness \( t \). As seen from the figures, for a range of imperfections from 0.1\( t \) to
0.4\( t \), reduction in ultimate strength was found to be minimal (< 5%). As already
described, only a very small or no imperfection was observed in the sandwich panels
produced by Australian manufacturers, it was therefore decided to use 10% of the
plate thickness (0.1\( t \)) as the maximum value of geometric imperfection in all the
finite element analyses. This geometric imperfection value provided good predictions
of ultimate strength when compared with the experimental results as presented later
in this chapter.
Figure 4.4 Variation of Ultimate Load with Maximum Geometric Imperfection
4.3 Half-Length Model

To simulate the foam-supported steel plate elements tested in the laboratory, a half-length model was used with appropriate boundary conditions including that of symmetry. To confirm the results, a full-length model was first analysed for some of the specimens. The length used for the full-length model was about 3 times the width as used in the experiments. Since the results from full-length and half-length models agreed well, further analyses were conducted using half-length models with only half width to save on computational time. A constant foam thickness of 100 mm was used to simulate the experimental conditions.

4.3.1 Model Geometry, Mesh Sizes and Boundary Conditions

All the foam supported steel plates tested in the experiments were modelled and analysed using half-length experimental model (see Figure 4.5). The width of each model was \( b/2 \) (half the plate width), length \( 3b/2 \) (half the length of the specimen), and thickness sum of the foam and steel thicknesses \( (t_f + t_c) \).

![Figure 4.5 Half-Length Model of Foam Supported Steel Plate](image)
The model geometry created to simulate experimental foam supported steel plate was discretised into a number of finite elements. As the mesh density increases, the accuracy of a finite element model generally increases and converges to a numerically correct solution. Therefore it is necessary to have a fine mesh to obtain the appropriate solution. The accuracy of the model can then be compared with the experimental results and other theoretical solutions. In order to determine the appropriate mesh density, a convergence study was conducted with gradually increasing or decreasing mesh sizes for half-length model. Based on the convergence study, a mesh with 10 mm square surface elements (for steel plate) and solid elements with 10×10×5 mm throughout foam depth was used for half-length model. This mesh size provided satisfactory results in terms of accuracy.

Figure 4.5 shows the model geometry, mesh size and the loading pattern along with appropriate boundary conditions for the half-length model. Appropriate boundary conditions were applied only to the steel plate at the loading end and one of the longitudinal edges to simulate the experiments whereas symmetric boundary conditions were applied to the entire surface (i.e., to the steel plate and foam core) along both the longitudinal direction and across the width to model half width and half length, respectively. As shown in Figure 4.5, the boundary condition 356 was applied to the steel plate only at the loading end. This boundary condition allows the translation in the X and Y directions and rotation about the X axis. However, it does not allow translation in the Z direction and rotation about either Y or Z axes. Similarly, the boundary condition 346 was applied to the longitudinal edge of the steel plate in order to model the simply supported edge of the sandwich panels. The symmetric boundary condition 156 was applied to the entire surface (plate and foam) along the length of the panel at the centre of the panel width (b/2). Similarly, symmetric boundary condition 246 was applied along the width at the centre of the panel length (L/2). An axial compressive load was applied to one end of the steel plate by distributing equally into the individual nodes. However, only half the nodal load was applied to the end nodes.

To validate the results obtained from the half-length model, some of the specimens were modelled and analysed using the full-length model. Figure 4.6 shows the model geometry, mesh size and the loading pattern along with appropriate boundary
conditions for the full-length model. The width of each full model was $b$ (plate width), length $3b$ (length of the specimen), and thickness being the sum of the foam and steel thicknesses ($t_f + t_c$). The same mesh size used in the half-length model was used for this full length model.

![Figure 4.6 Full-Length Model of Foam Supported Steel Plate](image)

In the full-length model, appropriate boundary conditions were applied to the four sides of the panel by restraining only the edges of the steel plate, not the surface of the foam core. This was done in order to simulate the experimental sandwich panels as shown in Figure 4.6. The boundary condition 356 was applied at the loading edge and 2356 was applied at the other fixed edge. Along the two longitudinal edges, the boundary conditions 346 were applied to simulate the simply supported end conditions. The critical buckling and ultimate stress results obtained from the full-length and half-length models were compared to confirm that one model can represent the other. Table 4.1 presents the comparison of these results for some of the foam supported steel plate elements considered in this study. As seen from Table 4.1, the results obtained from the full-length and the half-length models were very close with a maximum difference of 3%. This comparison confirmed that the full-length
model can be well represented by the symmetric half-length model. Therefore further analyses in this study were conducted using the half-length model with only half width to save on computational time.

**Table 4.1 Comparison of Half-Length and Full-Length Models**

<table>
<thead>
<tr>
<th>Plate width b (mm)</th>
<th>Length L (mm)</th>
<th>Thickness t (mm)</th>
<th>Buckling Stress (MPa)</th>
<th>Ultimate Stress (MPa)</th>
</tr>
</thead>
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<td></td>
<td></td>
<td></td>
<td>FLM</td>
<td>HLM</td>
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<td>80</td>
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<td>0.80</td>
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<tr>
<td>100</td>
<td>300</td>
<td>0.80</td>
<td>121.13</td>
<td>122.50</td>
</tr>
<tr>
<td>120</td>
<td>360</td>
<td>0.95</td>
<td>117.81</td>
<td>118.42</td>
</tr>
<tr>
<td>150</td>
<td>450</td>
<td>0.95</td>
<td>100.63</td>
<td>101.05</td>
</tr>
</tbody>
</table>

Note: FLM = Full-length model  
HLM = Half-length model

### 4.3.2 Validation of the Half-Length Model

The half-length model was first analysed using an elastic buckling analysis followed by a non-linear analysis. Buckling stresses of the foam-supported steel plate elements were obtained from the elastic buckling analysis and ultimate stresses were obtained from the non-linear analysis, respectively. Buckled shapes corresponding to the various buckling modes were visualised. Earlier studies conducted by Davies et al (1991) and Mahendran and Jeevaharan (1999) have indicated that the primary buckling mode is always critical. Therefore in this study, the eigen value of the first buckling mode was considered to determine the elastic buckling strength of the plate elements, however, other modes were also viewed. The first buckling mode which was very close to the experimental buckling shape was used to input geometric imperfection in the non-linear analysis. For all non-linear analyses, 0.1 times the plate thickness \((0.1t)\) was used as the magnitude of geometric imperfection as described in Section 4.2.6. Figure 4.7 shows the typical buckled shape of the half-length model. It should be noted that colours in this figure indicate the magnitude of out-of-plane deflection with red showing the largest deflection.
The FEA results obtained from the elastic buckling and non-linear analysis were compared with the corresponding experimental results given in Chapter 3. Tables 4.2 and 4.3 present the comparison of FEA and experimental buckling and ultimate stresses for G550 and G250 steel plates, respectively. As seen in these tables, the results from FEA and experiments agreed reasonably well for both G550 and G250 steel plates. The mean values of the ratio of FEA and experimental buckling and ultimate stresses were found to be 1.00 and 0.94, respectively, for G550 steel plates, and 1.05 and 0.93, respectively, for G250 steel plates. The corresponding coefficients of variation (COV) were 0.06 and 0.11, respectively, for G550 steel plates, and 0.08 and 0.12, respectively, for G250 steel plates. Figures 4.8 (a) and (b) present the comparison of buckling stress results from FEA and experiments for G550 and G250 steel plates, respectively. Similarly, Figures 4.9 (a) and (b) present the comparison of ultimate stress results from FEA with experiments for G550 and G250 steel plates, respectively. A comparison of typical load versus axial displacement curves from FEA and experiments are shown in Figures 4.10 (a) to (d), and typical load versus out-of-plane displacement curves are shown in Figures 4.10 (e) to (h). All these comparisons confirm that the half-length model can be satisfactorily used to analyse the local buckling behaviour of foam-supported steel plates used in the experiments.
Table 4.2 Comparison of Results from Half-Length Model and Experiments for G550 Steel Plates

<table>
<thead>
<tr>
<th>Test No.</th>
<th>b/t ratio</th>
<th>Buckling Stress</th>
<th>Ultimate Stress</th>
</tr>
</thead>
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<tr>
<td></td>
<td></td>
<td>FEA</td>
<td>Expt.</td>
</tr>
<tr>
<td></td>
<td></td>
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<td>275.00</td>
<td>293.00</td>
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<td>25</td>
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Mean 1.00 0.94
Coefficient of Variation (COV) 0.06 0.11
Table 4.3 Comparison of Results from Half-Length Model and Experiments for G250 Steel Plates

<table>
<thead>
<tr>
<th>Test No.</th>
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<th>Buckling Stress</th>
<th>Ultimate Stress</th>
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</tr>
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<td>461.5</td>
<td>74.64</td>
<td>64.10</td>
</tr>
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<td>215.1</td>
<td>83.76</td>
<td>84.95</td>
</tr>
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<td>274.0</td>
<td>78.29</td>
<td>72.74</td>
</tr>
<tr>
<td>24</td>
<td>370.4</td>
<td>74.26</td>
<td>65.74</td>
</tr>
<tr>
<td>25</td>
<td>512.8</td>
<td>72.69</td>
<td>64.62</td>
</tr>
</tbody>
</table>

Mean 1.05  0.93  
Coefficient of Variation (COV) 0.08  0.12
Figure 4.8 Comparison of FEA and Experimental Buckling Stress Results

(a) G550 Steel Plates

(b) G250 Steel Plates
Figure 4.9 Comparison of FEA and Experimental Ultimate Stress Results
Figure 4.10 Comparison of Typical Load-Deflection Curves

(a) Compressive Load vs Axial Displacement

(G550, $b = 180 \text{ mm}$, $t = 0.60 \text{ mm}$)

(b) Compressive Load vs Axial Displacement

(G250, $b = 200 \text{ mm}$, $t = 0.93 \text{ mm}$)
Figure 4.10 Comparison of Typical Load-Deflection Curves

(c) Compressive Load vs Axial Displacement
(G550, $b = 200$ mm, $t = 0.80$ mm)

(d) Compressive Load vs Axial Displacement
(G550, $b = 200$ mm, $t = 0.60$ mm)
Figure 4.10 Comparison of Typical Load-Deflection Curves

(e) Compressive Load vs Out-of-Plane Displacement
(G550, \( b = 150 \) mm, \( t = 0.95 \) mm)

(f) Compressive Load vs Out-of-Plane Displacement
(G550, \( b = 100 \) mm, \( t = 0.95 \) mm)
Figure 4.10 Comparison of Typical Load-Deflection Curves

(g) Compressive Load vs Out-of-Plane Displacement
(G250, b = 200 mm, t = 0.93 mm)

(h) Compressive Load vs Out-of-Plane Displacement
(G550, b = 80 mm, t = 0.95 mm)
Thinner G550 grade steels exhibit limited ductility with little strain hardening and reduced fracture strain characteristics due to their manufacturing process. Dhalla and Winter (1974a, b) developed suitable ductility criteria to ensure satisfactory performance of thin steel members under static load. Based on extensive experimental investigations, Rogers and Hancock (1997) showed that the G550 grade steels do not meet the Dhalla and Winter’s (1974b) ductility criteria. This characteristic could affect the load carrying capacity of members made of thin G550 steels. Hence AS/NZS 4600 (SA, 1996) recommends the use of a reduced yield stress of $0.75f_y$ for G550 steels with a thickness less than 0.9 mm. Yang and Hancock’s (2002) tests on G550 steel compression members showed the need for using a reduced yield stress of $0.9f_y$ to predict the lower experimental strengths.

Since G550 steel plates with different thicknesses (0.42, 0.60, 0.80 and 0.95 mm) were used in this study, it was considered useful to investigate the ultimate strength behaviour of foam supported plate elements made of G550 steels. In Table 4.4, the mean values of the ratio of buckling and ultimate stresses from finite element analyses and experiments (FEA/Expt.) are compared for different thicknesses of G250 and G550 grade steels.

**Table 4.4 Comparison of Results for Different Thicknesses of Steels**

<table>
<thead>
<tr>
<th>Steel Grade</th>
<th>Thickness $t$ (mm)</th>
<th>Mean of Buckling Stress Ratio FEA/Expt.</th>
<th>Mean of Ultimate Stress Ratio FEA/Expt.</th>
</tr>
</thead>
<tbody>
<tr>
<td>G550</td>
<td>0.95</td>
<td>1.01</td>
<td>0.90</td>
</tr>
<tr>
<td></td>
<td>0.80</td>
<td>0.96</td>
<td>0.86</td>
</tr>
<tr>
<td></td>
<td>0.60</td>
<td>1.02</td>
<td>0.97</td>
</tr>
<tr>
<td></td>
<td>0.42</td>
<td>1.01</td>
<td>1.05</td>
</tr>
<tr>
<td>G250</td>
<td>0.93</td>
<td>1.04</td>
<td>0.90</td>
</tr>
<tr>
<td></td>
<td>0.73</td>
<td>0.99</td>
<td>0.90</td>
</tr>
<tr>
<td></td>
<td>0.54</td>
<td>1.08</td>
<td>0.93</td>
</tr>
<tr>
<td></td>
<td>0.39</td>
<td>1.09</td>
<td>1.01</td>
</tr>
</tbody>
</table>
As seen from the table, the ratios of ultimate stresses from FEA and experiments were about the same (0.9) for thicker G550 steels (0.95 and 0.8 mm) and most of the G250 steels. However, the ultimate stress ratio increases as the G550 steel thickness decreases with a higher ratio of 1.05 for 0.42 mm G550 steel. This means that experimental strengths are less than the expected values for thinner G550 steels. Since the FEA does not include the possible strength reductions that could occur for thinner G550 steels with reduced ductility, the ultimate stress ratio is likely to increase with reducing thickness of G550 steel and will be greater than 1 for very thin steels. This can be seen in the results in Table 4.4. All of these observations therefore appear to confirm the AS/NZS 4600 (SA, 1996) requirement and Yang and Hancock’s (2002) recommendation of using a reduced yield stress ($0.75f_y, 0.9f_y$) for G550 steels with a thickness less than 0.9 mm in predicting the member strengths. As expected there is no such observation with buckling stress ratios (see Table 4.4).

As mentioned earlier following the comparisons in Tables 4.2 and 4.3 and Figures 4.8 to 4.10, the finite element model developed in this research is capable of simulating the local buckling behaviour of foam supported steel plate elements. However, it appears to overestimate the strength of thinner G550 steel plate elements compared with thicker G550 steels and most of the G250 steels. The reasons for this are given above, and the problem can be rectified by using a reduced yield stress. Therefore in this research it was assumed that further research is not required to investigate the reduction in ultimate strengths of members made of thinner G550 steels if a reduced yield stress $0.9f_y$ is used as recommended by Yang and Hancock (2002).

However, this research was continued using finite element models of both low and high strength steels. The finite element model does not simulate the behaviour of thinner G550 steels with reduced ductility, but is considered acceptable as the aim of this research is to study the effect of plate slenderness on the strength capacity of sandwich panels and not the effect of reduced ductility of G550 steels on the ultimate strength behaviour.
4.4  **Half-Wave Buckle Length Model**

The foam-supported steel plate elements used in the experiments do not represent exactly those in practical sandwich panels. For the simplicity of the experiments, the foam width of the tested specimens was made the same as the steel face width. In the test rig, only the steel plates were restrained along the four sides leaving the foam unrestrained, but the foam in real sandwich panels is continuous in the width direction. Hence the half-length model developed to simulate the experimental panels cannot be used for reviewing and developing the design rules for local buckling of sandwich panels. However, the validation of the half-length model by comparing its results with the experimental results provided the confidence in using FEA model for reviewing and developing design rules. The half-wave buckle length model matches with the theoretical model used to develop the buckling stress formula based on elastic half space method as given in Equations 2.4.13 to 2.4.16 (Mahendran and Jeevaharan, 1999). Hence a half-wave buckle length model was used with appropriate boundary conditions including that of symmetry to investigate the local buckling behaviour of practical profiled sandwich panels. This section presents the description of half-wave buckle models and their results.

4.4.1  **Model Geometry, Mesh Sizes and Boundary Conditions**

The width of each half-wave buckle length model was \( b/2 \) (half the plate width), length \( a/2 \) (half of the half-wave buckle length, \( a \), and thickness equal to the sum of the steel thickness \( t \) and a constant foam thickness \( t_c \) of 100 mm (see Figure 4.11). As in the half-length model, the appropriate mesh density for half-wave buckle length model was determined by conducting a convergence study on foam supported steel plate of thickness 0.6 mm and G550 grade with the above mentioned dimensions. Based on the convergence study presented in Table 4.5, a mesh with 10 mm square surface elements (for steel plate) and solid elements with 10×10×5 mm throughout foam depth provided satisfactory results in terms of accuracy as in the case of half-length model (< 2% difference in results). However, to obtain more accurate results, a mesh with 5 mm square surface elements (for steel plate) and 5×5×5 mm solid elements throughout foam depth was used for half-wave buckle length models.
Figure 4.11 Half-Wave Buckle Length Model of Foam Supported Steel Plate

Table 4.5 Convergence Study to Estimate Appropriate Mesh Size for Half-Wave Buckle Length Model

<table>
<thead>
<tr>
<th>Steel Plate Width $b$ (mm)</th>
<th>Length $a$ (mm)</th>
<th>Thickness $t$ (mm)</th>
<th>Mesh Density (mm)</th>
<th>Buckling Load (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Steel Face</td>
<td>Foam Core</td>
<td>Steel Face</td>
</tr>
<tr>
<td>100</td>
<td>46</td>
<td>0.6</td>
<td>100</td>
<td>10×10</td>
</tr>
<tr>
<td>100</td>
<td>46</td>
<td>0.6</td>
<td>100</td>
<td>10×10</td>
</tr>
<tr>
<td>100</td>
<td>46</td>
<td>0.6</td>
<td>100</td>
<td>5×5</td>
</tr>
<tr>
<td>100</td>
<td>46</td>
<td>0.6</td>
<td>100</td>
<td>5×5</td>
</tr>
<tr>
<td>100</td>
<td>46</td>
<td>0.6</td>
<td>100</td>
<td>4×4</td>
</tr>
</tbody>
</table>

Appropriate boundary conditions were applied to the entire surface (i.e., to the steel plate and foam core) along all four sides. As shown in Figure 4.11, the boundary condition 356 was applied along the loading end, and symmetric boundary condition 246 was applied along the width at the centre of the half wave buckle length ($a/2$). Similarly, the boundary condition 346 was applied to the longitudinal edge of the panel and the symmetric boundary condition 156 was applied along the length of the
panel at the centre of the panel width \((b/2)\). Similarly, the axial compressive load was applied to one end of the steel plate by distributing it equally to the individual nodes. Figure 4.11 shows the model geometry, mesh size and the loading pattern together with appropriate boundary conditions for the half-wave buckle length model.

### 4.4.2 A Single Half-Wave Buckle Length

In order to create the half wave buckle length model, a single half-wave buckling length \(a\) has to be determined first. The half-wave buckle length \(a\) depends on the width of the sandwich panels. The length of the half-wave buckle length model, \(a/2\), was found by varying \(a/2\) using a series of elastic buckling analyses until the minimum eigenvalue and thus the buckling stress was obtained. The theoretical approach of determining the half-wave buckling length \(a\) is based on the energy method (see Section 2.4). A steel plate supported on an infinitely deep foam core represents the plate on an elastic foundation as considered in the energy method. Hence, the half-wave buckling length \(a\) determined from the finite element analysis can be compared with the \(a\) obtained from the theoretical method.

The half-wave buckling length \(a\) was determined for all the specimens with varying steel grades and \(b/t\) ratios considered in this study by the above mentioned minimisation process. For this, a symmetric model was used and the model length \(a/2\) was varied until the minimum buckling stress was found. Although it was very tedious and time consuming process, the exact value of \(a/2\) was determined for every panel in order to obtain accurate finite element analysis results. The results from a typical study conducted on foam-supported steel plate elements of grade G550 with a width 100 mm and thickness 0.6 mm are given in Table 4.6. Figure 4.12 shows the variation of buckling load with different half-wave buckle lengths obtained in the minimisation process. As seen in Table 4.6 and Figure 4.12, when the assumed model length \(a/2\) is very close to the true value, buckling stress corresponding to this length is always the minimum. This length corresponding to the minimum buckling stress was taken as half of the half-wave buckle length \((a/2)\) of that particular foam supported steel plate. This process was repeated for all the specimens considered in this study.
### Table 4.6 Determination of Half-Wave Buckle Length

<table>
<thead>
<tr>
<th>a/2</th>
<th>Surface element (mm)</th>
<th>Solid Element (mm)</th>
<th>Eigen Value</th>
<th>( \sigma_{cr} ) (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>5\times5</td>
<td>5\times5\times5</td>
<td>432.62</td>
<td>144.21</td>
</tr>
<tr>
<td>20</td>
<td>5\times5</td>
<td>5\times5\times5</td>
<td>358.76</td>
<td>119.59</td>
</tr>
<tr>
<td>22</td>
<td>5\times5</td>
<td>5\times5\times5</td>
<td>352.40</td>
<td>117.47</td>
</tr>
<tr>
<td>23*</td>
<td>5\times5</td>
<td>5\times5\times5</td>
<td>351.54*</td>
<td>117.18*</td>
</tr>
<tr>
<td>24</td>
<td>5\times5</td>
<td>5\times5\times5</td>
<td>352.63</td>
<td>117.54</td>
</tr>
<tr>
<td>30</td>
<td>5\times5</td>
<td>5\times5\times5</td>
<td>381.19</td>
<td>127.06</td>
</tr>
<tr>
<td>35</td>
<td>5\times5</td>
<td>5\times5\times5</td>
<td>424.55</td>
<td>141.52</td>
</tr>
</tbody>
</table>

Note: * - Critical minimum buckling stress and corresponding \( a/2 \)

![Figure 4.12 Buckling Stress versus a/2](image)
The FEA buckling results are obtained only in terms of eigen values. The following formula was then used to convert the eigen value to a corresponding critical buckling stress.

\[
\text{Critical Buckling Stress, } \sigma_{cr} = \frac{\text{Eigenvalue} \times \text{Number of Elements} \times \text{Load per Node}}{\text{Model Width} \times \text{Steel Thickness}}
\]

For example for half of the half-wave buckle length \(a/2 = 23\) mm from Table 4.6, the values of various parameters are eigenvalue = 351.54, number of elements = 10, load per node = 1 N, model width = 50 mm and steel thickness = 0.6 mm. After substituting all these values, a critical buckling stress \(\sigma_{cr} = 117.18\) MPa can be obtained from the above equation.

### 4.4.3 Validation of the Half-Wave Buckle Length Model

As in the case of the half-length model, the half-wave buckle length model also was analysed first using elastic buckling analysis, followed by non-linear analysis. The first buckling mode from the elastic buckling analysis was used to input the geometric imperfection for the non-linear analysis with a magnitude of 0.1\(t\). Figure 4.13 shows the buckled shape of the half-wave buckle length model.
Table 4.7 Comparison of Results from Half-Wave Buckle Length Model and Theory for G550 Steel Plates

<table>
<thead>
<tr>
<th>Test No.</th>
<th>b/t Ratio</th>
<th>a/b Ratio</th>
<th>Buckling Stress (MPa)</th>
<th>Ultimate Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Theory</td>
<td>FEA</td>
<td>Theory</td>
</tr>
<tr>
<td>1</td>
<td>52.6</td>
<td>0.884</td>
<td>0.88</td>
<td>350.3</td>
</tr>
<tr>
<td>2</td>
<td>62.5</td>
<td>0.834</td>
<td>0.84</td>
<td>275.8</td>
</tr>
<tr>
<td>3</td>
<td>83.3</td>
<td>0.727</td>
<td>0.72</td>
<td>196.7</td>
</tr>
<tr>
<td>4</td>
<td>119.0</td>
<td>0.576</td>
<td>0.60</td>
<td>146.7</td>
</tr>
<tr>
<td>5</td>
<td>84.2</td>
<td>0.718</td>
<td>0.73</td>
<td>189.5</td>
</tr>
<tr>
<td>6</td>
<td>100.0</td>
<td>0.646</td>
<td>0.65</td>
<td>163.3</td>
</tr>
<tr>
<td>7</td>
<td>133.3</td>
<td>0.526</td>
<td>0.55</td>
<td>135.6</td>
</tr>
<tr>
<td>8</td>
<td>190.5</td>
<td>0.393</td>
<td>0.40</td>
<td>117.6</td>
</tr>
<tr>
<td>9</td>
<td>105.3</td>
<td>0.621</td>
<td>0.64</td>
<td>155.1</td>
</tr>
<tr>
<td>10</td>
<td>125.0</td>
<td>0.550</td>
<td>0.56</td>
<td>139.1</td>
</tr>
<tr>
<td>11</td>
<td>166.7</td>
<td>0.439</td>
<td>0.46</td>
<td>122.2</td>
</tr>
<tr>
<td>12</td>
<td>238.1</td>
<td>0.322</td>
<td>0.34</td>
<td>111.0</td>
</tr>
<tr>
<td>13</td>
<td>126.3</td>
<td>0.544</td>
<td>0.57</td>
<td>137.0</td>
</tr>
<tr>
<td>14</td>
<td>150.0</td>
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<td>0.48</td>
<td>126.5</td>
</tr>
<tr>
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<td>200.0</td>
<td>0.375</td>
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<tr>
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<td>157.9</td>
<td>0.454</td>
<td>0.47</td>
<td>122.6</td>
</tr>
<tr>
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<td>0.41</td>
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</tr>
<tr>
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<td>0.32</td>
<td>109.2</td>
</tr>
<tr>
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<td>300.0</td>
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<td>0.354</td>
<td>0.37</td>
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<td>0.234</td>
<td>0.24</td>
<td>104.8</td>
</tr>
<tr>
<td>25</td>
<td>476.2</td>
<td>0.167</td>
<td>0.17</td>
<td>102.5</td>
</tr>
</tbody>
</table>
### Table 4.8 Comparison of Results from Half-Wave Buckle Length Model and Theory for G250 Steel Plates

<table>
<thead>
<tr>
<th>Test No.</th>
<th>b/t Ratio</th>
<th>(a/b) Ratio</th>
<th>Buckling Stress (MPa)</th>
<th>Ultimate Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Theory</td>
<td>FEA</td>
<td>Theory</td>
</tr>
<tr>
<td>1</td>
<td>53.8</td>
<td>0.873</td>
<td>0.88</td>
<td>350.8</td>
</tr>
<tr>
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<td>0.80</td>
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</tr>
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<td>187.8</td>
</tr>
<tr>
<td>4</td>
<td>128.2</td>
<td>0.534</td>
<td>0.56</td>
<td>137.4</td>
</tr>
<tr>
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<td>86.0</td>
<td>0.702</td>
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<td>193.5</td>
</tr>
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</tr>
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<td>0.474</td>
<td>0.50</td>
<td>137.7</td>
</tr>
<tr>
<td>8</td>
<td>205.1</td>
<td>0.359</td>
<td>0.38</td>
<td>114.1</td>
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<td>0.606</td>
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<td>159.8</td>
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<td>11</td>
<td>185.2</td>
<td>0.392</td>
<td>0.40</td>
<td>126.7</td>
</tr>
<tr>
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<td>256.4</td>
<td>0.293</td>
<td>0.30</td>
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</tr>
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</tr>
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</tr>
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<td>0.35</td>
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</tr>
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<td>128.0</td>
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<td>0.200</td>
<td>0.20</td>
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<td>113.3</td>
</tr>
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<td>461.5</td>
<td>0.168</td>
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</tr>
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<td>117.2</td>
</tr>
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<td>0.275</td>
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<td>114.3</td>
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<td>112.1</td>
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<tr>
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<td>0.151</td>
<td>0.16</td>
<td>101.7</td>
</tr>
</tbody>
</table>
The half-wave buckle length $a$ and the critical buckling stresses were obtained from the elastic buckling analysis whereas the ultimate stress was obtained from the non-linear analysis. The half-wave buckling length $a$ and the critical buckling stress were compared with the theoretical results obtained from Equations 2.4.13, 2.4.14, 2.4.16 and 2.4.18 given in Chapter 2. Tables 4.7 and 4.8 present the comparison of these buckling stress results from FEA and theory along with the ultimate stresses obtained from the FEA for G550 and G250 steel plates, respectively. As seen from these results, both the half-wave buckle length $a$ and the critical buckling stresses from the FEA agreed reasonably well with the theoretical results. The mean and COV of the ratio of buckling stresses from FEA and theory were found to be 0.97 and 0.01, respectively, for G550 steel plates and 0.90 and 0.03, respectively, for G250 steel plates. Hence these agreements confirmed that the half-wave buckle length model can be successfully used to model the local buckling behaviour of sandwich panels, review the existing design rules, understand the inadequacy of the current effective width approach for slender plates, and develop new improved design formulae.

### 4.5 Comparison of Effective Width Results

Effective width $b_{eff}$ of foam supported steel plate elements were determined from the ultimate stresses given in Tables 4.7 and 4.8 obtained from the half-wave buckle length model using the following formula:

$$\frac{b_{eff}}{b} = \frac{Ultimate\ Stress}{Yield\ Stress}.$$  

These effective widths obtained from the FEA results and those evaluated from Equation 3.5.1 using $K$ values predicted by theory and different buckling formulae are plotted against the $b/t$ ratios in Figures 4.14 (a) and (b) for G550 and G250 steel plates, respectively. The $K$ values as given in Tables 3.8 and 3.9 were determined for the steel plate elements supported by foam core by using Equations 3.4.1, 3.4.2, and 3.4.3 proposed by Davies and Hakmi (1990), Mahendran and Jeevaharan (1999), and CIB (2000), respectively. Theoretical values of $K$ were evaluated by using Equations 2.4.14 and 2.4.18.
Figure 4.14 Effective Width of the Steel Plates Supported by Foam Core
It can be observed from Figures 4.14 (a) and (b) that the effective widths ($b_{eff}$) evaluated from Equation 3.5.1 using $K$ values predicted by theory and different buckling formulae agreed reasonably well with the FEA results for low $b/t$ ratios (< 100). However, for higher $b/t$ ratios, all the formulae predicted very high effective width values compared with the FEA results. Similar conclusions were drawn in the experimental investigation as presented in Chapter 3. The FEA results also indicated that none of the current design formulae could estimate reasonable values of effective width $b_{eff}$ for slender plates with high $b/t$ ratios (> 100).

Figures 4.14 (a) and (b) show that the effective width results of G550 steel plates are similar to G250 steel plates. This occurs because the FEA did not consider the effects of reduced ductility in thinner G550 steels. As stated earlier, this research does not consider the reduced ductility issue. Instead it is aimed at reviewing the current effective width rules for slender plate elements and hence the use of finite element models developed here are adequate.

The confirmation that the current design formulae are not applicable to the slender plates by FEA and experiments implies the inadequacy of conventional effective width formulae for the plate elements with high $b/t$ ratios. As explained earlier, the extension of plain plate effective width approach to the sandwich panels by means of a modified buckling coefficient $K$ can not address the inadequacy in formulation that may arise while applying to sandwich panels. New improved design formulae have to be developed based on the finite element analysis results to estimate accurate values of effective widths that can be used for design purposes.

Before developing any design rule for the profiled sandwich panels with any practical $b/t$ ratio, it is important that the buckling behaviour of steel plate elements with comparatively high $b/t$ ratios (slender plate) must be fully understood. If the $b/t$ ratio of the plate element in sandwich panels is very high, its strength will be governed by wrinkling failure and can be determined by the well established wrinkling stress formula (CIB 2000) as described in Chapter 2. The strength of the slender plates, the $b/t$ ratio of which lies in this wrinkling region, is very low compared with the local buckling strength. The strength of compact plates whose $b/t$ ratio lies in this Winter region is dominated by postbuckling strength. However, if
the $b/t$ ratio of the plate element lies in the intermediate region (i.e. between the Winter and wrinkling regions), the wrinkling formula can not be applied as it underestimates the true strength of the plate. Current effective width formula, which is based on the postbuckling strength of the plate element in the Winter region, overestimates the strength of the panels as already shown by experimental and FEA studies. Plate elements in profiled sandwich panels generally used in many sandwich constructions lie in this intermediate region. Until recently, no design guidelines exist to deal with sandwich panels in this intermediate region. Buckling and ultimate strength behaviour of this type of foam-supported plate elements need to be investigated and a new design rule that can be applied for all practical sandwich panels including those in the intermediate region has to be developed. The following section presents the investigation conducted to study the buckling and ultimate strength behaviour of foam supported steel plates using experiments and corresponding finite element analyses.

4.6 Strength Behaviour of the Plates in the Intermediate Region

With the fast advancement in manufacturing technology, thinner and high strength steel plates are being used in many structural and building systems. As they are very economical and exhibit a very high strength to weight ratio, their popularity has increased considerably among the designers and manufacturers of the Australian construction industry. Therefore thinner and high strength steel plates are also widely used in sandwich panel construction. Because of the use of such thin plates, the plate elements in sandwich panels are becoming more slender. Generally the plate elements in fully profiled sandwich panels are subjected to local buckling effects under compression or bending actions. The current effective width formula (CIB, 2000) to treat the local buckling effect is only valid for $b/t$ ratios less than 100. The plate elements in profiled sandwich panels are not extremely slender and do not fail by wrinkling as the $b/t$ ratio is generally less than 600. To investigate the structural behaviour of foam-supported plate elements beyond the Winter region, experimental investigation in Chapter 3 was extended to slender plate elements up to a $b/t$ ratio of 1000. Finite element analyses were also conducted for those slender plates considered in the experiments.
4.6.1 Compression Tests of very Slender Plates

To develop the design rule applicable to slender plates, it is necessary to understand the buckling and ultimate strength behaviour of foam-supported steel plate elements with \( b/t \) ratios higher than those used in Chapter 3. Hence, a series of experiments on foam-supported steel plates with \( b/t \) ratios ranging from 500 to 1000 was conducted here using a similar experimental set-up to investigate their behaviour. Details of the test specimens used and the experimental program are given in Table 4.9.

Table 4.9 Test Program and Specimens to Investigate the Behaviour of Foam-Supported Slender Steel Plates

<table>
<thead>
<tr>
<th>Test No.</th>
<th>Plate Width ( b ) (mm)</th>
<th>Thickness (mm)</th>
<th>Measured</th>
<th>( b/t ) Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Spec. bmt ( f_y ) (MPa) ( E_f ) (GPa)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>300</td>
<td>0.60 0.60 682 235</td>
<td>500.0</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>260</td>
<td>0.42 0.42 726 239</td>
<td>619.1</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>300</td>
<td>0.42 0.42 726 239</td>
<td>714.3</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>340</td>
<td>0.42 0.42 726 239</td>
<td>809.5</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>380</td>
<td>0.42 0.42 726 239</td>
<td>904.8</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>420</td>
<td>0.42 0.42 726 239</td>
<td>1000.0</td>
<td></td>
</tr>
</tbody>
</table>

\( f_y \) – measured yield stress of steel, \( E_f \) – measured Young’s modulus
\( b/t \) ratio – plate width \( b/bmt \), Spec. – specified thickness
\( bmt \) – estimated base metal thickness based on measured total coated thickness

Steel plates of all the test specimens used were made from G550 grade steel. The same mechanical properties of foam core and steel plates as used in Chapter 3 were used in this investigation. A foam core with a constant thickness of 100 mm was used in all the tests as before. As seen in Table 4.9, six compression tests were conducted. To obtain high \( b/t \) ratios, a lower thickness of 0.42 mm was used in most of the specimens.

A similar type of test rig as used in Chapter 3 (see Figure 3.5) was used to hold the test specimens. However, this test rig was very wide and tall to accommodate larger widths and heights of the specimens. Plate elements up to 420 mm width and 1300
mm length could be held in this test rig. All other properties of the test rig in relation to simulating simply supported end conditions, free rotation about the vertical edges, etc. are similar to those used in the previous test rig described in Chapter 3. Because of the limitations with the Tinius Olsen Testing Machine, plate elements with $b/t$ ratios more than 1000 could not be tested due to width and height limitations of the testing machine. The larger test rig with the dimensions described above was constructed based on the limitations of the Tinius Olsen Testing Machine. Test specimens were held in the new test rig between two loading blocks. The length of the loading blocks was made equal to the width of the test specimens.

Figure 4.15 Test Set-Up for Foam-Supported Slender Plate Specimens

Compression tests on foam-supported slender steel plate elements were conducted using a Tinius Olsen Testing Machine. The compression load was applied at a rate of 0.5 mm/min until failure of the specimen. The ultimate load which was the maximum load carried by the specimen was recorded by the testing machine. All the test set-up and procedures are the same as described in Chapter 3. Figure 4.15 shows the test arrangements and slender plate test specimens tested in the Structural Laboratory at Queensland University of Technology. Figure 4.16 shows a typical plot of load
versus deflection curve for one of the tested specimens. The ultimate strength results for all the specimens obtained from the tests are given in Table 4.10 where they are compared with the corresponding finite element analysis results.

![Graph showing load versus axial displacement curve](image)

(a) Load versus Axial Displacement Curve  
(b) Specimen after Test

**Figure 4.16 Typical Load versus Displacement Curve and Failed Specimen**

### 4.6.2 Comparison of Experimental and FEA Results for Slender Plates

All the foam supported slender plate elements tested in this experimental program were also investigated using an extensive series of finite element analyses. As already discussed and verified (see Section 4.3), sandwich panels tested in the laboratory can be simulated by finite element analysis based on the half-length model. All the slender plates tested in this section were therefore modelled and analysed using half-length models as before (see Figure 4.5). Details relating to the creation of half-length models including model geometry, boundary and load conditions are as described earlier in Section 4.3. A constant foam depth of 100 mm was used for all the FEA models. A mesh size of $10 \times 10$ mm for steel faces and $10 \times 10 \times 5$ mm for foam core was used to get satisfactory results.

The half-length model was analysed using both elastic buckling and non-linear analyses. For the non-linear analysis, the first buckling mode obtained from the
elastic buckling analysis was used as the geometric imperfection distribution shape and 10% of the plate thickness \((0.1t)\) was used as the maximum imperfection magnitude as used before. The ultimate stress of the foam-supported plate elements was obtained from the non-linear analysis of all the specimens.

**Table 4.10 Comparison of Ultimate Stresses from Experiments and Half-Length Models**

<table>
<thead>
<tr>
<th>Test No.</th>
<th>Plate Width (b) (mm)</th>
<th>Plate Thickness (t) (mm)</th>
<th>(b/t) Ratio</th>
<th>Ultimate Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Experiment</td>
</tr>
<tr>
<td>1</td>
<td>120</td>
<td>0.6</td>
<td>200.0</td>
<td>169.86</td>
</tr>
<tr>
<td>2</td>
<td>150</td>
<td>0.6</td>
<td>250.0</td>
<td>133.89</td>
</tr>
<tr>
<td>3</td>
<td>180</td>
<td>0.6</td>
<td>300.0</td>
<td>122.59</td>
</tr>
<tr>
<td>4</td>
<td>200</td>
<td>0.6</td>
<td>333.3</td>
<td>118.00</td>
</tr>
<tr>
<td>5</td>
<td>150</td>
<td>0.42</td>
<td>357.1</td>
<td>119.21</td>
</tr>
<tr>
<td>6</td>
<td>180</td>
<td>0.42</td>
<td>428.6</td>
<td>118.39</td>
</tr>
<tr>
<td>7</td>
<td>200</td>
<td>0.42</td>
<td>476.2</td>
<td>100.12</td>
</tr>
<tr>
<td>8</td>
<td>300</td>
<td>0.6</td>
<td>500.0</td>
<td>87.44</td>
</tr>
<tr>
<td>9</td>
<td>260</td>
<td>0.42</td>
<td>619.0</td>
<td>99.82</td>
</tr>
<tr>
<td>10</td>
<td>300</td>
<td>0.42</td>
<td>714.3</td>
<td>96.27</td>
</tr>
<tr>
<td>11</td>
<td>340</td>
<td>0.42</td>
<td>809.5</td>
<td>94.54</td>
</tr>
<tr>
<td>12</td>
<td>380</td>
<td>0.42</td>
<td>904.8</td>
<td>79.32</td>
</tr>
<tr>
<td>13</td>
<td>420</td>
<td>0.42</td>
<td>1000.0</td>
<td>51.02</td>
</tr>
</tbody>
</table>

The ultimate strengths obtained from the half-length models were then compared with the experimental results in Table 4.10. In this part of FEA and experimental studies, the buckling behaviour of slender plates with \(b/t\) ratio ranging from 500 to 1000 was investigated. However, in Table 4.10, experimental and FEA results of foam-supported steel plate elements with \(b/t\) ratios in the range of 200 to 500 were also included from the earlier investigation reported in Chapter 3. The reason for including the previous results was to observe and fully understand the change in buckling behaviour of the foam supported steel plate elements with increasing \(b/t\) ratios up to 1000. In Table 4.10, the ultimate strength results for Specimens 1 to 7
were taken from Chapter 3 and for Specimens 8 to 13 were obtained from this present investigation.

From Table 4.10, it can be observed that the ultimate strength results obtained from half-length models are reasonably close to the experimental results for all the test specimens including the very slender plates. The ultimate stress results obtained from the FEA for Specimen 13 did not compare well with the experimental results. In this case, the specimen failed at a lower load than expected. This may be due to the inability of the finite element model to simulate the G550 characteristics of 0.42 mm thick steel with a very slender plate as explained before.

Both experimental and FEA results indicated that the ultimate stress of the foam-supported steel plate elements is dependent on the $b/t$ ratio of the plate elements. For plates with low $b/t$ ratios, the ultimate stress is high because of the postbuckling strength of the plates as described in Chapter 3. As the plate $b/t$ ratio increases, ultimate stress decreases gradually. However, when the $b/t$ ratio is large (> 500), the ultimate stress is very low and does not vary significantly even if the $b/t$ ratio increases. For very high $b/t$ ratios (> 800), ultimate stress remains almost constant.

As seen in Table 4.10, when the $b/t$ ratio of plate element is in the range of 200 to 700, the ultimate stress is decreasing continually. But as the $b/t$ ratio increases further (say 800 to 1000), there is no significant change in the ultimate stress as seen from the FEA results. It shows that the ultimate stress of foam-supported plate elements is independent of plate width and thickness for the plate with very high $b/t$ ratio. Failure stress of very slender plates is always constant irrespective of the $b/t$ ratio. This indicates that very slender plates fail due to wrinkling and the wrinkling stress can be evaluated by the wrinkling formula using Equation 2.5.6. Using this formula, the wrinkling stress of very slender plates can be calculated as 95.17 MPa for 0.42 mm thick G550 plate and 94.64 MPa for 0.60 mm thick G550 plate, respectively. In the above investigation, it can be clearly observed that the ultimate stresses of the plate elements with a $b/t$ ratio > 800 are close to the above wrinkling stress values. Hence they are likely to fail by wrinkling. However, the ultimate stresses of the plate elements with a $b/t$ ratio less than 700 are more than the wrinkling stress. This
indicates that although these plates exhibit either very small or no post buckling strength, they do not fail by wrinkling.

**Table 4.11 Buckling and Ultimate Stresses from Half-Wave Buckle Length Model**

<table>
<thead>
<tr>
<th>Plate Width $b$ (mm)</th>
<th>Plate Thickness $t$ (mm)</th>
<th>$b/t$ Ratio</th>
<th>Half-Wave Buckle Length $a$ (mm)</th>
<th>Buckling Stress (MPa)</th>
<th>Ultimate Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>120</td>
<td>0.60</td>
<td>200.0</td>
<td>23</td>
<td>110.00</td>
<td>138.06</td>
</tr>
<tr>
<td>150</td>
<td>0.60</td>
<td>250.0</td>
<td>24</td>
<td>104.33</td>
<td>122.11</td>
</tr>
<tr>
<td>180</td>
<td>0.60</td>
<td>300.0</td>
<td>24</td>
<td>101.30</td>
<td>106.11</td>
</tr>
<tr>
<td>210</td>
<td>0.60</td>
<td>350.0</td>
<td>24</td>
<td>99.52</td>
<td>100.32</td>
</tr>
<tr>
<td>240</td>
<td>0.60</td>
<td>400.0</td>
<td>24</td>
<td>98.40</td>
<td>98.13</td>
</tr>
<tr>
<td>270</td>
<td>0.60</td>
<td>450.0</td>
<td>24</td>
<td>97.65</td>
<td>96.98</td>
</tr>
<tr>
<td>300</td>
<td>0.60</td>
<td>500.0</td>
<td>25</td>
<td>97.11</td>
<td>96.39</td>
</tr>
<tr>
<td>360</td>
<td>0.60</td>
<td>600.0</td>
<td>25</td>
<td>96.39</td>
<td>95.37</td>
</tr>
<tr>
<td>420</td>
<td>0.60</td>
<td>700.0</td>
<td>25</td>
<td>95.99</td>
<td>95.20</td>
</tr>
<tr>
<td>480</td>
<td>0.60</td>
<td>800.0</td>
<td>25</td>
<td>95.73</td>
<td>94.79</td>
</tr>
<tr>
<td>540</td>
<td>0.60</td>
<td>900.0</td>
<td>25</td>
<td>95.55</td>
<td>94.60</td>
</tr>
<tr>
<td>600</td>
<td>0.60</td>
<td>1000.0</td>
<td>25</td>
<td>95.44</td>
<td>94.36</td>
</tr>
<tr>
<td>660</td>
<td>0.60</td>
<td>1100.0</td>
<td>25</td>
<td>95.35</td>
<td>94.00</td>
</tr>
<tr>
<td>720</td>
<td>0.60</td>
<td>1200.0</td>
<td>25</td>
<td>95.30</td>
<td>94.00</td>
</tr>
<tr>
<td>900</td>
<td>0.60</td>
<td>1500.0</td>
<td>25</td>
<td>95.17</td>
<td>94.00</td>
</tr>
<tr>
<td>1200</td>
<td>0.60</td>
<td>2000.0</td>
<td>25</td>
<td>95.10</td>
<td>94.00</td>
</tr>
</tbody>
</table>

As described in the previous section, the experimental sandwich panels do not necessarily represent the actual sandwich panels used in the sandwich construction. To study the buckling and ultimate strength behaviour of realistic profiled sandwich panels with slender plates, the half-wave buckle length model was analysed for the wide range of $b/t$ ratio up to 2000. The half-wave buckle length $a$ was determined using elastic buckling analyses and a minimization process. The buckling and ultimate stresses were evaluated using buckling and non-linear analyses, respectively. The half-wave buckle length model used in this part of the study was
similar to that described in Section 4.4. The buckling stress and ultimate stress results obtained from the FEA based on the half-wave buckle length model for the foam-supported steel plate elements with \( b/t \) ratio ranging from 200 to 2000 are presented in Table 4.11.

As seen in Table 4.11, when the \( b/t \) ratio of foam-supported steel plate element is in the lower range (say < 250), the ultimate stress of the panel is higher than the buckling stress due to the presence of considerable postbuckling strength. When the plate \( b/t \) ratio increases, the postbuckling strength continually diminishes and the ultimate and buckling stresses are almost equal. For very slender plates, the failure stress is low and is almost constant. It can be clearly observed that the ultimate stress of the plate elements with a \( b/t \) ratio \( \geq 1100 \) is constant (94.00 MPa) and independent of the \( b/t \) ratio. This FEA ultimate stress value compares well with the theoretical wrinkling stress value of 94.64 MPa for 0.6 mm thick G550 steel plate. This obviously indicates the wrinkling failure of the foam-supported steel plate elements with extremely high \( b/t \) ratios. Once again the results confirmed that the failure mode of very slender plate is dominated by wrinkling. However, for the plates with \( b/t \) ratio less than 1100, the ultimate stresses are higher than the wrinkling stresses, but there is no postbuckling strength.

Summarising the results of Table 4.11, foam-supported steel plate elements exhibit considerable postbuckling strength when the \( b/t \) ratio is less than 350. For \( b/t \) ratios from 400 to 1000, no postbuckling strength can be observed, but the failure stresses are higher than the wrinkling stresses and buckling and ultimate stresses are almost equal. However, for \( b/t \) ratios of 1100 and more, failure stresses are always constant and independent of the \( b/t \) ratio. This investigation confirms that if the plate elements in fully profiled sandwich panels are very slender (\( b/t \) ratio more than 1100), strength can be evaluated by the wrinkling formula. For the slender plate elements with \( b/t \) ratio less than 1100, the wrinkling formula will underestimate the strength of sandwich panels as the failure stress is higher than the wrinkling stress. The ranges of \( b/t \) ratios, which do not produce any postbuckling strength but the failure stress (ultimate stress) is higher than the wrinkling stress can be termed as the intermediate region. In this observation, the of \( b/t \) ratios ranging from 400 to 1000, which neither
show postbuckling nor fail by wrinkling, can be considered as the intermediate region.

In the initial intermediate region, the ultimate stress is higher than the wrinkling stress, but the difference decreases considerably in the latter part of the region (700 to 1000). Therefore the failure stress of the plates in this latter part can still be determined by using the wrinkling formula. However, for the plate element with $b/t$ ratio less than 600, failure stress is higher than the wrinkling stress, and the wrinkling formula cannot be used to determine the strength of the panels in that region. The current effective width design rule is applicable only for the plate elements with $b/t$ ratios less than 100 as already examined in the previous experimental and FEA studies. No design rule exists for the safe design of profiled sandwich panels with slender plate elements in this initial part of the intermediate region. As the $b/t$ ratios of most practical sandwich panels are in the range of 100 to 600, it is necessary to develop a design rule that can be used for the profiled sandwich panels with any $b/t$ ratio up to 600. It can be concluded that for the plate element with $b/t$ ratio more than 600, the wrinkling formula can be used to determine the strength of fully profiled sandwich panels, although it is slightly conservative. Based on the results from this study, a $b/t$ ratio of 600 can be considered as a reasonable boundary to develop the new design rule for the safe design of fully profiled sandwich panels.

To achieve this objective, further FEA were undertaken to include $b/t$ ratios from 30 to 600 in addition to those reported in Section 4.4. This produced a large database covering a wider range of $b/t$ ratios for sandwich panels subject to local buckling effects. Based on these FEA results, an improved design equation has been formulated as described next.

### 4.7 Development of New Design Rules

Investigation of the local buckling behaviour of foam supported steel plate elements with simply supported longitudinal edges is an essential preliminary step towards the development of design rules for profiled sandwich panels. When foam supported steel plate elements are subjected to compression, bending, or a combination of these
actions, the plate element may buckle locally before the yield stress of the material is reached. This local buckling causes a loss of stiffness and redistribution of stresses resulting in postbuckling strength in the plates, in particular those with low $b/t$ ratios.

Uniform edge compression in the longitudinal direction prior to buckling results in a non-uniform stress distribution after buckling. Much of the load is carried by the region of the plate in the close vicinity of the edges. Thus only a fraction of the width is considered effective in resisting applied compression load. Based on this, a simplified assumption is made that the maximum edge stress acts uniformly over two strips of foam supported plate and the central region is unstressed as shown in Figure 4.17. The redistribution of stress continues until the stress at the edge reaches the yield point ($f_y$) of the steel and then the plate begins to fail.

![Figure 4.17 Definition of Effective Width of Foam-Supported Steel Plate](image)

**Figure 4.17 Definition of Effective Width of Foam-Supported Steel Plate**

Effective width $b_{eff}$ is considered as a particular width of the foam supported steel plate element which just buckles when the compressive stress reaches the yield point of the steel. Using this assumption, the value of $b_{eff}$ can be determined using the following formula (Yu, 2000):

\[
\sigma_{cr} = f_y = \frac{K \pi^2 E_f}{12(1-\nu_f^2)(b_{eff}/t)^2}
\]  

(4.7.1)

\[
b_{eff} = \sqrt{K} \sqrt{\frac{\pi^2}{12(1-\nu_f^2)}} \text{at} \left(\frac{E_f}{f_y}\right) = \sqrt{KCt} \sqrt{\frac{E_f}{f_y}}
\]  

(4.7.2)
Before buckling, the width of the plate is fully effective and hence the critical buckling stress can be determined by using the full width $b$ as follows:

$$
\sigma_{cr} = \frac{K\pi^2E_f}{12(1-\nu_f^2)(b/t)^2} \quad (4.7.3)
$$

$$
b = \sqrt{K\left(\frac{\pi^2}{12(1-\nu_f^2)}\right) t \frac{E_f}{\sigma_{cr}}} = \sqrt{KCt} \frac{E_f}{\sigma_{cr}} \quad (4.7.4)
$$

where $C = \sqrt{\frac{\pi^2}{12(1-\nu^2)}} = 0.95$ (assuming $\nu_f = 0.3$) \quad (4.7.5)

Taking the ratio of Equations 4.7.2 and 4.7.4, $b_{eff}$ and $b$ can be expressed as:

$$
\frac{b_{eff}}{b} = \sqrt{\frac{\sigma_{cr}}{f_y}} \quad (4.7.6)
$$

Equations 4.7.2 and 4.7.5 are the von Karman formulae for the design of stiffened elements developed in 1932. However, experimental investigations by Sechler (1933) and Winter (1947) showed that the term $C$ used in Equation 4.7.2 depends primarily on the non-dimensional parameter $\gamma$ expressed as below (Yu, 2000):

$$
\gamma = \frac{E_f}{f_y} \left(\frac{t}{b}\right) \quad (4.7.7)
$$

From Equation 4.7.2, the term $C$ can be rewritten as:

$$
C = \frac{b_{eff}}{t} \sqrt{\frac{f_y}{KE_f}} \quad (4.7.8)
$$

From the finite element analysis conducted in this study, effective widths $b_{eff}$ of foam-supported plate elements were determined based on the ultimate stresses. Using
Equation 4.7.8, the term $C$ was evaluated for all the specimens considered for both G550 and G250 steel plate elements. The corresponding non-dimensional parameter $\gamma$ was determined using Equation 4.7.7. All these parameters are presented in Table 4.12 for G550 steel plates and in Table 4.13 for G250 steel plates, respectively.

### Table 4.12 Evaluation of Various Parameters for Foam-Supported G550 Steel Plate Elements based on FEA Results

<table>
<thead>
<tr>
<th>$b/t$ Ratio</th>
<th>Plate Width $b$ (mm)</th>
<th>Thickness $t$ (mm)</th>
<th>Measured $f_y$ (MPa)</th>
<th>$E_f$ (GPa)</th>
<th>Buckling Coefficient $K$</th>
<th>$b_{off}/b$</th>
<th>$\gamma$</th>
<th>$C$</th>
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<td>682</td>
<td>235</td>
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<tr>
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<td>50</td>
<td>0.95</td>
<td>637</td>
<td>226</td>
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Table 4.13 Evaluation of Various Parameters for Foam-Supported G250 Steel Plate Elements based on FEA Results

<table>
<thead>
<tr>
<th>b/t Ratio</th>
<th>Plate Width b (mm)</th>
<th>Plate Thickness t (mm)</th>
<th>Measured ( f_Y ) (MPa)</th>
<th>Measured ( E_f ) (GPa)</th>
<th>Buckling Coefficient ( K )</th>
<th>( \frac{b_{eff}}{b} )</th>
<th>( \gamma )</th>
<th>C</th>
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As the finite element analyses results did not simulate the ductility characteristics of thinner G550 grade steels ($t < 0.9$ mm), the actual value of yield stress $f_y$ was used in all the calculations instead of $0.9f_y$. It was assumed that lower ultimate strengths of thinner G550 steel compression members could be separately dealt with by using $0.9f_y$ in design calculations.

A graph was plotted to establish the relationship between $C$ and $\gamma$ as shown in Figure 4.18. This is the total graph containing all the data from G250 and G550 steel plate elements. This graph shows that the relationship between $C$ and $\gamma$ for any grade of steel follows the same trend. The following equation has been developed from the graph for the term $C$ based on the finite element analysis results.

$$C = 0.322(1 + 7.32\gamma - 11.48\gamma^2 + 4.59\gamma^3) \quad (4.7.9)$$

Substituting the value of $\gamma$ from Equation 4.7.7 into Equation 4.7.9,

$$C = 0.322 \left[ 1 + 7.32 \left( \frac{t}{b} \right) \left( \frac{E_f}{f_y} \right)^{1/2} - 11.48 \left( \frac{t}{b} \right)^2 \left( \frac{E_f}{f_y} \right) + 4.59 \left( \frac{t}{b} \right)^3 \left( \frac{E_f}{f_y} \right)^{3/2} \right] \quad (4.7.10)$$
Substituting the value of $C$ in Equation 4.7.2,

$$b_{\text{eff}} = 0.322t \sqrt{K \frac{E_f}{f_y}} \left[ 1 + 7.32 \left( \frac{t}{b} \right) \left( \frac{E_f}{f_y} \right)^{1/2} - 11.48 \left( \frac{t}{b} \right)^2 \left( \frac{E_f}{f_y} \right) + 4.59 \left( \frac{t}{b} \right)^3 \left( \frac{E_f}{f_y} \right)^{3/2} \right]$$

(4.7.11)

By rearranging Equation 4.7.3,

$$\sigma_{cr} = K(0.95)^2 \frac{E_f}{(b/t)^2}$$

(4.7.12)

where

$$\sqrt{\frac{\pi^2}{12(1-\nu^2)}} = 0.95 \quad \text{(assuming } \nu_f = 0.3)$$

Equation 4.7.12 can be rearranged to express $E_f$ as follows:

$$E_f = \frac{b^2}{t^2} \frac{\sigma_{cr}}{K(0.95)^2}$$

(4.7.13)

Substituting the value of $E_f$ in Equation 4.7.11,

$$b_{\text{eff}} = 0.322b \left( \frac{\sigma_{cr}}{(0.95)^2 f_y} \right)^{1/2} \left[ 1 + 7.32 \left( \frac{\sigma_{cr}}{K(0.95)^2 f_y} \right)^{1/2} - 11.48 \left( \frac{\sigma_{cr}}{K(0.95)^2 f_y} \right)^{3/2} + 4.59 \left( \frac{\sigma_{cr}}{K(0.95)^2 f_y} \right)^{3/2} \right]$$

(4.7.14)

$$\frac{b_{\text{eff}}}{b} = 0.34 \left( \frac{\sigma_{cr}}{f_y} \right)^{1/2} \left[ 1 + 7.71 \left( \frac{\sigma_{cr}}{Kf_y} \right)^{1/2} - 12.72 \left( \frac{\sigma_{cr}}{Kf_y} \right) + 5.35 \left( \frac{\sigma_{cr}}{Kf_y} \right)^{3/2} \right]$$

(4.7.15)
By substituting \( \lambda \sigma = \sqrt{f_y / \sigma_{cr}} = \lambda \), Equation 4.7.15 reduces to

\[
\frac{b_{\text{eff}}}{b} = \frac{0.34}{\lambda} \left[ 1 + \frac{7.71}{\lambda \sqrt{K}} - \frac{12.72}{\lambda^2 K} + \frac{5.35}{\lambda^3 \sqrt{K}} \right]
\]  

(4.7.16)

Equation 4.7.16 is the new modified effective width formula for computing the effective width \( b_{\text{eff}} \) for foam supported plate elements subject to local buckling effects. The final form of this equation can be expressed as:

\[
\rho = \frac{b_{\text{eff}}}{b} = \frac{0.34}{\lambda} \left[ 1 + \frac{7.71}{\beta} - \frac{12.72}{\beta^2} + \frac{5.35}{\beta^3} \right]
\]  

(4.7.17)

where

\[
\lambda = \sqrt{\frac{f_y}{\sigma_{cr}}} = 1.052 \left[ \frac{b}{t} \right] \left[ \frac{f_y}{E_f K} \right]
\]  

(4.7.18)

\[
\beta = 1.052 \left[ \frac{b}{t} \right] \left[ \frac{f_y}{E_f} \right] = \lambda \sqrt{K}
\]  

(4.7.19)

if

\[
\rho \geq 1, \quad b_{\text{eff}} = b
\]

\[
\rho < 1, \quad b_{\text{eff}} = \rho b
\]

(4.7.20)

Alternatively, a simpler design formula with slightly reduced accuracy can also be developed based on the same procedure as mentioned above. The following simpler equation can be derived for the term \( C \) from the graph given in Figure 4.18.

\[
C = 0.3375(1 + 6.12 \gamma - 7.12 \gamma^2)
\]  

(4.7.21)

Substituting the value of \( \gamma \) from Equation 4.7.7 into Equation 4.7.21,
\[ C = 0.3375 \left[ 1 + 6.12 \left( \frac{t}{b} \right) \left( \frac{E_f}{f_y} \right)^{1/2} - 7.12 \left( \frac{t}{b} \right)^2 \left( \frac{E_f}{f_y} \right) \right] \quad (4.7.22) \]

By substituting the value of \( C \) in Equation 4.7.2, a simpler modified formula for computing the effective width \( b_{eff} \) for foam-supported steel plate elements can be obtained.

\[ \rho = \frac{b_{eff}}{b} = \frac{0.355}{\lambda} \left[ 1 + \frac{6.44}{\beta} - \frac{7.89}{\beta^2} \right] \quad (4.7.23) \]

where \( \lambda \) and \( \beta \) are as defined in Equations 4.7.18 and 4.7.19, respectively. Equation 4.7.23 can be used in design instead of Equation 4.7.17 with slightly reduced accuracy.

### 4.8 Validation of New Design Rules

Since the current effective width approach is inadequate for the profiled sandwich panels with slender plates, an improved effective width design formula (Equations 4.7.17 and 4.7.23) for the plate elements supported by foam core as used in fully profiled sandwich panels was developed in the previous section considering the ultimate strength due to composite action of foam and steel face. In developing the design rule, both low and high strength steel grades (G250 and G550) were used so that the design formula can be applied to any steel grades available. Also the new design rule can be applied for any practical profiled sandwich panels with plate \( b/t \) ratios less than 600.

To examine the reliability and accuracy of the new design rule, the effective widths for different grades (G550 & G250) of foam supported steel plate elements obtained from finite element analyses were compared with the prediction made by the new improved design Equation 4.7.17. The effective width results for the foam supported steel plate elements with wide ranging \( b/t \) ratios are presented in Tables 4.14 and 4.15 for G550 and G250 steel plates, respectively. From these tables it can be
observed that the effective widths values $b_{\text{eff}}$ predicted from the new the effective width design rule are in very good agreement with the FEA results for a very wide range of $b/t$ ratios. These agreements can further be visualised clearly from the effective width plots against the $b/t$ ratio as shown in Figures 4.19 (a) and (b) for G550 and G250 steel plates, respectively.

Table 4.14 Comparison of Effective Widths from FEA and New Design Equation for G550 Steel Plates

<table>
<thead>
<tr>
<th>$b/t$ Ratio</th>
<th>Measured</th>
<th>$b_{\text{eff}}/b$ (FEA)</th>
<th>$b_{\text{eff}}/b$ New Design Equation (4.7.17)</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>$f_y$ (MPa)</td>
<td>$E_f$ (GPa)</td>
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<tr>
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<td>235</td>
<td>0.94</td>
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<tr>
<td>35.0</td>
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<td>682</td>
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<td>637</td>
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<tr>
<td>62.5</td>
<td>656</td>
<td>230</td>
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</tr>
<tr>
<td>83.3</td>
<td>682</td>
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<td>84.2</td>
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<tr>
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Table 4.15 Comparison of Effective Widths from FEA and New Design Equation for G250 Steel Plates

<table>
<thead>
<tr>
<th>(b/t) Ratio</th>
<th>Measured</th>
<th>(\frac{b_{\text{eff}}}{b}) (FEA)</th>
<th>(\frac{b_{\text{eff}}}{b}) New Design Equation (4.7.17)</th>
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<tr>
<td>53.8</td>
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<tr>
<td>137.0</td>
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<td>161.3</td>
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<td>326</td>
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<td>222.2</td>
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<td>256.4</td>
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<td>512.8</td>
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Figure 4.19 Effective Widths from FEA and New Design Rule
The new design equation predicted accurate values of effective widths for any plate slenderness simulating that of compact plates with very low $b/t$ ratios to slender plates with very high $b/t$ ratios. Hence these comparisons confirmed that the new design formula (Equation 4.7.17) can be used for the safe design of fully profiled sandwich panels subject to local buckling effects for any practical plate slenderness.

### 4.9 Summary

An extensive series of finite element analyses was conducted to investigate the local buckling behaviour of foam-supported steel plate elements as used in the fully profiled sandwich panels. A series of elastic buckling and non-linear analyses was undertaken using two different types of finite element models. The first model was the half-length model analysed to validate the finite element model using the experimental results whereas the second model was the half-wave buckle length model analysed to simulate the real conditions of the sandwich panels used in building structures. The half-wave buckle length model was used to review the current design rules, to determine their applicability to foam-supported slender steel plate elements, and to develop new improved design formula. The finite element study revealed the inadequacy of using the conventional effective width approach for slender plates. The investigation indicated that for low $b/t$ ratios (<100) current effective width design rules can be applied, but for slender plates these rules can not be extended in their present form. Based on the results from the finite element study, an improved design equation has been developed by considering the local buckling, postbuckling and ultimate strength behaviour of sandwich panels for a wider range of $b/t$ ratios up to 600 to obtain safe design solutions.

The finite element model used in this study did not simulate the effects of reduced ductility for thinner G550 grade steels. Therefore the final outcomes do not include such effects on the member ultimate strengths. These effects can be dealt with by using a reduced yield stress of $0.9f_y$ for G550 steels with a thickness less than 0.9 mm as recommended by Yang and Hancock (2002). The new improved design rules (Equations 4.7.17 and 4.7.23) are applicable to such thinner G550 steels if a reduced yield stress of $0.9f_y$ is used in the design calculations.
CHAPTER 5.0  FULL-SCALE TESTS OF PROFILED SANDWICH PANELS

5.1 General

An extensive study using a series of laboratory experiments (Chapter 3) and corresponding numerical analyses (Chapter 4) on foam-supported thin steel plates has confirmed that the conventional effective width principles used in the current design documents (CIB 2000) can not be extended for the slender plate elements with higher width to thickness (b/t) ratios. Steel plate elements generally used in fully profiled sandwich panels are slender. The lack of design rules and standards for sandwich panels with such slender plate elements is of concern to the designers. To solve this problem, a new modified design rule applicable to a wider range of b/t ratios up to 600 was developed based on the results from a validated finite element model as outlined in Chapter 4. To examine the accuracy of the new design rule, a series of full-scale experiments on fully profiled sandwich panels was conducted in the Structural Laboratory at QUT. In the full-scale experimental investigation, fully profiled sandwich panels were subjected to a uniformly distributed wind pressure loading until failure using an available vacuum chamber. Experimental failure pressures were than compared with the predictions from the new improved and current design rules.

In this chapter, the current and new improved design rules are presented briefly in order to make the referencing easier. Details of the full-scale experiments on fully profiled sandwich panels including the types of test specimens, test program, test set-up, and test results are discussed. Comparison of test results with the predictions from current and new design rules are also presented. The results and findings of the study are discussed in detail.

5.2 Current and Modified Design Rules

European recommendation for sandwich panels, Part 1: design (CIB, 2000) states that if the outermost plate element in a fully profiled sandwich panel is in
compression and the width to thickness \((b/t)\) ratio exceeds the limit given in Equation 5.2.1, it will be subject to local buckling effects and due regard should be given to this phenomenon in the design of sandwich panels.

\[
\frac{b}{t} = 1.27 \sqrt{\frac{E_f}{f_y}} \tag{5.2.1}
\]

where \(f_y\) = yield stress and \(E_f\) = Young’s modulus of the steel face. In practice, the \(b/t\) ratios of the flat plates of the steel face in the profiled sandwich panels are normally higher than the limit specified by Equation 5.2.1 and hence, they are always susceptible to local buckling effects when subjected to compression, bending or their combinations. The compression strength \(f_{Fc}\) of the profiled faces can be evaluated using the following formula:

\[
f_{Fc} = \frac{b_{eff}}{b} f_y \tag{5.2.2}
\]

where \(b_{eff}\) is the effective width of flat components of a face profile and determined by using the effective width approach given in Equation 5.2.3.

\[
\frac{b_{eff}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] \tag{5.2.3}
\]

where \(\lambda\) is the modified slenderness parameter with a modified buckling coefficient \(K\). CIB (2000) recommends that \(K\) should be calculated using Equation 3.4.3 proposed by Davies and Hakmi (1990), and Davies et al. (1991).

The procedure based on Equations 5.2.3 is the current design rule included in the “European Recommendation for Sandwich Panels, Part 1: Design” (CIB, 2000) for the fully profiled sandwich panels subjected to local buckling effects. However, this study as described in Chapters 3 and 4 has confirmed that the design method included in CIB (2000) can only be applied for plate elements with a \(b/t\) ratio less than 100. Based on the experimental investigations and the large amount of data
obtained from the finite element studies, a new improved effective width design formula was formulated for the profiled sandwich panels to estimate accurate values of effective widths that can be used for design purposes for all fully profiled sandwich panels with any practical $b/t$ ratio. This new effective width formula as given in Equation 5.2.4 can be used for a wider range of $b/t$ ratios commencing from very compact to very slender ($b/t < 600$) foam-supported steel plate elements.

$$\frac{b_{\text{eff}}}{b} = \frac{0.34}{\lambda} \left[ 1 + \frac{7.71}{\beta} - \frac{12.72}{\beta^2} + \frac{5.35}{\beta^3} \right]$$

(5.2.4)

where $\lambda$ is the same as in Equation 3.5.1 and $\beta$ is expressed as:

$$\beta = \lambda \sqrt{K}$$

(5.2.5)

The buckling coefficient $K$ can be evaluated either using Equation 3.4.1 proposed by Davies and Hakmi (1990) or Equation 3.4.2 proposed by Mahendran and Jeevaharan (1999). A simpler design formula with slightly reduced accuracy has also been developed as given in Equation 5.2.6 and can be used instead of Equation 5.2.4.

$$\frac{b_{\text{eff}}}{b} = \frac{0.355}{\lambda} \left[ 1 + \frac{6.44}{\beta} - \frac{7.89}{\beta^2} \right]$$

(5.2.6)

As explained in the last chapter, the new effective width formula is equally applicable to both the lower and high strength steels (G250 and G550) provided the appropriate reduction factor of 0.9 is applied to the yield stress of G550 steels with thickness less than 0.9 mm.

To examine the accuracy of the new effective width design rule (Equation 5.2.4) and to investigate further the inadequacy of the current effective width design rule (Equations 5.2.3) for slender plates, a series of full scale tests on fully profiled sandwich panels subjected to uniformly distributed wind pressure loading were conducted. Details of the experimental procedure and the results are given next.
5.3 Full-Scale Experimental Investigation

5.3.1 Test Specimens and Test Program

Sandwich panels made of thin cold-formed steel faces and polystyrene foam core bonded together using separate adhesives were used in this study. The polystyrene foam used in the panels was classified as SL grade (density 13.5 kg/m$^3$). The top steel face was profiled whereas the bottom steel face was flat. Two different types of sandwich panels (Type A and Type B) were used in the experimental investigation. In Type A sandwich panels, only the flat parts of steel face were supported by foam core and the profiled parts were unsupported (see Figure 5.1 (a)). In Type B sandwich panels, both the flat and profiled parts of steel face were supported by foam core (see Figure 5.1 (b)).

Table 5.1 presents the details of various elements of sandwich panels used in the experiments. This table contains the properties of both types of sandwich panels including base metal thickness (bmt), grades of top and bottom steel faces, and thickness and grade of polystyrene foam core. As seen from the table, the top steel face in Type A sandwich panels was made of 0.42 mm G550 steel and the bottom steel face was made of 0.60 mm G300 steel. Similarly, in Type B sandwich panels, the top steel face was made of 0.42 mm G550 steel and the bottom steel face was made of 0.40 mm G250 steel. The profiled depth (rib height) of the panels in both types was 28 mm. The foam thickness in all the panels was 50 mm as shown in Figure 5.1.

A total of six full-scale profiled sandwich panels was tested in this experimental study. Three of them were Type A panels and the remaining three were Type B panels. Table 5.2 presents the details of six full-scale test panels considered in this study and the experimental program. This includes types, spans and widths of the panels. The spans of the panels varied from 2200 mm to 3300 mm. The width of all the Type A panels was 855 mm and that of Type B panels was 466 mm.
Note: all dimensions are in mm.

**Figure 5.1 Types of Tested Sandwich Panels**

**Table 5.1 Details of Tested Sandwich Panels**

<table>
<thead>
<tr>
<th>Elements</th>
<th>Type A</th>
<th>Type B</th>
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<tr>
<td></td>
<td>Thickness</td>
<td>Grade</td>
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<tr>
<td>Top Steel Face</td>
<td>0.42 mm</td>
<td>G550</td>
</tr>
<tr>
<td>Bottom Steel Face</td>
<td>0.60 mm</td>
<td>G300</td>
</tr>
<tr>
<td>Polystyrene Foam Core</td>
<td>50.00 mm</td>
<td>SL</td>
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<tr>
<td>base metal thickness (bmt)</td>
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### Table 5.2 Details of Test Programs

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<th>Overall Width $B$ (mm)</th>
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<td>2</td>
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<tr>
<td>6</td>
<td>B</td>
<td>2800</td>
<td>466</td>
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</table>

#### 5.3.2 Test Set-Up and Procedure

The bending tests of six full-scale profiled sandwich panels under a uniform wind pressure loading were conducted in the Structural Laboratory at Queensland University of Technology. A large vacuum chamber (rectangular air box) was used to simulate the uniformly distributed transverse wind pressure loading on the underside of the sandwich panels. This arrangement produced a compressive stress in the top steel face and a tensile stress in the bottom steel face. The top profiled face in compression was thus subjected to a local buckling failure. A photograph showing the test set-up is given in Figure 5.2.

![Figure 5.2 Test Set-Up](image)
A detailed schematic diagram of the test set-up showing the vacuum chamber and the test panel is shown in Figure 5.3. As shown in Figure 5.3, test panels were simply supported over 70 mm wide RHS beams and were not restrained by the timber casing. The RHS beams were kept over the wooden base in order to acquire the
required height. These beams could be easily moved to vary the length between the supports to provide the required span of the panels. In each test, the profiled steel face was placed on the top so that it was subjected to compressive stresses. Once the panel was positioned in the vacuum chamber, the remaining space of the vacuum chamber was filled with polystyrene foam core. A polythene sheet was placed loosely over the panel and the entire top surface of the vacuum chamber in order to make the chamber air tight.

A vacuum pump was used to produce the suction pressure in the chamber. A total of five linear variable displacement transducers (LVDTs) were used to measure the deflections at midspan and at the supports while a pressure transducer was used to measure the suction pressure in the chamber. Three LVDTs were used at midspan and one each was used at the two supports. The LVDTs at the support were used to monitor any local deformation and to ensure that the deflection at the support is negligible. The pressure transducers and LVDTs were connected to a computer to enable continuous data acquisition until panel failure. During the test, the pressure applied to the panel was increased slowly until the panel collapsed under bending.

5.3.3 Test Observations and Discussions

For all the panels tested, bending failure occurred in the vicinity of midspan, that is, at the location of greatest bending moment. With the continuous application of wind pressure, the top profiled face, which was subjected to a compressive stress, first buckled locally, then developed postbuckling strength, and collapsed when it reached the ultimate load. As expected, the top steel plate element of the profiled face failed first due to yielding as it was located further away from the neutral axis. Figure 5.4 (a) shows the typical failure pattern of the Type A sandwich panel after the test while Figure 5.4 (b) shows the close-up view of the failed panel. Figure 5.4 (c) shows the typical failure pattern of Type B sandwich panel. Figures 5.5 (a) to (f) show typical plots of pressure versus deflection curves. The deflection shown in the plot is the mid span deflection minus the average local deformation at the supports.
(a) Failure of Sandwich Panel (Type A)

(b) Close-Up View

Figure 5.4 Typical Failure of Sandwich Panel
(c) Failure of Sandwich Panel (Type B)

Figure 5.4 Typical Failure of Sandwich Panel

(a) Type A Panel with $L = 2200$ mm

Figure 5.5 Pressure Versus Deflection Diagrams
(b) Type A Panel with $L = 2800$ mm

(c) Type A Panel with $L = 3300$ mm

Figure 5.5 Pressure Versus Deflection Diagrams
(d) Type B Panel with $L = 2200$ mm

(e) Type B Panel with $L = 2550$ mm

Figure 5.5 Pressure Versus Deflection Diagrams
As explained earlier, six full-scale profiled sandwich panels with two different cross-sections (Type A and Type B) were tested in this investigation to examine the new design rule and to determine the applicability and reliability of the current design rule (CIB, 2000). The measured experimental failure pressure was compared with the predictions from the new design rule (Equation 5.2.4) and CIB (2000) design rule (Equation 5.2.3). For both design rules, experimentally measured values of mechanical properties of steel faces and foam core as given in Chapter 3 were used to determine the various design parameters such as $R$, $K$ and $\lambda$. These measured values for SL grade foam core are $E_c = 3.80$ MPa and $G_c = 1.76$ MPa. Similarly, the measured values for 0.42 mm thick G550 grade steel are $E_f = 239$ GPa and $f_y = 726$ MPa. Poisson’s ratio of steel was taken as 0.3. The reduction factor of 0.9 was applied to the yield stress of 0.42 mm thick G550 steel while calculating the effective width as recommended by Yang and Hancock (2002). Since the full-scale panels were purchased from the sandwich panel manufacturers, it was difficult to obtain the required tensile coupons from the panels as the steel plates were firmly attached to the foam core. Because of this difficulty, Jeevaharan’s (1997) test result of $f_y = 726$
MPa was used. However, a sensitivity study was conducted on 0.42 mm G550 grade steel with yield stresses from 700 to 800 MPa, which confirmed that small variations in yield stress does not make any significant changes to the strength results and the final conclusions. Results of the sensitivity study are presented later in Table 5.4.

For predicting the failure pressure of the sandwich panels, it is necessary to determine the effective cross-sectional area and effective second moment of area of the panels. Both effective cross-sectional area and second moment of area are evaluated using the effective widths of the compressive steel face (profiled face) of the section. Hence, the determination of the effective widths of the whole section is the preliminary step to calculate the failure pressure of sandwich panels.

In the effective width calculations for profiled steel face, lightly profiled ridges (< 1.0 mm height) located between the fully profiled ridges (see Figure 5.1) were ignored as their effects on local buckling were considered small. It was assumed that only the ridge part reaches yield stress level. The lower stress in the flat part was then calculated based on the yield stress in the ridge part. For Type A sandwich panels, the effective widths of profiled ridge plates without any foam support were evaluated using the standard effective width formula for plain plates using $K = 4.0$. The effective widths of foam-supported flat plates were evaluated using both the current CIB (2000) design rule (Equation 5.2.3) and the improved design rule (Equation 5.2.4). For Type B sandwich panels, the effective widths of plate elements in both the profile ridges and flat parts were evaluated using the latter method as both of them are fully supported by foam core.

It is necessary that the effect of the stress gradient is considered in calculating the effective width of the inclined steel plate elements of the profiled ridge part. The stress gradient increases the effective width of the plate elements by enhancing the buckling coefficient of foam-supported steel plate elements. AS/NZS 4600 (SA, 1996) has recommended an explicit formula to determine the increased buckling coefficient $K_{inc}$ for the inclined plate elements without any foam support. The expression to find $K_{inc}$ due to a stress gradient is given by the following equation from AS/NZS 4600.
\[ K_{inc} = 4 + 2(1-\Psi)^3 + 2(1-\Psi) \] \hspace{1cm} (5.4.1)

where \( \Psi \) is the ratio of the stresses at the ends of the inclined plate element. The first term in Equation 5.4.1 is the normal buckling coefficient \( K \) of the plate element with simply supported boundary conditions \( (K = 4.0) \). As inclined plate elements in Type A sandwich panels are not supported by foam core, Equation 5.4.1 recommended by AS/NZS 4600 was directly applied to calculate the increased buckling coefficient \( K_{inc} \) of this element in the sandwich panel. However, inclined elements in Type B sandwich panels are supported by foam core and AS/NZS 4600 has not recommended any formula to calculate the increased value \( K_{inc} \) for such plate elements. The following formula was adopted to calculate the increased value of \( K_{inc} \) for foam-supported steel plate elements.

\[ K_{inc} = K + 2(1-\Psi)^3 + 2(1-\Psi) \] \hspace{1cm} (5.4.2)

where \( K \) is the normal buckling coefficient of the foam-supported steel plate element (more than 4.0 because of the composite action of foam core and steel faces) since the normal buckling coefficient of plain plate element \( (K = 4.0) \) was used in Equation 5.4.1. The normal buckling coefficient \( K \) for foam-supported steel plate elements can be determined by using Equation 3.4.1 proposed by Davies and Hakmi (1990) or Equation 3.4.2 proposed by Mahendran and Jeevaharan (1999). In the effective width calculation (see Section 5.5), Equation 5.4.1 was used for Type A sandwich panels whereas Equation 5.4.2 was used for Type B sandwich panels to include the increase in buckling strength due to the stress gradient.

Figure 5.6 shows the typical effective widths of fully profiled sandwich panels used in the full-scale experiments. Based on the effective widths of the top profiled face and full width of the bottom face which is in tension, an effective second moment of area \( (I_{eff}) \) was calculated (see Figure 5.6). The small flat parts on both edges of the panel along its length were ignored while calculating the effective second moment of area as their effective widths are very small and they do not contribute much to the total strength of the panel. The failure pressure load \( w_u \) was then determined by equating the applied bending moment to the moment of resistance of the effective
The applied mid-span bending moment in the panel at failure due to a uniformly distributed wind pressure loading per unit length $w_u$ is given by:

$$M_u = \frac{w_u L^2}{8} \quad (5.4.3)$$

where $L$ is the span of the panel. Moment of resistance of the sandwich panel with an effective cross-section $M_u$ (see Figure 5.6) can be expressed as:

$$M_u = \frac{f_y}{y_{\text{max}}} I_{\text{eff}} \quad (5.4.4)$$

where $y_{\text{max}}$ is the distance between the centroid and the topmost fibre of the profiled steel face, $f_y$ is the yield stress of the steel and $I_{\text{eff}}$ is the effective second moment of area of the effective section. By equating Equations 5.4.3 and 5.4.4, the expression for uniformly distributed wind loading per unit length $w_u$ at failure can be written as:

$$w_u = \frac{8f_y I_{\text{eff}}}{y_{\text{max}} L^2} \quad (5.4.5)$$

The wind load per unit length $w_u$ was converted to a failure pressure $p_u$ by using Equation 5.4.6.
where $B$ is the overall panel width. A detailed derivation of the applied pressure using both the current and modified design rules for both Type A and Type B panels based on the above mentioned procedure is given in the following section.

5.5 Effective Width Calculations

5.5.1 Calculations using the New Design Rule

(i) Type A Sandwich Panel

Mechanical properties of various elements of Type A sandwich panels

Top steel face: $t = 0.42$ mm (G550), $E_f = 239$ GPa, $v_f = 0.3$

$f_y = 90\%$ of 726 MPa = 653.4 MPa

Bottom steel face: $t = 0.60$ mm (G300)

Foam Core: $E_c = 3.8$ MPa, $G_c = 1.76$ MPa

(a) Iteration 1: The neutral axis of the panel section is calculated by a trial and error method. For the first trial, consider that elements 1 and 2 are fully effective and element 3 is not fully effective (see Figure 5.7). For dimensions, see Figure 5.1 (a).

Determining the effective width of element 3:
Since element 3 is not supported by foam core, its effective width is calculated by using the standard effective width equation with $K = 4$.

$$\lambda = 1.052 \left( \frac{b}{t} \right) \left[ \frac{f_y}{E_f K} \right] = 1.052 \times 61.90 \times \sqrt{\frac{653.4}{239000 \times 4}} = 1.703 > 0.673$$

$$\frac{b_{eff}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.511 \quad b_{eff} = 0.511 \times 26 = 13.30 \text{ mm} \quad b_{eff}/2 = 6.65 \text{ mm}$$

**Determining the centroid of composite section from the bottom steel face:**

$$y = \frac{37.2 \times 0.42 \times 64 \times 8 + 6.65 \times 0.42 \times 78 \times 8 + 175 \times 0.42 \times 50 \times 3}{37.2 \times 0.42 \times 8 + 6.65 \times 0.42 \times 8 + 175 \times 0.42 \times 3 + 855 \times 0.6} = 23.58 \text{ mm}$$

Based on above centroid position of the composite section, effective widths of elements 2 and 3 are determined to check whether the previous assumption is correct.

**Determining the effective width of element 2**

Element 2 is a stiffened element with a stress gradient. The effective width of such a section is determined in accordance with Clause 2.2.3.2 of AS/NZS 4600 (SA, 1996). The calculations are given next:

$$f_1^* = 653.4 \text{ MPa} \quad f_2^* = 653.4 \times \frac{26.42}{54.42} = 317.21 \text{ MPa} \quad \Psi = \frac{f_2^*}{f_1^*} = \frac{317.21}{653.4} = 0.485$$

$$K_{inc} = 4 + 2(1 - \Psi)^3 + 2(1 - \Psi) = 5.30$$

$$\lambda = 1.052 \left( \frac{b}{t} \right) \left[ \frac{f_1^*}{E_f K_{inc}} \right] = 1.052 \times \frac{37.2}{0.42} \times \sqrt{\frac{653.4}{239000 \times 5.3}} = 2.116$$

$$\frac{b_{eff}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.423 \quad b_{eff} = 0.423 \times 37.2 = 15.74 \text{ mm}$$

Since the effective width of inclined steel element is less than the total width, element 2 is not fully effective.
Determining the effective width of element 1

Since element 1 is supported by foam core, its effective width is determined using the new design rule as follows:

\[
R = \frac{12(1 - \nu^2)}{\pi^3} \sqrt{\frac{E \cdot G_c}{E_f}} \left[ \frac{b}{t} \right]^3 = \frac{12(1 - 0.3^2)}{\pi^3} \times \frac{\sqrt{3.8 \times 1.76}}{239000} \times \left[ \frac{175}{0.42} \right]^3 = 275.67
\]

\[
K = \left[ 16 + 11.8R + 0.055R^2 \right]^{1/2} = 86.31 \quad \lambda = 1.052 \left[ \frac{b}{t} \right] \sqrt{\frac{f_2^*}{E_f K}} = 1.719
\]

\[
\beta = \lambda \sqrt{K} = 15.97 \quad \frac{b_{eff}}{b} = 0.34 \left[ \frac{1}{\lambda} \right] + \frac{7.71}{\beta} - \frac{12.72}{\beta^2} + \frac{5.35}{\beta^3} = 0.284
\]

\[
b_{eff} = 0.284 \times 175 = 49.70 \text{ mm} \quad b_{eff}/2 = 24.85 \text{ mm}
\]

Determining the location of the centroid of composite section from the bottom steel face using the revised effective widths:

\[
\begin{align*}
\bar{y} &= \frac{8(9.48 \times 0.42 \times 53.565 + 6.26 \times 0.42 \times 75.645 + 6.65 \times 0.42 \times 78) + 24.85 \times 0.42 \times 50 \times 6}{8(9.48 + 6.26 + 6.65) \times 0.42 + 24.85 \times 0.42 \times 6 + 555 \times 0.6} = 12.55 \text{ mm}
\end{align*}
\]

(b) Iteration 2

Determining the effective width of element 2 (see Figure 5.8)

\[
f_1^* = 653.4 \text{ MPa} \quad f_2^* = 653.4 \times \frac{37.45}{65.45} = 373.87 \text{ MPa} \quad \Psi = \frac{f_2^*}{f_1^*} = \frac{373.87}{653.4} = 0.572
\]

\[
K_{inc} = 4 + 2(1 - \Psi)^3 + 2(1 - \Psi) = 5.012 \quad \lambda = 1.052 \left[ \frac{b}{t} \right] \sqrt{\frac{f_2^*}{E_f K_{inc}}} = 2.176
\]

\[
\frac{b_{eff}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.413 \quad b_{eff} = 0.413 \times 37.2 = 15.36 \text{ mm}
\]
\[ b_{eff} = \frac{b_{eff}}{3 - \Psi} = \frac{15.36}{3 - 0.572} = 6.33 \text{ mm} \]

\[ b_{eff2} = b_{eff} - b_{eff1} = 15.36 - 6.33 = 9.03 \text{ mm} \]

**Figure 5.8 Idealized Effective Section of Type A Panel for Iteration 2**

Determining the effective width of element 1

\[ R = 275.67 \quad K = 86.31 \quad \lambda = 1.052 \left[ \frac{b}{t} \sqrt{\frac{f_y^2}{E_f K}} \right] = 1.866 \]

\[ \beta = \lambda \sqrt{K} = 17.34 \quad \frac{b_{eff}}{b} = 0.34 \left[ \frac{1 + 7.71}{\beta} - \frac{12.72}{\beta^2} + \frac{5.35}{\beta^3} \right] = 0.256 \]

\[ b_{eff} = 0.256 \times 175 = 44.80 \text{ mm} \quad b_{eff}/2 = 22.40 \text{ mm} \]

Determining the location of the centroid of composite section from the bottom steel face using the revised effective widths:

\[ \bar{y} = \frac{8(9.03 \times 0.42 \times 53.40 + 6.33 \times 0.42 \times 75.62 + 6.65 \times 0.42 \times 78) + 22.4 \times 0.42 \times 50 \times 6}{8(9.03 \times 0.42 + 6.33 \times 0.42 + 6.65 \times 0.42) + 22.4 \times 0.42 \times 6 + 855 \times 0.6} = 12.11 \text{ mm} \]

\( \bar{y} \) (12.11 mm) from iteration 2 is close to \( \bar{y} \) (12.55 mm) from iteration 1, however, for more accurate results, consider one more iteration.

(c) Iteration 3
Determining the effective width of element 2

\[ f_1^* = 653.4 \text{ MPa} \quad f_2^* = 653.4 \times \frac{37.89}{65.89} = 375.74 \text{ MPa} \quad \Psi = \frac{f_2^*}{f_1^*} = \frac{375.74}{653.4} = 0.575 \]

\[ K_{inc} = 4 + 2(1 - \Psi)^3 + 2(1 - \Psi)^2 = 5.003 \quad \lambda = 1.052 \left[ \frac{b}{t} \right] \left[ \frac{f_1^*}{E_f K_{inc}} \right] = 2.178 \]

\[ \frac{b_{eff}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.413 \quad b_{eff} = 0.413 \times 37.2 = 15.36 \]

\[ b_{eff1} = \frac{b_{eff}}{3 - \Psi} = 6.33 \quad b_{eff2} = b_{eff} - b_{eff1} = 9.03 \]

Determining the effective width of element 1

\[ R = 275.67 \quad K = 86.31 \quad \lambda = 1.052 \left[ \frac{b}{t} \right] \left[ \frac{f_2^*}{E_f K} \right] = 1.871 \]

\[ \beta = \lambda \sqrt{K} = 17.38 \quad \frac{b_{eff}}{b} = \frac{0.34}{\lambda} \left[ 1 + \frac{7.71}{\beta^2} - \frac{12.72}{\beta^3} + \frac{5.35}{\beta^3} \right] = 0.2549 \]

\[ b_{eff} = 0.2549 \times 175 = 44.60 \text{ mm} \quad b_{eff}/2 = 22.30 \text{ mm} \]

Determining the location of the centroid of composite section from the bottom steel face using the revised effective widths:

\[ \bar{y} = \frac{8(9.03 \times 0.42 \times 53.40 + 6.33 \times 0.42 \times 75.62 + 6.65 \times 0.42 \times 78) + 22.3 \times 0.42 \times 50 \times 6}{8(9.03 \times 0.42 + 6.33 \times 0.42 + 6.65 \times 0.42) + 22.3 \times 0.42 \times 6 + 855 \times 0.6} = 12.10 \text{ mm} \]

The centroid position \( \bar{y} \) (12.10 mm) from iteration 3 agrees with \( \bar{y} \) (12.11 mm) from iteration 2. These new values of effective widths are therefore adopted as final values.

Determining the effective moment of area of the reduced effective section using the new effective width values:
$I_{ef} = \left[ \frac{9.03^3 \times 0.42}{12} \sin^2 48.81 + 9.03 \times 0.42 \times 41.30^2 \right] \times 8$

$+ \left[ \frac{6.33^3 \times 0.42}{12} \sin^2 48.81 + 6.33 \times 0.42 \times 63.52^2 \right] \times 8$

$+ \left[ \frac{6.65 \times 0.42^3}{12} + 6.65 \times 0.42 \times 65.90^2 \right] \times 8$

$+ \left[ \frac{22.30 \times 0.42^3}{12} + 22.30 \times 0.42 \times 37.9^2 \right] \times 6 + \left[ \frac{855 \times 0.6^3}{12} + 855 \times 0.6 \times 12.1^2 \right]$

$I_{ef} = 390605.24 \text{ mm}^4$

**Determining the failure pressure $p_u$**

**Case 1:** Span $L = 2200$ mm

$p_u = \frac{8 f_i I_{ef}}{y_{\max} BL^2} = \frac{8 \times 653.4 \times 390605.24}{65.90 \times 855 \times 2200^2} = 7.49 \times 10^{-3} \text{ N/mm}^2 = 7.49 \text{ kPa}$

**Case 2:** Span $L = 2800$ mm

$p_u = \frac{8 f_i I_{ef}}{y_{\max} BL^2} = \frac{8 \times 653.4 \times 390605.24}{65.90 \times 855 \times 2800^2} = 4.62 \times 10^{-3} \text{ N/mm}^2 = 4.62 \text{ kPa}$

**Case 3:** Span $L = 3300$ mm

$p_u = \frac{8 f_i I_{ef}}{y_{\max} BL^2} = \frac{8 \times 653.4 \times 390605.24}{65.90 \times 855 \times 3300^2} = 3.33 \times 10^{-3} \text{ N/mm}^2 = 3.33 \text{ kPa}$

(ii) **Type B Sandwich Panel**

**Mechanical properties of various elements of Type B sandwich panels**

Top steel face: \( t = 0.42 \text{ mm} \) (G550), \( E_f = 239 \text{ GPa}, \nu_f = 0.3 \)

\( f_i = 90\% \) of 726 MPa = 653.4 MPa

Bottom steel face: \( t = 0.40 \text{ mm} \) (G250)

Foam Core: \( E_c = 3.8 \text{ MPa}, \ G_c = 1.76 \text{ MPa} \)
(a) **Iteration 1:** For the first trial, consider elements 1 and 2 are fully effective and element 3 is not fully effective (see Figure 5.9). For dimensions, see Figure 5.1 (b).

![Figure 5.9 Idealized Effective Section of Type B Panel for Iteration 1](image)

**Determining the effective width of element 3:**

Since element 3 is supported by foam core, its effective width is calculated by using the new design rule.

\[
R = \frac{12(1 - \nu_f^2)}{\pi^3} \sqrt{\frac{E_f G_f}{E_f}} \left(\frac{b}{t}\right)^3 = \frac{12(1 - 0.3^2)}{\pi^3} \times \sqrt{\frac{3.8 \times 1.76}{239000}} \times \left[\frac{26}{0.42}\right]^3 = 0.90
\]

\[
K = \left[16 + 11.8R + 0.055R^2\right]^{1/2} = 5.16 \quad \lambda = 1.052 \left[\frac{b}{t}\right] \frac{f_y}{E_f K} = 1.50
\]

\[
\beta = \lambda \sqrt{K} = 3.41 \quad \frac{b_{eff}}{b} = 0.34 \left[1 + \frac{7.71}{\beta} - \frac{12.72}{\beta^2} + \frac{5.35}{\beta^3}\right] = 0.52
\]

\[
b_{eff} = 0.52 \times 26 = 13.52 \text{ mm} \quad \frac{b_{eff}}{2} = 6.76 \text{ mm}
\]

**Determining the centroid of composite section from the bottom steel face:**

\[
y = \frac{33.30 \times 0.42 \times 64 \times 6 + 6.76 \times 0.42 \times 78 \times 6 + 125 \times 0.42 \times 2 + 466 \times 0.4}{33.30 \times 0.42 \times 6 + 6.76 \times 0.42 \times 6 + 125 \times 0.42 \times 2 + 466 \times 0.4} = 30.45 \text{ mm}
\]

**Determining the effective width of element 2**
Since element 2 is also supported by foam core, the effective width of this element is calculated by using the new design rule. Element 2 is a stiffened element with a stress gradient. The effective width for such a section is determined in accordance with Clause 2.2.3.2 of AS/NZS 4600 (SA, 1996). The calculations are given next:

\[
f_i^* = 653.4 \text{ MPa} \quad f_2^* = 653.4 \times \frac{19.55}{47.55} = 268.64 \text{ MPa} \quad \Psi = \frac{f_2^*}{f_i^*} = \frac{268.64}{653.4} = 0.411
\]

\[
R = \frac{12(1 - \nu_f^2)}{\pi^3} \sqrt{\frac{E_c G_c}{E_f}} \left[\frac{b}{t}\right]^3 = \frac{12(1 - \nu_f^2)}{\pi^3} \sqrt{\frac{3.8 \times 1.76}{239000}} \times \left[\frac{33.3}{0.42}\right]^3 = 1.90
\]

\[
K = \left[16 + 11.8R + 0.055R^2\right]^{1/2} = 6.21 \quad K_{mc} = 6.21 + 2(1 - \Psi)^3 + 2(1 - \Psi) = 7.797
\]

\[
\lambda = 1.052 \left[\frac{b}{t}\right] \sqrt{\frac{f_i^*}{E_f K_{mc}}} = 1.56 \quad \beta = \lambda \sqrt{K_{mc}} = 4.36
\]

\[
\frac{b_{eff}}{b} = 0.34 \left[1 + \frac{7.71}{\beta} - \frac{12.72}{\beta^2} + \frac{5.35}{\beta^3}\right] = 0.4715 \quad b_{eff} = 0.4715 \times 33.3 = 15.70 \text{ mm}
\]

Since the effective width of inclined steel element is less than the total width, element 2 is not fully effective.

\[
b_{eff1} = \frac{b_{eff}}{3 - \Psi} = \frac{15.70}{3 - 0.411} = 6.06 \text{ mm} \quad b_{eff2} = b_{eff} - b_{eff1} = 15.70 - 6.06 = 9.64 \text{ mm}
\]

**Determining the effective width of element 1**

Since element 1 is supported with foam core, its effective width is determined using the new design rule as follows:

\[
R = \frac{12(1 - \nu_f^2)}{\pi^3} \sqrt{\frac{E_c G_c}{E_f}} \left[\frac{b}{t}\right]^3 = \frac{12(1 - \nu_f^2)}{\pi^3} \sqrt{\frac{3.8 \times 1.76}{239000}} \times \left[\frac{125}{0.42}\right]^3 = 100.46
\]

\[
K = \left[16 + 11.8R + 0.055R^2\right]^{1/2} = 41.91 \quad \lambda = 1.052 \left[\frac{b}{t}\right] \sqrt{\frac{f_i^*}{E_f K}} = 1.621
\]

\[
\beta = \lambda \sqrt{K} = 10.50 \quad \frac{b_{eff}}{b} = 0.34 \left[1 + \frac{7.71}{\beta} - \frac{12.72}{\beta^2} + \frac{5.35}{\beta^3}\right] = 0.3405
\]
\[ b_{\text{eff}} = 0.3405 \times 125 = 42.56 \text{ mm} \quad b_{\text{eff}}/2 = 21.28 \text{ mm} \]

Determining the location of the centroid of composite section from the bottom steel face using the revised effective widths:

\[ y = \frac{6(9.64 \times 0.42 \times 54.055 + 6.06 \times 0.42 \times 75.45 + 6.76 \times 0.42 \times 78) + 21.28 \times 0.42 \times 50 \times 4}{6(9.64 + 6.06 + 6.76) \times 0.42 + 21.28 \times 0.42 \times 4 + 466 \times 0.4} = 20.02 \text{ mm} \]

(b) Iteration 2

Determining the effective width of element 2 (see Figure 5.10)

\[ f_1^* = 653.4 \text{ MPa} \quad f_2^* = 653.4 \times \frac{29.98}{57.98} = 337.86 \text{ MPa} \quad \Psi = \frac{f_2^*}{f_1^*} = \frac{337.86}{653.4} = 0.517 \]

\[ R = 1.90 \quad K = 6.21 \quad K_{nc} = 6.21 + 2(1 - \Psi)^3 + 2(1 - \Psi) = 7.40 \]

\[ \lambda = 1.052 \left[ \frac{b}{t} \right] \left( \frac{f_1^*}{E_f K_{nc}} \right) = 1.603 \quad \beta = \lambda \sqrt{K_{nc}} = 4.36 \]

\[ \frac{b_{\text{eff}}}{b} = \frac{0.34}{\lambda} \left[ 1 + \frac{7.71}{\beta} - \frac{12.72}{\beta^2} + \frac{5.35}{\beta^3} \right] = 0.459 \quad b_{\text{eff}} = 0.459 \times 33.3 = 15.28 \text{ mm} \]

\[ \frac{b_{\text{eff}1}}{3 - \Psi} = \frac{15.28}{3 - 0.517} = 6.15 \text{ mm} \quad b_{\text{eff}2} = b_{\text{eff}} - b_{\text{eff}1} = 15.28 - 6.15 = 9.13 \text{ mm} \]

Figure 5.10 Idealized Effective Section of Type B Panel for Iteration 2

Determining the effective width of element 1

\[ b_{\text{eff}} = 0.3405 \times 125 = 42.56 \text{ mm} \quad b_{\text{eff}}/2 = 21.28 \text{ mm} \]
\[ R = 100.46 \quad K = 41.91 \quad \lambda = 1.052 \left[ \frac{b}{t} \right] \sqrt{\frac{f^*}{E_j K}} = 1.818 \quad \beta = \lambda \sqrt{K} = 11.77 \]

\[ \frac{b_{\text{eff}}}{b} = 0.34 \left[ 1 + \frac{7.71}{\beta} - \frac{12.72}{\beta^2} + \frac{5.35}{\beta^3} \right] = 0.293 \]

\[ b_{\text{eff}} = 0.293 \times 125 = 36.62 \text{ mm} \quad b_{\text{eff}}/2 = 18.31 \text{ mm} \]

Determining the location of the centroid of composite section from the bottom steel face using the revised effective widths:

\[ \bar{y} = \frac{6(9.13 \times 0.42 \times 53.84 + 6.15 \times 0.42 \times 75.415 + 6.76 \times 0.42 \times 78) + 18.31 \times 0.42 \times 50 \times 4}{6(9.31 + 6.15 + 6.76) \times 0.42 + 18.31 \times 0.42 \times 4 + 466 \times 0.4} = 19.34 \text{ mm} \]

\[ \bar{y} (19.34 \text{ mm}) \] from iteration 2 is close to \[ \bar{y} (20.02 \text{ mm}) \] from iteration 1, however, for more accurate results, consider one more iteration.

c) Iteration 3

Determining the effective width of element 2

\[ f_1^* = 653.4 \text{ MPa} \quad f_2^* = 653.4 \times \frac{30.66}{58.66} = 341.51 \text{ MPa} \quad \Psi = \frac{f_2^*}{f_1^*} = \frac{341.51}{653.4} = 0.523 \]

\[ R = 1.90 \quad K = 6.21 \quad K_{\text{inc}} = 6.21 + 2(1 - \Psi)^3 + 2(1 - \Psi) = 7.38 \]

\[ \lambda = 1.052 \left[ \frac{b}{t} \right] \sqrt{\frac{f_1^*}{E_j K_{\text{inc}}}} = 1.605 \quad \beta = \lambda \sqrt{K_{\text{inc}}} = 4.36 \]

\[ \frac{b_{\text{eff}}}{b} = 0.34 \left[ 1 + \frac{7.71}{\beta} - \frac{12.72}{\beta^2} + \frac{5.35}{\beta^3} \right] = 0.4584 \quad b_{\text{eff}} = 0.4584 \times 33.3 = 15.26 \text{ mm} \]

\[ b_{\text{eff}} = \frac{b_{\text{eff}}}{3 - \Psi} = \frac{15.26}{3 - 0.523} = 6.16 \text{ mm} \quad b_{\text{eff} 2} = b_{\text{eff}} - b_{\text{eff} 1} = 15.26 - 6.16 = 9.10 \text{ mm} \]

Determining the effective width of element 1
\[
R = 100.46 \quad K = 41.91 \quad \lambda = 1.052 \left[ \frac{b}{t} \right] \left[ \frac{f_y^2}{E_f K} \right] = 1.828 \quad \beta = \lambda \sqrt{K} = 11.83
\]

\[
\frac{b_{eff}}{b} = 0.34 \left[ 1 + 7.71 \frac{1}{\beta^2} + 12.72 \frac{1}{\beta^3} + 5.35 \right] = 0.2909
\]

\[b_{eff} = 0.2909 \times 125 = 36.36 \text{ mm} \quad b_{eff} / 2 = 18.18 \text{ mm}\]

**Determining the location of the centroid of composite section from the bottom steel face using the revised effective widths:**

\[
\bar{y} = \frac{6(9.10 \times 0.42 \times 53.825 + 6.16 \times 0.42 \times 75.41 + 6.76 \times 0.42 \times 78)}{6(9.1 \times 0.42 + 6.16 \times 0.42 + 6.76 \times 0.42) + 18.18 \times 0.42 \times 4 + 466 \times 0.4} = 19.31 \text{ mm}
\]

The centroid position \(\bar{y}\) (19.31 mm) from iteration 3 agrees with \(\bar{y}\) (19.34 mm) from iteration 2. These new values of effective widths are therefore adopted as final values.

**Determining the effective moment of area of the reduced effective section using the new effective width values:**

\[
I_{eff} = \left[ \frac{9.10^3 \times 0.42}{12} \sin^2 57.26 + 9.10 \times 0.42 \times 34.515^2 \right] \times 6 + \left[ \frac{6.16^3 \times 0.42}{12} \sin^2 57.26 + 6.16 \times 0.42 \times 56.10^2 \right] \times 6 + \left[ \frac{6.76 \times 0.42^3}{12} + 6.76 \times 0.42 \times 58.69^2 \right] \times 6 + \left[ \frac{18.18 \times 0.42^3}{12} + 18.18 \times 0.42 \times 30.69^2 \right] \times 4 + \left[ \frac{466 \times 0.4^3}{12} + 466 \times 0.4 \times 19.31^2 \right]
\]

\[I_{eff} = 233272.48 \text{ mm}^4\]

**Determining the failure pressure \(p_u\)**
Case 1: Span $L = 2200$ mm

$$p_u = \frac{8 f_y I_{ef}}{y_{max} BL^2} = \frac{8 \times 653.4 \times 233272.48}{58.69 \times 466 \times 2200^2} = 9.21 \times 10^{-3} \text{ N/mm}^2 = 9.21 \text{ kPa}$$

Case 2: Span $L = 2550$ mm

$$p_u = \frac{8 f_y I_{ef}}{y_{max} BL^2} = \frac{8 \times 653.4 \times 233272.48}{58.69 \times 466 \times 2550^2} = 6.86 \times 10^{-3} \text{ N/mm}^2 = 6.86 \text{ kPa}$$

Case 3: Span $L = 2800$ mm

$$p_u = \frac{8 f_y I_{ef}}{y_{max} BL^2} = \frac{8 \times 653.4 \times 233272.48}{58.69 \times 466 \times 2800^2} = 5.69 \times 10^{-3} \text{ N/mm}^2 = 5.69 \text{ kPa}$$

5.5.2 Calculations using Current Design Rule (CIB, 2000)

(i) Type A Sandwich Panel

(a) Iteration 1: For the first iteration, consider elements 1 and 2 are fully effective and element 3 is not fully effective (see Figure 5.7).

Determining the effective width of element 3:

Since element 3 is not supported by foam core, its effective width is calculated by using standard effective width equation with $K = 4$. The effective width and centroid of the section from bottom steel face of element 3 of Type A panels are taken from Section 5.5.1 (i) and given below.

$$b_{eff} / 2 = 6.65 \text{ mm} \quad \bar{y} = 23.58 \text{ mm}$$

Determining the effective width of element 2

Element 2 is a stiffened element with a stress gradient. The effective width of such a section is determined in accordance with Clause 2.2.3.2 of AS/NZS 4600 (SA, 1996).
The effective width of element 2 of Type A panels from iteration 1 is taken from Section 5.5.1 (i) and given below.

\[
\begin{align*}
    f_1^* &= 653.4 \text{ MPa} \\
    f_2^* &= 317.21 \text{ MPa} \\
    b_{\text{eff}1} &= 6.26 \text{ mm} \\
    b_{\text{eff}2} &= 9.48 \text{ mm}
\end{align*}
\]

**Determining effective width for element 1**

Since element 1 is supported by foam core, its effective width is determined using CIB (2000) design rule as follows:

\[
R = 275.67 \quad K = \left[16 + 7R + 0.02R^2 \right]^{1/2} = 58.87 \quad \lambda = 1.052 \left[ \frac{b}{t} \right] \sqrt{\frac{f_2^*}{E_f K}} = 2.081
\]

\[
\frac{b_{\text{eff}}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.4297 \quad b_{\text{eff}} = 0.4297 \times 175 = 75.20 \text{ mm} \quad b_{\text{eff}}/2 = 37.60 \text{ mm}
\]

**Determining the location of the centroid of composite section from the bottom steel face using the revised effective widths:**

\[
\bar{y} = \frac{8(9.48 \times 0.42 \times 53.565 + 6.26 \times 0.42 \times 75.645 + 6.65 \times 0.42 \times 78) + 37.60 \times 0.42 \times 50 \times 6}{8(9.48 \times 0.42 + 6.26 \times 0.42 + 6.65 \times 0.42) + 37.6 \times 0.42 \times 6 + 855 \times 0.6} = 14.32 \text{ mm}
\]

**(b) Iteration 2**

**Determining the effective width of element 2** (see Figure 5.8)

\[
\begin{align*}
    f_1^* &= 653.4 \text{ MPa} \\
    f_2^* &= 653.4 \times \frac{35.68}{63.68} = 366.10 \text{ MPa} \\
    \Psi &= \frac{f_2^*}{f_1^*} = \frac{366.10}{653.4} = 0.560 \\
    K_{\text{inc}} &= 4 + 2(1 - \Psi)^3 + 2(1 - \Psi) = 5.05 \quad \lambda = 1.052 \left[ \frac{b}{t} \right] \sqrt{\frac{f_1^*}{E_f K_{\text{inc}}}} = 2.168
\end{align*}
\]

\[
\frac{b_{\text{eff}}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.414 \quad b_{\text{eff}} = 0.414 \times 37.2 = 15.40
\]
\[ b_{\text{eff}1} = \frac{b_{\text{eff}}}{3 - \Psi} = \frac{15.40}{3 - 0.560} = 6.31 \text{ mm} \quad b_{\text{eff}2} = b_{\text{eff}} - b_{\text{eff}1} = 15.40 - 6.31 = 9.09 \text{ mm} \]

Determining the effective width of element 1

\[ R = 275.67 \quad K = 58.87 \quad \lambda = 1.052 \left[ \frac{b}{t} \sqrt{\frac{f_2^*}{E_f K}} \right] = 2.236 \]

\[ \frac{b_{\text{eff}}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.4032 \quad b_{\text{eff}} = 0.4032 \times 175 = 70.56 \text{ mm} \quad b_{\text{eff}} / 2 = 35.28 \text{ mm} \]

Determining the location of the centroid of composite section from the bottom steel face using the revised effective widths:

\[ \bar{y} = \frac{8(9.09 \times 0.42 \times 53.42 + 6.31 \times 0.42 \times 75.625 + 6.65 \times 0.42 \times 78) + 35.28 \times 0.42 \times 50 \times 6}{8(9.09 + 6.31 + 6.65) \times 0.42 + 35.28 \times 0.42 \times 6 + 855 \times 0.6} = 13.94 \text{ mm} \]

\( \bar{y} \) (13.94 mm) from iteration 2 is close to \( \bar{y} \) (14.32 mm) from iteration 1, however, for more accurate results, consider one more iteration.

(c) Iteration 3

Determining the effective width of element 2

\[ f_1^* = 653.4 \text{ MPa} \quad f_2^* = 653.4 \times \frac{36.06}{64.06} = 367.80 \text{ MPa} \quad \Psi = \frac{f_2^*}{f_1^*} = \frac{367.80}{653.4} = 0.563 \]

\[ K_{\text{inc}} = 4 + 2(1 - \Psi)^3 + 2(1 - \Psi) = 5.04 \quad \lambda = 1.052 \left[ \frac{b}{t} \sqrt{\frac{f_1^*}{E_f K_{\text{inc}}}} \right] = 2.17 \]

\[ \frac{b_{\text{eff}}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.4141 \quad b_{\text{eff}} = 0.4141 \times 37.2 = 15.40 \text{ mm} \]

\[ b_{\text{eff}1} = \frac{b_{\text{eff}}}{3 - \Psi} = \frac{15.40}{3 - 0.563} = 6.32 \text{ mm} \quad b_{\text{eff}2} = b_{\text{eff}} - b_{\text{eff}1} = 15.40 - 6.32 = 9.08 \text{ mm} \]
Determining the effective width of element 1

\[ R = 275.67 \quad K = 58.87 \quad \lambda = 1.052 \left[ \frac{b}{t} \right] \sqrt{\frac{f^*}{E_t K}} = 2.241 \]

\[ \frac{b_{\text{eff}}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.4024 \quad b_{\text{eff}} = 0.4024 \times 175 = 70.42 \text{ mm} \quad b_{\text{eff}}/2 = 35.21 \text{ mm} \]

Determining the location of the centroid of composite section from the bottom steel face using the revised effective widths:

\[ y = \frac{8(9.08 \times 0.42 \times 53.415 + 6.32 \times 0.42 \times 75.62 + 6.65 \times 0.42 \times 78)}{8(9.08 + 6.32 + 6.65) \times 0.42 + 35.21 \times 0.42 \times 6 + 855 \times 0.6} = 13.93 \text{ mm} \]

The centroid position \( \bar{y} \) (13.93 mm) from iteration 3 agrees with \( \bar{y} \) (13.94 mm) from iteration 2. These new values of effective widths are therefore adopted as final values.

Determining the effective moment of area of the reduced effective section using the new effective width values:

\[ I_{\text{eff}} = \left[ \frac{9.08^3 \times 0.42}{12} \sin^2 48.81 + 9.08 \times 0.42 \times 39.485^2 \right] \times 8 \]
\[ + \left[ \frac{6.32^3 \times 0.42}{12} \sin^2 48.81 + 6.32 \times 0.42 \times 61.69^2 \right] \times 8 \]
\[ + \left[ \frac{6.65 \times 0.42^3}{12} + 6.65 \times 0.42 \times 64.07^2 \right] \times 8 \]
\[ + \left[ \frac{35.21 \times 0.42^3}{12} + 35.21 \times 0.42 \times 36.07^2 \right] \times 6 + \left[ \frac{855 \times 0.6^3}{12} + 855 \times 0.6 \times 13.93^2 \right] \times 8 \]

\[ I_{\text{eff}} = 435261.87 \text{ mm}^4 \]

Determining the failure pressure \( p_u \)
Case 1: Span \( L = 2200 \) mm

\[
p_u = \frac{8 f_y I_{eff}}{y_{max} BL^2} = \frac{8 \times 653.4 \times 435261.87}{64.06 \times 855 \times 2200^2} = 8.58 \times 10^{-3} \, \text{N/mm}^2 = 8.58 \, \text{kPa}
\]

Case 2: Span \( L = 2800 \) mm

\[
p_u = \frac{8 f_y I_{eff}}{y_{max} BL^2} = \frac{8 \times 653.4 \times 435261.87}{64.06 \times 855 \times 2800^2} = 5.30 \times 10^{-3} \, \text{N/mm}^2 = 5.30 \, \text{kPa}
\]

Case 3: Span, \( l = 3300 \) mm

\[
p_u = \frac{8 f_y I_{eff}}{y_{max} BL^2} = \frac{8 \times 653.4 \times 435261.87}{64.06 \times 855 \times 3300^2} = 3.81 \times 10^{-3} \, \text{N/mm}^2 = 3.81 \, \text{kPa}
\]

(ii) Type B Sandwich Panel

(a) Iteration 1: For the first iteration, consider elements 1 and 2 are fully effective and element 3 is not fully effective (see Figure 5.9).

Determining the effective width of element 3:

Since element 3 is fully supported by foam core, effective width for this element is calculated by using current design rule (CEB 2000).

\[
R = 0.90 \quad K = \left[16 + 7R + 0.02R^2\right]^{1/2} = 4.72 \quad \lambda = 1.052 \left[\frac{b}{t}\right] \sqrt{\frac{f_y}{E_y K}} = 1.567
\]

\[
\frac{b_{eff}}{b} = \frac{1}{\lambda} \left[1 - \frac{0.22}{\lambda}\right] = 0.5485 \quad b_{eff} = 0.5485 \times 26 = 14.26 \, \text{mm} \quad b_{eff}/2 = 7.13 \, \text{mm}
\]

Determining the centroid of composite section from the bottom steel face:

\[
y = \frac{33.3 \times 0.42 \times 64 \times 6 + 7.13 \times 0.42 \times 78 \times 6 + 125 \times 0.42 \times 50 \times 2}{33.3 \times 0.42 \times 6 + 7.13 \times 0.42 \times 6 + 125 \times 0.42 \times 2 + 466 \times 0.4} = 30.57 \, \text{mm}
\]

Determining the effective width of element 2
Since element 2 is also supported by foam core, the effective width of this element is calculated by using CIB (2000) design rule. Element 2 is a stiffened element with a stress gradient. The effective width for such a section is determined in accordance with Clause 2.2.3.2 of AS/NZS 4600 (SA, 1996). The calculations are given next:

\[ f_1^* = 653.4 \text{ MPa} \quad f_2^* = 653.4 \times \frac{19.43}{47.43} = 267.67 \text{ MPa} \quad \Psi = \frac{f_2^*}{f_1^*} = \frac{267.67}{653.4} = 0.410 \]

\[ R = 1.90 \quad K = \left[ 16 + 7R + 0.02R^2 \right]^{1/2} = 5.42 \]

\[ K_{inc} = 5.42 + 2(1- \Psi)^3 + 2(1- \Psi) = 7.01 \quad \lambda = 1.052 \left[ \frac{b}{t} \right] \frac{f_1^*}{E_f K_{inc}} = 1.647 \]

\[ \frac{b_{eff}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.526 \quad b_{eff} = 0.526 \times 33.3 = 17.52 \text{ mm} \]

Since the effective width of inclined steel element is less than the total width, element 2 is not fully effective.

\[ b_{eff1} = \frac{b_{eff}}{3 - \Psi} = \frac{17.52}{3 - 0.410} = 6.76 \text{ mm} \quad b_{eff2} = b_{eff} - b_{eff1} = 17.52 - 6.76 = 10.76 \text{ mm} \]

**Determining the effective width of element 1**

Since element 1 is supported with foam core, its effective width is determined using CIB (2000) design rule as follows:

\[ R = 100.46 \quad K = \left[ 16 + 7R + 0.02R^2 \right]^{1/2} = 30.35 \quad \lambda = 1.052 \left[ \frac{b}{t} \right] \frac{f_2^*}{E_f K} = 1.902 \]

\[ \frac{b_{eff}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.465 \quad b_{eff} = 0.465 \times 125 = 58.12 \text{ mm} \quad b_{eff} / 2 = 29.06 \text{ mm} \]

**Determining the location of the centroid of composite section from the bottom steel face using the revised effective widths:**
(b) Iteration 2

Determining the effective width of element 2 (see Figure 5.10)

\[ f_1^* = 653.4 \text{ MPa} \quad f_2^* = 653.4 \times \frac{27.80}{55.80} = 325.53 \text{ MPa} \quad \Psi = \frac{f_2^*}{f_1^*} = \frac{325.53}{653.4} = 0.498 \]

\[ R = 1.90 \quad K = 5.42 \quad K_{inc} = 5.42 + 2(1 - \Psi)^3 + 2(1 - \Psi') = 6.68 \]

\[ \lambda = 1.052 \left[ \frac{b}{t} \right] \sqrt{\frac{f_1^*}{E_f K_{inc}}} = 1.687 \quad \frac{b_{eff}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.515 \]

\[ b_{eff} = 0.515 \times 33.3 = 17.15 \text{ mm} \]

\[ b_{eff1} = \frac{b_{eff}}{3 - \Psi} = 6.85 \text{ mm} \quad b_{eff2} = b_{eff} - b_{eff1} = 10.30 \text{ mm} \]

Determining the effective width of element 1

\[ R = 100.46 \quad K = 30.35 \quad \lambda = 1.052 \left[ \frac{b}{t} \right] \sqrt{\frac{f_2^*}{E_f K}} = 2.097 \]

\[ \frac{b_{eff}}{b} = \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.427 \quad b_{eff} = 0.427 \times 125 = 53.38 \text{ mm} \quad b_{eff}/2 = 26.69 \text{ mm} \]

Determining the location of the centroid of composite section from the bottom steel face using the revised effective widths:

\[ \bar{y} = \frac{6(10.30 \times 0.42 \times 54.33 + 6.85 \times 0.42 \times 75.12 + 7.13 \times 0.42 \times 78) + 26.69 \times 0.42 \times 50 \times 4}{6(10.30 \times 0.42 + 6.85 \times 0.42 + 7.13 \times 0.42) + 26.69 \times 0.42 \times 4 + 466 \times 0.4} = 21.72 \text{ mm} \]

*y* (21.72 mm) from iteration 2 is close to *y* (22.20 mm) from iteration 1, however, for more accurate results, consider one more iteration.

(c) Iteration 3

Determining the effective width of element 2

\[
\begin{align*}
  f_1^* &= 653.4 \text{ MPa} & f_2^* &= 653.4 \times \frac{28.28}{56.28} = 328.33 \text{ MPa} & \Psi &= \frac{f_2^*}{f_1^*} = \frac{328.33}{653.4} = 0.502 \\
  R &= 1.90 & K &= 5.42 & K_{inc} &= 5.42 + 2(1 - \Psi)^3 + 2(1 - \Psi) = 6.66 \\
  \lambda &= 1.052 \left[ \frac{b}{t} \right] \sqrt{\frac{f_1^*}{E_f K_{inc}}} = 1.690 & \frac{b_{eff}}{b} &= \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.5147 \\
  b_{eff} &= 0.5147 \times 33.3 = 17.14 \text{ mm} \\
  b_{eff_1} &= \frac{b_{eff}}{3 - \Psi} = 6.86 \text{ mm} & b_{eff_2} &= b_{eff} - b_{eff_1} = 10.28 \text{ mm}
\end{align*}
\]

Determining the effective width of element 1

\[
\begin{align*}
  R &= 100.46 & K &= 30.35 & \lambda &= 1.052 \left[ \frac{b}{t} \right] \sqrt{\frac{f_2^*}{E_f K}} = 2.106 \\
  \frac{b_{eff}}{b} &= \frac{1}{\lambda} \left[ 1 - \frac{0.22}{\lambda} \right] = 0.425 & b_{eff} &= 0.425 \times 125 = 53.12 \text{ mm} & b_{eff}/2 &= 26.56 \text{ mm}
\end{align*}
\]

Determining the location of the centroid of composite section from the bottom steel face using the revised effective widths:

\[
\bar{y} = \frac{6(10.28 \times 0.42 \times 54.325 + 6.86 \times 0.42 \times 75.115 + 7.13 \times 0.42 \times 78) + 26.56 \times 0.42 \times 50 \times 4}{6(10.28 + 6.86 + 7.13) \times 0.42 + 26.56 \times 0.42 \times 4 + 466 \times 0.4} = 21.70 \text{ mm}
\]
The centroid position $y$ (21.70 mm) from iteration 3 agrees with $y$ (21.72 mm) from iteration 2. These new values of effective widths are therefore adopted as final values.

**Determining the effective moment of area of the reduced effective section using the new effective width values:**

$$I_{eff} = \left[ \frac{10.28^3 \times 0.42}{12} \sin^2 57.26 + 10.28 \times 0.42 \times 32.625^2 \right] \times 6$$
$$+ \left[ \frac{6.86^3 \times 0.42}{12} \sin^2 57.26 + 6.86 \times 0.42 \times 53.415^2 \right] \times 6$$
$$+ \left[ \frac{7.13 \times 0.42^3}{12} + 7.13 \times 0.42 \times 56.30^2 \right] \times 6$$
$$+ \left[ \frac{26.56 \times 0.42^3}{12} + 26.56 \times 0.42 \times 28.30^2 \right] \times 4 + \left[ \frac{466 \times 0.4^3}{12} + 466 \times 0.4 \times 21.70^2 \right]$$

$$I_{eff} = 257571.61 \text{ mm}^4$$

**Determining the failure pressure $p_u$**

**Case 1:** Span $L = 2200$ mm

$$p_u = \frac{8f_{y}I_{eff}}{y_{max}BL^2} = \frac{8 \times 653.4 \times 257571.61}{56.30 \times 466 \times 2200^2} = 10.60 \times 10^{-3} \text{ N/mm}^2 = 10.60 \text{ kPa}$$

**Case 2:** Span $L = 2550$ mm

$$p_u = \frac{8f_{y}I_{eff}}{y_{max}BL^2} = \frac{8 \times 653.4 \times 257571.61}{56.30 \times 466 \times 2550^2} = 7.90 \times 10^{-3} \text{ N/mm}^2 = 7.90 \text{ kPa}$$

**Case 3:** Span $L = 2800$ mm

$$p_u = \frac{8f_{y}I_{eff}}{y_{max}BL^2} = \frac{8 \times 653.4 \times 257571.61}{56.30 \times 466 \times 2800^2} = 6.55 \times 10^{-3} \text{ N/mm}^2 = 6.55 \text{ kPa}$$
5.6 Test Results and Comparison with the Design Predictions

The experimental failure pressures of all six specimens obtained from the full-scale tests were compared with the corresponding predictions from the current (CIB, 2000) and new improved design rules as calculated in the last section. These results and comparisons are presented in Table 5.3.

Table 5.3 Comparison of Failure Pressures

<table>
<thead>
<tr>
<th>Test No.</th>
<th>Panel Type</th>
<th>Experimental Failure Pressure (kPa)</th>
<th>Predicted Failure Pressure (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>New Design Rule</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Ratio New design/Expt.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>CIB 2000</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Ratio CIB 2000/Expt.</td>
</tr>
<tr>
<td>1</td>
<td>A</td>
<td>7.65</td>
<td>7.49</td>
</tr>
<tr>
<td>2</td>
<td>A</td>
<td>5.17</td>
<td>4.62</td>
</tr>
<tr>
<td>3</td>
<td>A</td>
<td>3.72</td>
<td>3.33</td>
</tr>
<tr>
<td>4</td>
<td>B</td>
<td>9.93</td>
<td>9.21</td>
</tr>
<tr>
<td>5</td>
<td>B</td>
<td>7.05</td>
<td>6.86</td>
</tr>
<tr>
<td>6</td>
<td>B</td>
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<td>5.69</td>
</tr>
<tr>
<td>Mean</td>
<td></td>
<td>0.94</td>
<td>1.08</td>
</tr>
<tr>
<td>Coefficient of Variation (COV)</td>
<td>0.04</td>
<td>0.04</td>
<td></td>
</tr>
</tbody>
</table>

As seen from the table, the results obtained from the new improved design rule are in good agreement with the experimental results for all six tested sandwich panels although they are slightly conservative. On the other hand, the CIB (2000) design rule always overestimates the panel strength in comparison with the experimental results. The mean value of the ratio of the failure pressures predicted by the new design rule and the experiments was found to be 0.94 and the corresponding coefficient of variation (COV) was 0.04. However, the mean value of the ratio of the failure pressures predicted from the current design rule (CIB, 2000) and the experiments was found to be 1.08 and the corresponding coefficient of variation (COV) was to be 0.04. This shows that the mean of the ratio of the failure pressures from CIB (2000) and the experiments is consistently high and is greater than one. This implies that CIB (2000) design rule is not safe to use in design. It must be noted
here that COV of the ratio of the failure pressures from CIB (2000) and the experiments is 0.04. This is due to very low dispersion of the mean values for all six panels, although they are very high.

Since the profiled steel faces used were made of 0.42 mm G550 steels, the design predictions from both CIB (2000) and new design rules are based on 90% reduction in yield stress ($0.9f_y$). It is possible that the assumed reduced value of yield stress might be too conservative to use for sandwich panels, in which case, the predicted failure pressures from both the design rules should be considered as lower bound values. If actual yield stress was used, design predictions based on both design rules would be higher than the values shown in Table 5.3. In this case, the predictions from the new design rule would be very close to experimental results and less conservative. But the predictions based on the CIB (2000) design rule would be even more unsafe compared with the experimental results.

The effective width results calculated in the last section can also be used here to compare the reliability of the two design rules. For example, the effective width of element 3 of Type B sandwich panels (Figure 5.9) obtained from the new and CIB (2000) design rules were 13.52 mm and 14.26 mm, respectively. These predictions from the two design rules are very close as the $b/t$ ratio of plate element 3 is low (61.90). However, the effective width of element 1 of Type B sandwich panels (Figure 5.9) obtained from the new and CIB (2000) design rules were 36.36 mm and 53.12 mm, respectively. These predictions from the two design rules vary considerably as the $b/t$ ratio of plate element 1 is high (297.62). This clearly indicated that the CIB (2000) design rule predicted a higher effective width value for slender plates and this resulted in higher estimates of failure pressure making the design unsafe when compared with experimental results. All these comparisons with the experimental results confirmed that the new design rule predicted satisfactory values of failure pressures and hence can be used in the design of profiled sandwich panels to achieve safe and reliable design solutions.

As mentioned earlier in Section 5.4, Jeevaharan’s (1997) test result of $f_y = 726$ MPa was used in all the calculations because of the difficulty in obtaining the required tensile coupons from the panels. However, a sensitivity study was conducted on 0.42
mm G550 grade steel to investigate the effect of the possible variation in yield stress, ie. from 700 to 800 MPa, on the capacity of the panels. Table 5.4 shows the comparison of panel strengths predicted by the new design rule based on different yield stresses ($f_y = 726$ and 800 MPa). As seen in this table, the differences in the capacity of the panels were minimal with a maximum difference of 5% when compared with the capacity of the panel with $f_y = 726$ MPa. This showed that the variation in yield stress does not have a significant effect in the capacity of the panels. In fact, the use of $f_y = 800$ MPa improves the comparison of the panel capacities predicted by the new design rule with experimental results. The full scale experiments reported in this chapter were not undertaken to develop any design rules, but instead to examine the reliability of the developed design rules in Chapter 4. Small variation in the yield stress of steel faces does not make any significant changes to the panel strength results, and most importantly the final conclusions are not affected by the assumption of $f_y = 726$ MPa in the calculations.

### Table 5.4 Effect of Yield Stress on the Calculated Failure Pressures

<table>
<thead>
<tr>
<th>Test No.</th>
<th>Panel Type</th>
<th>Experimental Failure Pressure (kPa)</th>
<th>Failure Pressure based on New Design Rule ($f_y = 800$ MPa (1))</th>
<th>% Difference between (1) and (2)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>$f_y = 726$ MPa (2)</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>A</td>
<td>7.65</td>
<td>7.96</td>
<td>5.90</td>
</tr>
<tr>
<td>2</td>
<td>A</td>
<td>5.17</td>
<td>4.91</td>
<td>5.91</td>
</tr>
<tr>
<td>3</td>
<td>A</td>
<td>3.72</td>
<td>3.54</td>
<td>5.93</td>
</tr>
<tr>
<td>4</td>
<td>B</td>
<td>9.93</td>
<td>9.67</td>
<td>4.76</td>
</tr>
<tr>
<td>5</td>
<td>B</td>
<td>7.05</td>
<td>7.20</td>
<td>4.72</td>
</tr>
<tr>
<td>6</td>
<td>B</td>
<td>5.84</td>
<td>5.97</td>
<td>4.69</td>
</tr>
</tbody>
</table>

It should be noted that there are several factors that affect the experimental strength of sandwich panels. One such important factor is the proper bonding between steel faces and foam core. In Type B sandwich panels, attachment of foam core with the steel face at the profiled ridge part is very complicated in the Australian method of sandwich panel construction. Without the use of a good manufacturing method and workmanship, it is difficult to ensure proper bonding in that part of the panel. Any
such improper bonding ultimately results in the reduction of panel strength. In the absence of such problems, the predictions from the new design rule are very close to the true strength of the profiled sandwich panels. Hence it is recommended that the new design rule is used in the design of profiled sandwich panels subjected to local buckling effects.

5.7 Summary

A series of full-scale tests on six profiled sandwich panels was conducted under a uniformly distributed wind pressure loading. Experimental failure pressures were compared with predictions from a recently developed design rule and the current design rule included in “European Recommendations for Sandwich Panels, Part 1: Design” (CIB, 2000). The comparisons indicated that the current design rule overestimated the strength of sandwich panels whereas the strength predicted by the new improved design rule agreed well with the experimental results. This confirmed that the new design rule developed based on an extensive series of experimental and numerical studies predicted accurately the true strength of profiled sandwich panels subjected to local buckling effects. Hence, the new design rule is recommended for the design of profiled sandwich panels to achieve safe design solutions.
CHAPTER 6.0 LIGHTLY PROFILED SANDWICH PANELS SUBJECT TO LOCAL BUCKLING AND FLEXURAL WRINKLING

6.1 General

Sandwich panels exhibit various types of buckling behaviour depending upon the types of faces used. Local buckling behaviour is the critical failure mode for fully profiled sandwich panels as their flat plate elements have relatively low width to thickness \((b/t)\) ratios (see Figure 6.1 (a)). This local buckling behaviour of fully profiled sandwich panels was investigated in Chapters 3 to 5. If the width to thickness \((b/t)\) ratio of the flat plate elements is very large, sandwich panels undergo a flexural wrinkling type failure. This type of buckling failure does not include any postbuckling strength, and occurs at a stress well below the yield stress of the steel. Hence flexural wrinkling is an extremely important design criterion for flat and lightly profiled sandwich panels as they are always susceptible to wrinkling failures (see Figure 6.1 (b)).

This figure is not available online. Please consult the hardcopy thesis available from the QUT Library

(a) Local Buckling of Fully Profiled Panels (Davies and Hakmi, 1990)  (b) Flexural Wrinkling of Lightly Profiled Panels (McAndrew, 1999)

Figure 6.1 Local Buckling and Flexural Wrinkling of Sandwich Panels

Flexural wrinkling is a form of local instability of the steel faces of sandwich panels associated with short waves of buckling. When a sandwich panel with flat or lightly profiled faces is subjected to a compression load or one of the steel faces is in compression under bending action, a series of buckling waves is developed in the
compression face of sandwich panels at a relatively low load level. As the amplitude of the waves increases with increasing load, one of the waves forms a wrinkle and fails. Unlike the local buckling failure, wrinkling failure is rather sudden with very little warning before failure.

The flexural wrinkling capacity of flat faced sandwich panels can be increased considerably by replacing the flat face with a lightly profiled face. Lightly profiled sandwich panels are generally considered to be those panels with a rib depth of less than 2 mm. Past research (Kech, 1991; Davies, 1993; Hassinen, 1995) has shown that even with such a small profile depth a significant increase in wrinkling stress can result along with good aesthetic appearance. As sandwich panels are increasingly used in many buildings as roof and wall cladding systems, the wrinkling behaviour of flat and lightly profiled panels has been investigated extensively by many researchers. Past research (Davies et al., 1991; Davies, 1993) has led to a well established analytical solution for the wrinkling of flat faced sandwich panels. This wrinkling formula agrees well with the test and numerical results and hence can be used for the design of flat faced sandwich panels with confidence (McAndrew and Mahendran, 1999). The current wrinkling formula of lightly profiled panels is based on modifying the methods utilized for the flat faces by taking into account the flexural stiffness of the lightly profiled faces.

There are two different types of lightly profiled faces used in Australian sandwich panel construction, namely, satinlined profile (Figure 6.2) and ribbed profile (Figure 6.3). The study conducted by McAndrew (1999) showed that the wrinkling stresses obtained from finite element analysis (FEA) agreed reasonably well with the theoretical predictions for the panels with satinlined profiles if the ridge height is less than 1 mm. However, the same study revealed that the wrinkling stresses from FEA and theoretical predictions did not agree well for the panels with ribbed profiles. McAndrew (1999) found that the current theoretical formula overestimated the wrinkling strength of ribbed profiles compared with FEA results. From the finite element analysis, McAndrew (1999) identified that the buckle shape of satinlined panel was very similar to that of flat panels. This implies that satinlined panels behave very similar to flat panels and hence wrinkling stress prediction from theoretical and FEA results are in good agreement. However, ribbed panels showed
the presence of flat plate buckling between the ribs along with wrinkling of overall panel (see Figure 2.3). This interactive buckling phenomenon caused failure to occur at stresses lower than that predicted by the wrinkling formula. As the current theoretical equation can still be applied for satinlined panels with ridge height less than 1 mm, the wrinkling behaviour of this type of panel is not considered in this study. In contrast, McAndrew (1999) has shown that the current wrinkling formula is inadequate for the lightly profiled panels with ribbed profiles. This research project has therefore focussed its study on panels with ribbed profiles to investigate their mixed mode buckling behaviour and to resolve the problems associated with the current design rules.

Some important structural parameters are defined here to avoid any confusion in the discussion. In the ribbed panel, small profiles as shown in Figure 6.3 are called ribs and the depth of rib is indicated by $h_c$. The clear width between the two ribs is called $b$. The depth of rib is indicated by $h_c$. The clear width between the two ribs is called
the width of the flat plate element or spacing of ribs and indicated by \( b \). As \( b \) and \( b/t \) ratio of flat plate elements are related parameters, the use of one also represents the other when plate thickness is the same and should be understood in the discussion accordingly. The failure stress of ribbed panels is expressed as interactive buckling stress as the failure occurs due to the interaction of wrinkling and local buckling modes.

This chapter discusses the currently available methods to evaluate the wrinkling stress of lightly profiled sandwich panels. A detailed finite element investigation of the lightly profiled panels with varying rib depths and \( b/t \) ratios of flat plates is presented. A series of experiments and currently available theoretical formulae have been used to validate the finite element model. Details of the experimental programs are discussed. A half-length model was used to simulate the experimental lightly profiled panels while a half-wave buckle length model was used to review the current design method. A detailed comparison of finite element analysis results with the predictions based on the current design rules is also presented. A new improved design rule which takes into account the interaction of the two buckling modes was developed for the design of lightly profiled panels based on extensive FEA results.

### 6.2 Current Rules for the Wrinkling Stress of Lightly Profiled Panels

Some mathematical formulations have been proposed by different researchers to determine the wrinkling stress of lightly profiled sandwich panels. Detailed mathematical derivations of these wrinkling formulae are given in Chapter 2. However, a brief overview of important wrinkling equations is given in this section for easy referencing in this chapter. Currently, the following methods are generally used to determine the wrinkling stress of lightly profiled panels.

**a) By modifying the method used for flat panels (Davies et al., 1991)**

The theoretical expression for wrinkling stress \( \sigma_{wr} \) for flat faced sandwich panels was derived based on the elastic half space method (Davies et al., 1991) and is expressed in the following form.
A general form of wrinkling stress formula for flat faced sandwich panels is given by converting Equation 6.2.1 into the following simplified form.

\[ \sigma_{wr} = K (E_f E_c G_c)^{1/3} \]  \hspace{1cm} (6.2.2)

where \( K \) is the numerical constant, \( E_f \) is the elastic modulus of steel face, \( E_c \) is the elastic modulus of foam core, \( G_c \) is the shear modulus of the foam core. \( E_c \) and \( G_c \) are the characteristic values when used for design purposes and are the 5% fractile values of the population (CIB, 2000).

To obtain the theoretical expression for the lightly profiled sandwich panels, Equation 6.2.1 was modified to take into account the bending stiffness of the lightly profiled faces (Davies et al., 1991) as shown next.

\[ \sigma_{wr} = \frac{1.89}{A_f} \left( \frac{8(1-\nu)^2 E_c G_c B_f}{(1+\nu)(3-4\nu^2)} \right)^{1/3} \]  \hspace{1cm} (6.2.3)

where \( A_f \) is the cross-sectional area of the face per unit width, \( B_f \) is the flexural rigidity of the face per unit width \((E_f I_f / b)\).

The energy method results in the following mathematical expression for a half-wave buckle length \( a \) of the buckling mode of lightly profiled faces.

\[ a = \pi \left( \frac{2B_f}{R} \right)^{1/3} \]  \hspace{1cm} (6.2.4)

where \( R = \frac{2(1-\nu)E_c}{(1+\nu)(3-4\nu^2)} \)  \hspace{1cm} (6.2.5)
(b) By using the empirical factor in the wrinkling stress of flat faced sandwich panels (Davies, 1993)

In this method, the wrinkling stress of lightly profiled sandwich panel is determined by applying an empirical factor to the formula (Equation 6.2.2) for a flat faced sandwich panel. The modified equation is given by:

$$\sigma_{wr} = K\alpha\left(E_cG_fB_f\right)^{1/3} \quad (6.2.6)$$

where $\alpha$ is an empirical factor, which is always greater than one ($\alpha > 1$). The value of $\alpha$ is determined by testing. The difficulty with this equation is that $\alpha$ is not a constant, but changes with the cross-section of the compressed face and thickness and properties of the foam core material. For each particular profile, a new value of $\alpha$ should be determined by testing which is an expensive process and not always convenient.

(c) Wrinkling stress formula developed by Kech (1991)

Kech’s (1991) study of lightly profiled sandwich panels focused on the fact that as the depth and distance between the ribs increase, flat plate buckling occurs and the failure occurs due to the interaction of local buckling and flexural wrinkling modes. As the current wrinkling stress formulae do not take into account this mixed mode type of buckling failure, the wrinkling stress predicted by them is always far from the realistic value. Based on his investigation, Kech (1991) developed a new wrinkling stress formula for lightly profiled sandwich panels as an improvement to that developed using an energy method (Equation 6.2.3) to take into account the interaction between the two buckling modes. Kech’s model which treats the folded area, including the effective width of flat elements on either side of it, as an axially compressed column on an elastic foundation results in the following mathematical expression to calculate the wrinkling stress of lightly profiled panels.
The cross-section of the column and the detailed nomenclature of model along with the definitions of all the parameters used in Equation 6.2.7 are given in Section 2.7 of Chapter 2. Although Equation 6.2.7 is an improvement to the previous method, this method is valid only for small ratios of profile depth to the plate thickness and is verified by comparison with a limited number of test results. To prove its reliability for the use in design practice, a more extensive comparison with test results or exact analysis is needed (Davies et al., 1991).

(d) Wrinkling stress formula recommended by CIB (2000)

In the current European design standards “European Recommendation for Sandwich Panels, Part I: Design” (CIB, 2000) the wrinkling formula derived using the energy method has been adopted for design with modification via an empirical factor. This design formula is a slight modification of Equation 6.2.3 and takes the form:

\[
\sigma_{wr} = \frac{b_{df1} + b_{df2} + 2b_s}{b_h} \sigma_k \tag{6.2.7}
\]

where \(K_p\) is the numerical constant and is 0.95 for design purposes as recommended by CIB (2000). This value represents a reduction of approximately 50% in the original wrinkling strength calculated using Equations 6.2.3. Such a big reduction in strength has been made in order to consider some practical limitations such as effect of initial imperfections in the face, finite depth and non-linear behaviour of the foam core, and bond between steel faces and foam core. However, this reduction is not intended to take into account the reduction in wrinkling stress due to the interaction of wrinkling and local buckling modes.

As the depth or spacing of the ribs increases, flat plate buckling can occur leading to the failure of the entire panel due to the interaction between wrinkling and local buckling. This mixed mode type buckling behaviour must be considered in the
design to predict the true value of wrinkling stress of lightly profiled sandwich panels (Kech, 1991; Davies, 1993; Mahendran and McAndrew, 2000). Since the present design document (CIB, 2000) does not address this problem, an extensive study using a series of experiments and finite element analyses is necessary to fully understand and investigate the interactive behaviour of lightly profiled panels in order to develop a new design rule that considers this interactive buckling behaviour.

6.3 Experimental Study to Investigate the Interactive Buckling Behaviour

As there is no reliable design formula for the lightly profiled sandwich panels, CIB (2000) recommends that testing is the most suitable method to determine the wrinkling stress with regard to lightly profiled faces. Therefore in this research project flexural wrinkling of lightly profiled sandwich panels was investigated using an extensive series of experiments.

6.3.1 Test Specimens and Programs

In order to investigate experimentally the wrinkling behaviour of lightly profiled sandwich panels, compression tests of lightly profiled steel plate elements supported by a polystyrene foam core were conducted. All the specimens required for the tests were prepared in the Structural Laboratory at Queensland University of Technology. Flat steel plate elements with the required length and width were first cut longitudinally from cold-formed steel sheets of known grade and thickness. Light profiles of certain depth and width were then introduced into the flat plates using special equipment. The overall length of the panel was three times the width plus 10 mm for clamping. Polystyrene foam cores (SL grade – 13.5 kg/m³) were cut to the required size (width \( b \) and length \( 3b \)) using a hot-wire machine. The lightly profiled steel plates were cleaned and dried. The polystyrene foam core and corresponding lightly profiled steel plates were then glued together using an adhesive (Bostik). Unlike flat plates, it was very difficult to attach lightly profiled plates with the foam core due to the small profiles in the steel plates. To ensure full attachment, large pressure was applied to the top of foam-supported steel plate elements and
maintained for 24 to 48 hours. This allowed the lightly profiled steel plates and foam core to be fully attached to each other without any gaps.

![Diagram](image)

(a) Type A

![Diagram](image)

(b) Type B

**Figure 6.4 Lightly Profiled Steel Faces**

As discussed earlier, the depth and spacing of the ribs have a greater effect on the flexural wrinkling strength of lightly profiled sandwich panels. To study this behaviour experimentally, test specimens with different spacings between the ribs were prepared and tested. However, only one type of rib depth was tested due to the difficulties involved in preparing them in the laboratory. The rib depth of the foam-supported steel plate elements used was 1.0 mm. The widths of flat plates between
the ribs (rib spacing) chosen were 28.5 and 78.5 mm (see Figure 6.4). To vary the \( b/t \) ratio of the flat plates, steel plates with different thicknesses were used. For this experimental investigation, steel thicknesses used were 0.42, 0.60 and 0.95 mm. The G550 steel grade was used in the experimental study since it is commonly used in Australian sandwich panels. With the different spacings of the ribs and thicknesses of steel, the \( b/t \) ratio of the flat plate was varied from 47 to 187. A constant foam thickness of 100 mm was used in the experiments. Two different profiles (Type A and Type B) were tested in this study. Type A profile consisted of 4 ribs with the width of flat plates equal to 78.5 mm. Similarly, Type B profile consisted of 8 ribs with the width of flat plates equal to 28.5 mm. Overall width of all the panels tested was 400 mm and length 1200 mm. The schematic diagrams of Type A and Type B lightly profiled faces are shown in Figure 6.4.

Table 6.1 shows the details of the test program and the test specimens. Although the overall width of the specimens was 400 mm, the actual width of each specimen was slightly more than 400 mm due to the ribs in the steel plate. The actual width of the specimens along with the width of the flat plates, the thickness of the steel plate, the \( b/t \) ratio of the flat plates for both Type A and Type B sandwich panels and the number of ribs are shown in Table 6.1. A total of five tests was conducted to investigate the wrinkling behaviour of lightly profiled sandwich panels. Some of the test specimens are shown in Figure 6.5.

### Table 6.1 Details of Test Specimens and Test Program

<table>
<thead>
<tr>
<th>Test No.</th>
<th>Specimen Type</th>
<th>Number of Ribs</th>
<th>Actual Overall width (mm)</th>
<th>Width of Flat Plate ( b ) (mm)</th>
<th>Thickness ( t ) (mm)</th>
<th>( b/t ) Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>A</td>
<td>4</td>
<td>400.93</td>
<td>78.5</td>
<td>0.95</td>
<td>82.6</td>
</tr>
<tr>
<td>2</td>
<td>A</td>
<td>4</td>
<td>400.93</td>
<td>78.5</td>
<td>0.60</td>
<td>130.8</td>
</tr>
<tr>
<td>3</td>
<td>A</td>
<td>4</td>
<td>400.93</td>
<td>78.5</td>
<td>0.42</td>
<td>186.9</td>
</tr>
<tr>
<td>4</td>
<td>B</td>
<td>8</td>
<td>401.86</td>
<td>28.5</td>
<td>0.60</td>
<td>47.5</td>
</tr>
<tr>
<td>5</td>
<td>B</td>
<td>8</td>
<td>401.86</td>
<td>28.5</td>
<td>0.42</td>
<td>67.8</td>
</tr>
</tbody>
</table>
6.3.2 Test Set-Up and Procedure

The compression tests to investigate the interactive buckling behaviour of lightly profiled sandwich panels were carried out using a Tinius Olsen Testing Machine. A large test rig with identical characteristics to the smaller one used in Chapter 3 was used in this study to hold the test specimens. However, because of the limitation with the Tinius Olsen Testing Machine, the height and width of the large test rig were limited to 1300 mm and 400 mm, respectively.

Test specimens were placed in the test rig between the two loading blocks. Since a constant width of 400 mm was used for all the lightly profiled plates in this test, top and bottom loading blocks were made 400 mm wide in order to satisfy the plate width. The test rig along with the test specimens was then positioned in the Tinius Olsen Testing Machine. The axial compression load was applied to the lightly profiled plate via the top loading block. Axial and out-of-plane deflections were measured using linear variable displacement transducers (LVDTs). The ultimate load of the specimen before failure was recorded. The complete arrangement of the test
set-up is shown in Figure 6.6. The test procedures and arrangements described here are similar to the compression tests in Chapter 3.

![Test Set-Up for Lightly Profiled Test Specimens](image)

**Figure 6.6 Test Set-Up for Lightly Profiled Test Specimens**

### 6.3.3 Test Observations and Results

During the tests, it was observed that all the test specimens failed in a similar manner. With the continuous application of compression load, small buckles were seen in the flat plates between the ribs. These local buckles were clearly seen in Type A panels compared with Type B panels. When the panel reached the ultimate capacity, the applied load decreased very rapidly without any warning. This failure pattern showed that the panels failed due to wrinkling of the panel with slight local buckling in the flat plates between the ribs. Specimen 1 showed a slightly different failure pattern although local buckling was still seen in the flat plate as in the other panels. After the applied load reached 26 kN, it decreased at first, but increased again until the specimen failed. Finally, the specimen failed suddenly after reaching the ultimate load of 30.41 kN. Figure 6.7 shows the typical failure mode of lightly profiled panels tested in this study. Figure 6.8 shows the typical plot of compressive load versus axial displacement curve.
Figure 6.7 Typical Failure Mode

Figure 6.8 Typical Load versus Axial Displacement Curve
Table 6.2 Experimental Ultimate Loads and Stresses

<table>
<thead>
<tr>
<th>Test No.</th>
<th>Panel Type</th>
<th>Rib Number</th>
<th>Width of Flat Plate $b$ (mm)</th>
<th>Thickness $t$ (mm)</th>
<th>$b/t$ Ratio</th>
<th>Ultimate Load (kN)</th>
<th>Ultimate Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>A</td>
<td>4</td>
<td>78.5</td>
<td>0.95</td>
<td>82.6</td>
<td>30.41</td>
<td>79.84</td>
</tr>
<tr>
<td>2</td>
<td>A</td>
<td>4</td>
<td>78.5</td>
<td>0.60</td>
<td>130.8</td>
<td>19.09</td>
<td>79.36</td>
</tr>
<tr>
<td>3</td>
<td>A</td>
<td>4</td>
<td>78.5</td>
<td>0.42</td>
<td>186.9</td>
<td>11.53</td>
<td>68.47</td>
</tr>
<tr>
<td>4</td>
<td>B</td>
<td>8</td>
<td>28.5</td>
<td>0.60</td>
<td>47.5</td>
<td>21.48</td>
<td>89.09</td>
</tr>
<tr>
<td>5</td>
<td>B</td>
<td>8</td>
<td>28.5</td>
<td>0.42</td>
<td>67.8</td>
<td>14.48</td>
<td>85.79</td>
</tr>
</tbody>
</table>

As all the test specimens failed due to the interaction of the two buckling modes, the ultimate loads and stresses of the specimens can be termed as interactive buckling loads and stresses, respectively. The test results of ultimate loads and stresses are shown in Table 6.2. From the results, it can be observed that the width of the flat plate has a considerable effect on the ultimate capacity of lightly profiled panels. For instance, Test Specimens 2 and 4 were prepared from 0.60 mm thick G550 grade steel. The overall width and length of both the specimens were 400 and 1200 mm, respectively. The width of flat plate between the ribs in Specimen 2 was 78.5 mm whereas in Specimen 4 it was 28.5 mm. The ultimate stress of Specimen 4 was found to be 89.09 MPa whereas the ultimate stress of Specimen 2 was found to be 79.36 MPa. It indicated that interactive buckling stress decreases with the increasing rib spacing.

A similar outcome can be observed by comparing the results of Specimens 3 and 5. They were also prepared from the same grade (G550) and thickness (0.42 mm) of steel. However, the failure stress of Specimen 5 (85.79 MPa) with a smaller spacing (28.5 mm) of ribs was higher than that of Specimen 3 (68.47 MPa) with a rib spacing of 78.5 mm. The test results presented in Table 6.2 were used to calibrate the finite element models developed to simulate the experimental lightly profiled panels later in this chapter.
6.4 Finite Element Analysis of Lightly Profiled Panels

6.4.1 Background

This section presents the description of the finite element study conducted to investigate the flexural wrinkling of lightly profiled sandwich panels interacting with local buckling behaviour. In this study, MSC/PATRAN was used for model generation (pre-processor) and visualization of the results (post-processor) as in the case of the local buckling investigation of fully profiled sandwich panels (Chapter 4). For the numerical computation, a finite element code ABAQUS was used.

In order to simulate the theoretical approach of determining wrinkling stress using the elastic half-space method, the depth and width of the foam core were made sufficiently large in developing the finite element model. The finite element model is based on the application of a compressive load at one end of the steel face with all four sides simply supported as assumed in the energy method used to derive the theoretical wrinkling stress equation. This type of model with all four sides simply supported allows for the simulation of the interactive buckling failure of lightly profiled panels. If the two longitudinal edges parallel to the loading direction are free, global buckling type failure will occur instead of interactive buckling failure.

As in Chapter 4, two different types of finite element models were developed and analysed. The first model was the half-length model used to simulate the experimental lightly profiled panels. The second model was the half-wave buckle length model calibrated using the available theoretical equations, which was then used to review the current design rule and to develop new design rules for lightly profiled sandwich panels. Detailed description of these two types of finite element models are presented in the relevant sections of this chapter.

6.4.2 Elements and Analysis Method

Considering the differences in their mechanical properties, strength behaviour and geometry, steel faces and foam core were modelled using different types of elements.
Shell elements were used to model the lightly profiled steel faces. S4R5 three dimensional (3D) thin shell elements (Quad4) with four nodes and five degrees of freedom per node (three displacement components and two in surface rotation components) were the type of shell element chosen to model the steel faces. These shell elements are small strain elements and are particularly applicable for investigating various buckling modes of sandwich panels. Solid (continuum) elements were used to model the polystyrene foam core as its depth is very large when compared with the thickness of the steel faces. C3D8 three dimensional (3D) solid elements (Hex8) with eight nodes and three degrees of freedom per node (no rotational degrees of freedom) were the types of solid elements chosen to model the foam core.

Since this study was not aimed at investigating the strength of the bond between foam and steel faces, it was assumed that the adhesive agent was sufficiently strong to prevent any delamination between the foam and steel faces. Therefore the steel faces and foam core were modelled as a single unit. This greatly simplified the model by having to avoid the modelling of any contact between the foam and the steel face.

The interactive buckling behaviour of lightly profiled sandwich panels was investigated using both elastic buckling and non-linear analyses. As it is obvious that interactive buckling of the panels occurs in the elastic region, i.e. well below the yield stress of the steel faces, it is considered that an elastic buckling analysis alone is sufficient to study the interactive buckling behaviour. However, a non-linear analysis was also conducted to improve the understanding of interactive buckling behaviour of lightly profiled sandwich panel. In the non-linear analysis, the first buckling mode obtained from the elastic buckling analysis was used as the imperfection shape where the magnitude of imperfection was 0.1\(t\) for the same reasons described in Section 4.2.6.

### 6.4.3 Mechanical Properties

As in Chapter 4, an isotropic material model was used for both steel faces and polystyrene foam core for all the analyses in this study. Previous studies (Jeevaharan,
1997 and McAndrew 1999) have also indicated that the differences in mechanical properties in the longitudinal and transverse directions are not significant. For buckling analysis, a linear elastic material model was used. For the non-linear analysis, the ABAQUS classical plasticity model with elastic perfectly plastic bi-linear stress-strain characteristics was used.

The polystyrene foam considered in this study was a SL grade type, so the mechanical properties of the SL grade foam were used in the finite element analysis. These properties were Young’s modulus of polystyrene foam core $E_c = 3.80$ MPa, shear modulus $G_c = 1.76$ MPa and Poisson’s ratio $\nu_c = 0.08$. The mechanical properties of steel such as Young’s modulus and yield stress as given Table 3.4 of Chapter 3 were used in the analysis. These properties were taken from the experiments conducted by Jeevaharan (1997) as the steel plates used in this investigation were from the same batch of steel. Poisson’s ratio of $\nu = 0.3$ for steel was used.

6.4.4 Loading and Boundary Conditions

In order to simulate the behaviour of the real sandwich panel accurately, finite element models should be assigned with suitable boundary conditions. If the boundary conditions of the model are inappropriate, it does not represent the real conditions and any analysis results from such inaccurate models will lead to wrong solutions. Therefore, finite element models of lightly profiled sandwich panels were analysed using different possible boundary conditions to determine the accurate boundary conditions.

While assigning the boundary conditions in the model, various notations were used to define the constraint conditions. Numbers 1, 2 and 3 were used to represent X-axis, Y-axis, and Z-axis translations, respectively. Similarly, numbers 4, 5 and 6 were used to represent X-axis, Y-axis, and Z-axis rotations, respectively. All these symbols were the standard notations used in the finite element programs ABAQUS and MSC PATRAN. Types of boundary conditions used in the half-length and half-wave buckle length models are described in Sections 6.5 and 6.6, respectively.
The outcome of the results from the numerical model depends also on the type of loading conditions used in the analysis. In the theoretical derivation based on the half-space method, the load was applied to one end of the plate supported on an indefinitely deep elastic foundation. To simulate this approach closely in the finite element models, the load was applied to one end of the steel face of a lightly profiled panel. No load was applied to the foam core as it represents the indefinitely deep elastic foundation. Since the profiled steel face was discretised as a smaller finite element mesh, the load was distributed at individual nodes. The total load to be applied was divided by the number of nodes and applied equally to each individual node. However, two end nodes were applied with only half of the load applied to the middle nodes. For example, if each individual node carries 1N load, then the two end nodes carry only 0.5N load. Figure 6.9 shows the load application method for the finite element models.

![Figure 6.9 Loading Method used in Finite Element Models](image)

6.5 Half-Length Model to Simulate the Behaviour of Experimental Panels

In Chapter 4 it was shown that the behaviour of sandwich panels tested in the laboratory can be simulated well by using a half-length model. All the lightly profiled panels considered in the experimental study were therefore investigated numerically using half-length models. A constant foam thickness of 100 mm was used to simulate the experimental conditions.
6.5.1 Model Geometry, Mesh Sizes and Boundary Conditions

The width of each half-length model was $b/2$ (half the panel width), length $3b/2$ (half the length of the specimen), and thickness equal to the sum of the foam and steel face thicknesses. Appropriate symmetric boundary conditions were applied along the length and width directions to reduce the geometry of the models. The full size experimental panel was thus reduced to a quarter size finite element model. Simply supported boundary conditions were applied along one of the longitudinal edges.

A convergence study was conducted with gradually increasing or decreasing mesh size to determine the appropriate mesh density. Based on this study, mesh sizes of $10 \times 10$ mm for steel face and $10 \times 10 \times 5$ mm for foam core were used for all the half-length models in the analyses.

Figure 6.10 shows the complete geometry of the half-length model along with appropriate boundary conditions and mesh size. This half-length model is similar to that used in Chapter 4 for foam-supported flat plate elements.

![Figure 6.10 Half-Length Model for Lightly Profiled Panels](image-url)
6.5.2 Validation of Half-Length Model using Experimental Results

The half-length model for lightly profiled panels was first analysed using an elastic buckling analysis followed by a non-linear analysis. For all non-linear analyses, the first buckling mode obtained from the elastic buckling analysis was used as the geometric imperfection shape and 10% of the plate thickness \(0.1t\) was used as the maximum imperfection magnitude. The buckling stress corresponding to the first eigen mode was obtained from the elastic buckling analysis whereas the ultimate stress carried by foam-supported lightly profiled steel faces was obtained from the non-linear analysis.

**Figure 6.11 Typical Interactive Buckling Mode of Half-Length Model**

Figure 6.11 shows the typical buckling mode obtained from the finite element analysis of one of the tested panels. From the figure it can be observed that the buckling mode is neither pure wrinkling nor local buckling. Occurrence of flat plate buckling between the ribs can be seen clearly along with the wrinkling of the entire panel. It confirms that lightly profiled panels are subjected to interactive buckling modes and the failure strength is dominated by this interactive buckling behaviour.

The ultimate stresses obtained from the half-length model were compared with the experimental results for all five test specimens as shown in Table 6.3. This table also includes the buckling stress obtained from the FEA based on elastic buckling.
analysis and the wrinkling stress obtained from the theoretical equation (Equation 6.2.3). As seen in Table 6.3, the ultimate stress results obtained from the half-length model agreed reasonably well with those from the experiments. The mean value of the ratio of FEA and experimental ultimate stresses was found to be 0.99. The corresponding coefficient of variation (COV) was 0.09. This comparison confirmed that the half-length model can be successfully used to simulate the interactive local buckling and wrinkling behaviour of lightly profiled sandwich panels.

On the other hand, the theoretical wrinkling stresses are very high when compared with the experimental and FEA ultimate stresses. As the current theoretical wrinkling formula does not take into account the interactive buckling mode as observed in Figure 6.11, its predictions of wrinkling stress are unrealistic.

<table>
<thead>
<tr>
<th>Test No.</th>
<th>Panel Type</th>
<th>Width of Flat Plate $b$ (mm)</th>
<th>Thickness $t$ (mm)</th>
<th>$b/t$ Ratio</th>
<th>Buckling Stress (MPa)</th>
<th>Ultimate Stress (MPa)</th>
<th>Wrinkling Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>FEA</td>
<td>Expt. FEA</td>
<td>Theory</td>
</tr>
<tr>
<td>1</td>
<td>A</td>
<td>78.5</td>
<td>0.95</td>
<td>82.6</td>
<td>71.68</td>
<td>79.84</td>
<td>75.61</td>
</tr>
<tr>
<td>2</td>
<td>A</td>
<td>78.5</td>
<td>0.60</td>
<td>130.8</td>
<td>73.79</td>
<td>73.16</td>
<td>156.66</td>
</tr>
<tr>
<td>3</td>
<td>A</td>
<td>78.5</td>
<td>0.42</td>
<td>186.9</td>
<td>75.60</td>
<td>68.47</td>
<td>192.60</td>
</tr>
<tr>
<td>4</td>
<td>B</td>
<td>28.5</td>
<td>0.60</td>
<td>47.5</td>
<td>85.02</td>
<td>83.62</td>
<td>175.90</td>
</tr>
<tr>
<td>5</td>
<td>B</td>
<td>28.5</td>
<td>0.42</td>
<td>67.8</td>
<td>92.43</td>
<td>85.79</td>
<td>218.80</td>
</tr>
</tbody>
</table>

Overall comparison of FEA and experimental ultimate stresses was found to be reasonably good, however, it must be noted here that the comparison of FEA and experimental results for 0.42 mm thick G550 steel was found to be less satisfactory. G550 grade steel with reduced thickness does not satisfy the required ductility criteria. From the experimental and finite element analysis results in Chapter 4, it was found that finite element models could not simulate this characteristic of G550 steel. It treated G550 grade steel as a normal steel with high yield stress and thus overestimated the strengths. Therefore in this investigation also, the experimental stress results for 0.42 mm thick steel were slightly less than the FEA results.
It is important to note here that wrinkling of sandwich panels always occurs in the elastic region. Since lightly profiled sandwich panels are susceptible to wrinkling failure interacting with flat plate buckling, the interactive buckling stress will be less than the wrinkling stress. Therefore the interactive buckling failure of lightly profiled panels also occurs well below the yield stress of the plate. It can be observed from Table 6.3 that the ultimate stresses obtained from non-linear analyses were very close to the buckling stresses obtained from elastic buckling analyses. These results indicate that interactive buckling occurs in the elastic region and there is limited postbuckling strength.

Further, the buckling stresses obtained from elastic buckling analyses were compared with the experimental ultimate stress as given in Table 6.3. It can be observed from the results that the FEA buckling stress and experimental ultimate stress results are in good agreement. The mean value of the ratio of FEA buckling stress and experimental ultimate stress was found to be 0.99 and the corresponding coefficient of variation (COV) was 0.09. This comparison also confirms that interactive buckling always occurs in the elastic region. The buckling stress results obtained from elastic buckling analysis are the failure stresses (interactive buckling stresses) of lightly profiled sandwich panels. Therefore elastic buckling analysis was adequate to investigate the interactive buckling behaviour and to determine the interactive buckling stress of the lightly profiled sandwich panels. In this study, further analysis into the interactive buckling behaviour of lightly profiled sandwich panels was conducted using elastic buckling analysis.

### 6.6 Half-Wave Buckle Length Model

The validation of the half-length model by using the experimental results improved the confidence that the lightly profiled sandwich panels can be modelled using finite element analyses with an acceptable degree of accuracy. However, the experimental panels do not represent the lightly profiled panels used in sandwich construction as described in Chapter 4. Therefore, to study the interactive buckling behaviour of realistic lightly profiled sandwich panels, the half-wave buckle length model was
used as this model matches with the theoretical model based on the energy method and the realistic sandwich panel conditions. In this smaller model, a finer finite element mesh can be used that will ultimately increase the level of accuracy of numerical results.

6.6.1 Model Geometry, Mesh Sizes and Boundary Conditions

The length of the half-wave buckle length model used in the analysis was \(a/2\) (half of half-wave buckle length), the width was \(b/2\) (half the width of the panel) and the thickness was \(t_c + t_f\) (sum of foam and steel thicknesses). A convergence study was conducted to determine the appropriate mesh size for accurate finite element analyses. A mesh with 5 mm square surface elements for steel faces and 5×5×5 mm solid elements for the foam core was found to be appropriate in terms of accuracy. Similar mesh size was obtained from the convergence study conducted in Chapter 4 for foam-supported flat plate elements. Hence, for the present study of lightly profiled panels, the steel faces were discretised using 5 mm square surface elements whereas the foam core was discretised using 5×5×5 mm solid (continuum) elements.

Appropriate symmetric boundary conditions were applied to enable the half panel width \((b/2)\) and half of half-wave buckle length \((a/2)\) to be modelled. The boundary condition 356 was applied at the loading end. Boundary condition 356 represents that the entire surface can translate in the X and Y directions and rotates about the X-axis, however, it can not translate in the Z-direction and rotate about either the Y or Z axes. A symmetric boundary condition 246 was applied at the other end across the width at a length of half of the half-wave buckle length \((a/2)\) which is the position of maximum amplitude of half-wave. Boundary condition 346, which models the simply supported boundary condition, was applied along one of the longitudinal edges of sandwich panels. A symmetric boundary condition 156 was applied along the other longitudinal edge of the panel at the centre of the panel width. Hence, with two symmetric boundary conditions along the length and across the width, the half-wave buckle length model actually reduces to quarter size. Figure 6.12 shows the geometry, mesh size and appropriate boundary conditions of the half-wave buckle FEA model.
In modelling the geometry, the width \( b \) of the panel is a very important parameter for wrinkling or interactive buckling failures. Wrinkling theory based on the half-space method assumes the width of the panel to be infinite. This implies that the width of the panel in finite element modelling should be wide enough for the panel to fail by wrinkling or interactive buckling. Mahendran and McAndrew (2000, 2001) investigated the effect of the panel width in wrinkling using an extensive series of finite element analyses. From the parametric study, they confirmed that a width of 600 mm or more satisfies the requirement of infinite width in terms of accuracy with regard to wrinkling stress. Hence, a width \( b \) of 600 mm (\( b/2 = 300 \) mm) was used in all the half-wave buckle length models considered in this study for lightly profiled sandwich panels. From the same study, Mahendran and McAndrew (2000, 2001) found that a foam depth of 75 mm or more satisfies the infinite depth considered in the half-space method. This study therefore used a constant foam depth of 75 mm instead of 100 mm to save computational time.

### 6.6.2 Half-Wave Buckle Length

A half-wave buckle model is an ideal model that can be used to investigate the interactive buckling behaviour of lightly profiled sandwich panels. However, the main difficulty associated with this model is to determine the half-wave buckle...
length $a$ needed to create the geometry of the model. The half-wave buckle length $a$ is the length of the panel that gives the minimum eigen value and thus minimum buckling stress. This can be determined by conducting a series of elastic buckling analyses. The optimum single half-wave buckle length $a$ was found by varying $a$ until the minimum eigen value was obtained. A typical example of minimization process conducted to determine the half-wave buckle length $a$ is given in Table 6.4. In this example, a lightly profiled panel with a rib depth of 1.3 mm and the width of flat plate between ribs of 53.5 mm were considered. The grade and thickness of steel considered were G550 and 0.6 mm, respectively. Also, the mesh sizes of 5×5 mm for surface elements and 5×5×5 mm for solid elements were used.

<table>
<thead>
<tr>
<th>$a/2$ (mm)</th>
<th>Surface Element (mm)</th>
<th>Solid Element (mm)</th>
<th>Eigen Value</th>
<th>$\sigma_{wr}$ (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>35</td>
<td>5×5</td>
<td>5×5×5</td>
<td>471.52</td>
<td>156.38</td>
</tr>
<tr>
<td>30</td>
<td>5×5</td>
<td>5×5×5</td>
<td>461.04</td>
<td>152.91</td>
</tr>
<tr>
<td>27</td>
<td>5×5</td>
<td>5×5×5</td>
<td>457.83</td>
<td>151.84</td>
</tr>
<tr>
<td>26</td>
<td>5×5</td>
<td>5×5×5</td>
<td>457.59</td>
<td>151.76</td>
</tr>
<tr>
<td>25</td>
<td>5×5</td>
<td>5×5×5</td>
<td>457.89</td>
<td>151.86</td>
</tr>
<tr>
<td>24</td>
<td>5×5</td>
<td>5×5×5</td>
<td>458.85</td>
<td>152.18</td>
</tr>
<tr>
<td>20</td>
<td>5×5</td>
<td>5×5×5</td>
<td>473.24</td>
<td>156.95</td>
</tr>
<tr>
<td>15</td>
<td>5×5</td>
<td>5×5×5</td>
<td>545.45</td>
<td>180.90</td>
</tr>
</tbody>
</table>

Table 6.4 results show that the buckling stress is very sensitive to the assumed value of the half-wave buckle length $a$. When the assumed value $a/2$ is very close to the true value, the buckling stress corresponding to this length is always the minimum as seen in Table 6.4 and Figure 6.13. This assumed length corresponding to the minimum buckling stress was taken as the half-wave buckle length $a$ of that panel. For the lightly profiled sandwich panel considered in this example, the half of half-wave buckle length $a/2$ was found to be 26 mm from this minimization process. This process was repeated for every lightly profiled sandwich panel considered in this study to determine the appropriate value of half-wave buckle length.
Figure 6.13 Determination of Half-Wave Buckle Length, $a/2$

Figure 6.14 Typical Buckling Mode of Half-Wave Buckle Length Model

Figure 6.14 shows the typical buckling mode of the half-wave buckle length model. From the figure, it is obvious that the failure behaviour of lightly profiled sandwich panels is dominated by the interaction of the two buckling modes, namely, flexural wrinkling and flat plate buckling as in the case of half-length model. In this interactive buckling phenomenon, the wrinkling stress of the panel is affected by
local buckling effects and hence the failure stress can be termed as interactive buckling stress. Therefore the buckling stress in Figure 6.14 is essentially an interactive buckling stress.

6.6.3 Validation of Half-Wave Buckle Length Model

The wrinkling stress and half-wave buckle length \( a \) of flat panels can be calculated accurately by using the theoretical equation based on the elastic half-space method. Therefore the wrinkling stress and half-wave buckle length of flat panels predicted by the half-wave buckle length model can be compared with the theoretical results. However, a theoretical equation is not available to calculate the interactive buckling stress of lightly profiled sandwich panels and hence interactive buckling stress obtained from the half-wave buckle length model can not be compared directly with the theoretical results. Under these circumstances, the half-wave buckle length model of flat panels validated using the theoretical wrinkling equation can be extended to the lightly profiled sandwich panels with identical loading and boundary conditions. This is an indirect method of validating the half-wave buckle length model of lightly profiled sandwich panels.

For flat panels with 0.6 mm G550 steel face, the critical wrinkling stress from the theoretical wrinkling formula (Equation 6.2.1) was found to be 94.64 MPa. Using the half-wave buckle length model with half width \( b/2 = 300 \) mm and half of the half-wave buckle length \( a/2 = 25 \) mm, the critical wrinkling stress was found to be 95.20 MPa for the same grade and thickness of steel. This FEA result is in close agreement with the theoretical result based on the energy method. Therefore this FEA model can be used to simulate the wrinkling behaviour of flat faced sandwich panels. The lightly profiled panels are the same as flat panels except that the flat faces in the latter model are replaced by lightly profiled faces. Loading and boundary conditions are identical. Therefore the validation of half-wave buckle length model for flat panels using the theoretical result confirmed that this model can be extended to the lightly profiled sandwich panels with identical loading and boundary conditions. Hence, the half-wave buckle length model was used in the rest of the chapter to study the interactive buckling behaviour of lightly profiled sandwich panels.
6.7 Parametric Study to Investigate Interactive Buckling Behaviour

To develop a full understanding of the interactive buckling behaviour, lightly profiled sandwich panels with varying rib depths and flat plate widths (rib spacing) were investigated in detail in this study using finite element analysis. Details of the geometry variations considered in this parametric study are shown in Table 6.5. This table includes various rib depths \( h_c \), widths of the flat plate between the ribs \( b \), steel plate thickness \( t \) and the \( b/t \) ratio of flat plate.

Table 6.5 Details of Geometry Variations Considered in the Parametric Study

<table>
<thead>
<tr>
<th>Series Number</th>
<th>Rib Depth ( h_c ) (mm)</th>
<th>Width of Flat Plate ( b ) (mm)</th>
<th>Thickness ( t ) (mm)</th>
<th>( b/t ) Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>1a</td>
<td>0.7</td>
<td>16.0</td>
<td>0.6</td>
<td>26.67</td>
</tr>
<tr>
<td>1b</td>
<td>0.7</td>
<td>28.5</td>
<td>0.6</td>
<td>47.50</td>
</tr>
<tr>
<td>1c</td>
<td>0.7</td>
<td>38.5</td>
<td>0.6</td>
<td>64.17</td>
</tr>
<tr>
<td>1d</td>
<td>0.7</td>
<td>53.5</td>
<td>0.6</td>
<td>89.17</td>
</tr>
<tr>
<td>1e</td>
<td>0.7</td>
<td>78.5</td>
<td>0.6</td>
<td>130.83</td>
</tr>
<tr>
<td>1f</td>
<td>0.7</td>
<td>128.5</td>
<td>0.6</td>
<td>214.17</td>
</tr>
<tr>
<td>2a</td>
<td>1.0</td>
<td>16.0</td>
<td>0.6</td>
<td>26.67</td>
</tr>
<tr>
<td>2b</td>
<td>1.0</td>
<td>28.5</td>
<td>0.6</td>
<td>47.50</td>
</tr>
<tr>
<td>2c</td>
<td>1.0</td>
<td>38.5</td>
<td>0.6</td>
<td>64.17</td>
</tr>
<tr>
<td>2d</td>
<td>1.0</td>
<td>53.5</td>
<td>0.6</td>
<td>89.17</td>
</tr>
<tr>
<td>2e</td>
<td>1.0</td>
<td>78.5</td>
<td>0.6</td>
<td>130.83</td>
</tr>
<tr>
<td>2f</td>
<td>1.0</td>
<td>128.5</td>
<td>0.6</td>
<td>214.17</td>
</tr>
<tr>
<td>3a</td>
<td>1.3</td>
<td>16.0</td>
<td>0.6</td>
<td>26.67</td>
</tr>
<tr>
<td>3b</td>
<td>1.3</td>
<td>28.5</td>
<td>0.6</td>
<td>47.50</td>
</tr>
<tr>
<td>3c</td>
<td>1.3</td>
<td>38.5</td>
<td>0.6</td>
<td>64.17</td>
</tr>
<tr>
<td>3d</td>
<td>1.3</td>
<td>53.5</td>
<td>0.6</td>
<td>89.17</td>
</tr>
<tr>
<td>3e</td>
<td>1.3</td>
<td>78.5</td>
<td>0.6</td>
<td>130.83</td>
</tr>
<tr>
<td>3f</td>
<td>1.3</td>
<td>128.5</td>
<td>0.6</td>
<td>214.17</td>
</tr>
</tbody>
</table>
Lightly profiled faces are generally considered to be those with a rib depth less than 2 mm. Hence, three different practical rib depths less than 2 mm were selected. They were 0.7, 1.0, and 1.3 mm. For each rib depth, six different widths of flat plate between the ribs were considered. Flat plate widths chosen were 16, 28.5, 38.5, 53.5, 78.5 and 128.5 mm. As seen in Table 6.5, the wide range of flat plate widths ( spacings of ribs) and various rib depths were selected to enable the investigation of interactive buckling behaviour of lightly profiled sandwich panels. It covered \( b/t \) ratios of flat plates from 26.67 to 214.17. This range was selected because the lightly profiled sandwich panels used in the buildings and many other structural systems fall within this range.

### 6.7.1 Comparison of Half-Wave Buckle Length with Theoretical Predictions

The results of half of the half-wave buckle length \((a/2)\) obtained from the half-wave buckle length models for all the panels considered in this investigation are given in Tables 6.6, 6.7 and 6.8 for the steel faces with rib depths of 0.7, 1.0 and 1.3 mm, respectively. These tables also present the \(a/2\) values obtained from the theoretical predictions given by Equation 6.2.4 based on the elastic half-space method. A detailed example calculation for the half-wave buckle length using this equation is given later in this chapter (Section 6.8).

#### Table 6.6 Comparison of Half-Wave Buckle Lengths from FEA and Theory for Lightly Profiled Panels with Rib Depth of 0.7 mm

<table>
<thead>
<tr>
<th>Series Number</th>
<th>Width of Flat Plate (b) (mm)</th>
<th>Thickness (t) (mm)</th>
<th>(b/t) Ratio</th>
<th>(a/2) (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Theory</td>
</tr>
<tr>
<td>1a</td>
<td>16.0</td>
<td>0.6</td>
<td>26.67</td>
<td>38.56</td>
</tr>
<tr>
<td>1b</td>
<td>28.5</td>
<td>0.6</td>
<td>47.50</td>
<td>38.08</td>
</tr>
<tr>
<td>1c</td>
<td>38.5</td>
<td>0.6</td>
<td>64.17</td>
<td>37.31</td>
</tr>
<tr>
<td>1d</td>
<td>53.5</td>
<td>0.6</td>
<td>89.17</td>
<td>36.14</td>
</tr>
<tr>
<td>1e</td>
<td>78.5</td>
<td>0.6</td>
<td>130.83</td>
<td>34.49</td>
</tr>
<tr>
<td>1f</td>
<td>128.5</td>
<td>0.6</td>
<td>214.17</td>
<td>32.19</td>
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</table>
Table 6.7 Comparison of Half-Wave Buckle Lengths from FEA and Theory for Lightly Profiled Panels with Rib Depth of 1.0 mm

<table>
<thead>
<tr>
<th>Series Number</th>
<th>Width of Flat Plate $b$ (mm)</th>
<th>Thickness $t$ (mm)</th>
<th>$b/t$ Ratio</th>
<th>$a/2$ (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Theory</td>
</tr>
<tr>
<td>2a</td>
<td>16.0</td>
<td>0.6</td>
<td>26.67</td>
<td>47.35</td>
</tr>
<tr>
<td>2b</td>
<td>28.5</td>
<td>0.6</td>
<td>47.50</td>
<td>46.58</td>
</tr>
<tr>
<td>2c</td>
<td>38.5</td>
<td>0.6</td>
<td>64.17</td>
<td>45.47</td>
</tr>
<tr>
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<td>0.6</td>
<td>89.17</td>
<td>43.78</td>
</tr>
<tr>
<td>2e</td>
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<td>0.6</td>
<td>130.83</td>
<td>41.38</td>
</tr>
<tr>
<td>2f</td>
<td>128.5</td>
<td>0.6</td>
<td>214.17</td>
<td>37.98</td>
</tr>
</tbody>
</table>

Table 6.8 Comparison of Half-Wave Buckle Lengths from FEA and Theory for Lightly Profiled Panels with Rib Depth of 1.3 mm

<table>
<thead>
<tr>
<th>Series Number</th>
<th>Width of Flat Plate $b$ (mm)</th>
<th>Thickness $t$ (mm)</th>
<th>$b/t$ Ratio</th>
<th>$a/2$ (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Theory</td>
</tr>
<tr>
<td>3a</td>
<td>16.0</td>
<td>0.6</td>
<td>26.67</td>
<td>55.66</td>
</tr>
<tr>
<td>3b</td>
<td>28.5</td>
<td>0.6</td>
<td>47.50</td>
<td>54.65</td>
</tr>
<tr>
<td>3c</td>
<td>38.5</td>
<td>0.6</td>
<td>64.17</td>
<td>53.26</td>
</tr>
<tr>
<td>3d</td>
<td>53.5</td>
<td>0.6</td>
<td>89.17</td>
<td>51.15</td>
</tr>
<tr>
<td>3e</td>
<td>78.5</td>
<td>0.6</td>
<td>130.83</td>
<td>48.12</td>
</tr>
<tr>
<td>3f</td>
<td>128.5</td>
<td>0.6</td>
<td>214.17</td>
<td>43.79</td>
</tr>
</tbody>
</table>

As seen from the results, the theoretical predictions overestimate the half-wave buckle length for all the cases investigated in this study compared with the FEA results. For small rib depths $h_c$ and flat plate width $b$, differences between FEA results and theoretical predictions are comparatively low. However, with the increase in $h_c$ and $b$, the differences between theory and FEA results increase making the results incomparable. For instance, theoretical prediction of $a/2$ for the sandwich panel with $h_c = 0.7$ mm and $b = 16.0$ mm (Table 6.6) is 17% higher than the FEA results (Theory = 38.56 mm, FEA = 33 mm). With the same $h_c$ of 0.7 mm but with $b$
= 128.5 mm, theoretical prediction is almost 29% higher than the FEA results (Theory = 32.19 mm, FEA = 25 mm). This illustrates that the theoretical prediction always overestimates the value of half-wave buckle length and the differences increase with increasing flat plate width $b$. Considering another example, theoretical prediction of $a/2$ for sandwich panels with $b = 78.5$ mm and $h_c = 0.7$ mm (Table 6.6) is 33% higher than the FEA results (Theory = 34.49 mm, FEA = 26 mm). However, for the same width $b$ of 78.5 mm but with different rib depth $h_c = 1.3$ mm (Table 6.8), the theoretical prediction is 101% higher than the FEA results (Theory = 48.12 mm, FEA = 24 mm). This clearly indicates that the theoretical predictions overestimate the half-wave buckle length not only with increasing flat plate width $b$, but also with increasing rib depth $h_c$.

Figures 6.15 and 6.16 also show these results as $a/2$ versus $b/t$ ratio curves for rib depths of 0.7 and 1.3 mm, respectively. The flat plate width $b$ is used in the discussion as the thickness is the same (0.6 mm), however, the more appropriate $b/t$ ratio is used in the graphs. Because of the low rib depth, the differences between the theoretical predictions and FEA results with changing $b/t$ ratios are considerably low as shown in Figure 6.15. However, it can be clearly seen from Figure 6.16 that the differences are increasing rapidly with increasing $b/t$ ratios for the lightly profiled panels with higher rib depths.

![Graph showing half-wave buckle length versus $b/t$ ratio for sandwich panels](image-url)

**Figure 6.15 Half-Wave Buckle Length versus $b/t$ Ratio for Lightly Profiled Panels with a Rib Depth of 0.7 mm**
Since there will be a close relationship between the half-wave buckle length and buckling stresses, detailed discussions are made for the former parameter. Some of the results are also plotted in Figures 6.17 to 6.19 to show the effect of rib depth $h_c$ on $a/2$. From these figures, it can be seen that the $a/2$ value obtained from the theory increases with the increasing rib depth $h_c$ for panels with all flat plate widths. Similar trends can be observed in the case of FEA results for those panels with comparatively lower flat plate widths. As the flat plate width $b$ increases, FEA shows limited increase in $a/2$ values despite increasing rib depth $h_c$. For the largest width $b$, $a/2$ remains constant for any rib depth $h_c$ as shown in Figure 6.19. Hence FEA results indicated that for higher flat plate widths ($b$), the value of half-wave buckle length ($a$) is almost constant and is independent of rib depth $h_c$.

It can further be observed from Figures 6.17 to 6.19 that, for low rib depths, the difference between the theoretical prediction and FEA results is reasonably low. Results also indicate that if the rib depth is zero (i.e. flat panel), the theoretical prediction of half-wave buckle length ($a/2 = 24.62$ mm) is very close to the FEA results ($a/2 = 25$ mm). However, the differences between the FEA results and theoretical predictions increase rapidly with increasing rib depth $h_c$. All of these
comparisons show that the current theoretical equation is not adequate to determine the true value of half-wave buckle length of lightly profiled panels. This formula (Equation 6.2.4), which was originally derived for flat panels and later extended to lightly profiled panels with a simple modification, can not therefore be used in its present form.

Figure 6.17 Half-Wave Buckle Length versus Rib Depth $h_c$ for Lightly Profiled Panel with Flat Plate Width of 16 mm

Figure 6.18 Half-Wave Buckle Length versus Rib Depth $h_c$ for Lightly Profiled Panel with Flat Plate Width of 38.5 mm
6.7.2 Buckling Behaviour and Comparison of Interactive Buckling Stresses

The actual buckling behaviour of lightly profiled sandwich panels is discussed here through a series of buckling examples obtained from finite element analyses. In these examples, a sandwich panel with a constant rib depth of 1.3 mm was considered with increasing flat plate widths between the ribs. Figures 6.20 to 6.25 show the buckled shape of lightly profiled sandwich panels with a rib depth of 1.3 mm and flat plate widths of 16.0 mm, 28.5 mm, 38.5 mm, 53.5 mm, 78.5 mm and 128.5 mm, respectively. In Figure 6.20, it can be observed from the buckle shape that the lightly profiled sandwich panel fails due to wrinkling as the effect of flat plate buckling is minimal due to the small flat plate width (16.0 mm). When the width is increased to 28.5 mm, the panel still shows wrinkling failure as seen in Figure 6.21. However, when the flat plate width increases to 38.5 mm, the buckle shape in Figure 6.22 shows that the panel does not fail due to wrinkling alone, instead local buckling occurs in the flat plate between ribs. This flat plate buckling is more obvious when the width increases to 53.5 mm in Figure 6.23. In Figures 6.24 and 6.25, the buckle shapes show very clearly the occurrence of local buckling in the flat plates as the flat plate widths are very high (78.5 mm and 128.5 mm, respectively). All these observations indicate that local buckling occurs along with wrinkling in lightly profiled sandwich panels with increasing flat plate widths (spacing of the ribs).
Similarly, the rib depth also plays a very important role in determining the buckling behaviour of lightly profiled sandwich panels. In Figure 6.26, the buckle shape of the lightly profiled panel with rib depth of 0.7 mm and flat plate width between the ribs of 53.5 mm is shown. This figure also shows the occurrence of both local buckling in the flat plate between the ribs and wrinkling in the panel. However, when compared with the buckle shape of the panel with a rib depth of 1.3 mm and flat plate width of 53.5 mm (Figure 6.23), the different scale of local buckling can be clearly seen. Although both panels have the same flat plate width, the panel with 1.3 mm rib depth has greater local buckling effects compared with the panel with 0.7 mm rib depth (see Figures 6.23 and 6.26). This demonstrates that the higher the rib depth, the greater will be the effects of flat plate local buckling even for the panels with the same flat plate width.

This systematic discussion through an example confirms that the interactive buckling behaviour is the dominant failure criterion of lightly profiled sandwich panels. Because of this complicated buckling phenomenon, the interactive buckling stress of the panel is always less than the wrinkling stress. In the following discussion, the interactive buckling stresses are compared with the wrinkling stresses from theory and CIB (2000). In the discussion, wrinkling stress from both theory and CIB (2000) are also described as interactive buckling stress as they are the predictions of failure stress of lightly profiled sandwich panels.

![Figure 6.20 Buckling Shape of Lightly Profiled Panel with Rib Depth of 1.3 mm and Flat Plate Width of 16 mm](image)
Figure 6.21 Buckling Shape of Lightly Profiled Panel with Rib Depth of 1.3 mm and Flat Plate Width of 28.5 mm

Figure 6.22 Buckling Shape of Lightly Profiled Panel with Rib Depth of 1.3 mm and Flat Plate Width of 38.5 mm
Figure 6.23 Buckling Shape of Lightly Profiled Panel with Rib Depth of 1.3 mm and Flat Plate Width of 53.5 mm

Figure 6.24 Buckling Shape of Lightly Profiled Panel with Rib Depth of 1.3 mm and Flat Plate Width of 78.5 mm
Figure 6.25 Buckling Shape of Lightly Profiled Panel with Rib Depth of 1.3 mm and Flat Plate Width of 128.5 mm

Figure 6.26 Buckling Shape of Lightly Profiled Panel with Rib Depth of 0.7 mm and Flat Plate Width of 53.5 mm
Interactive buckling stress results obtained from the finite element analyses of lightly profiled panels with rib depths $h_c$ of 0.7 mm, 1.0 mm and 1.3 mm with increasing $b/t$ ratios of flat plates between ribs are presented in Tables 6.9, 6.10 and 6.11, respectively, along with those predicted by the theory and the current design rule (CIB, 2000). Equation 6.2.3 based on the elastic half-space method was used for the theoretical predictions while Equation 6.2.8 included in the current European standard (CIB, 2000) was used for the design predictions.

**Table 6.9 Comparison of Interactive Buckling Stress Results for Lightly Profiled Panels with Rib Depth of 0.7 mm**

<table>
<thead>
<tr>
<th>Series Number</th>
<th>Width of Flat Plate $b$ (mm)</th>
<th>Thickness $t$ (mm)</th>
<th>$b/t$ Ratio</th>
<th>Interactive Buckling Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Theory</td>
</tr>
<tr>
<td>1a</td>
<td>16.0</td>
<td>0.6</td>
<td>26.67</td>
<td>147.76</td>
</tr>
<tr>
<td>1b</td>
<td>28.5</td>
<td>0.6</td>
<td>47.50</td>
<td>146.03</td>
</tr>
<tr>
<td>1c</td>
<td>38.5</td>
<td>0.6</td>
<td>64.17</td>
<td>143.15</td>
</tr>
<tr>
<td>1d</td>
<td>53.5</td>
<td>0.6</td>
<td>89.17</td>
<td>138.73</td>
</tr>
<tr>
<td>1e</td>
<td>78.5</td>
<td>0.6</td>
<td>130.83</td>
<td>132.42</td>
</tr>
<tr>
<td>1f</td>
<td>128.5</td>
<td>0.6</td>
<td>214.17</td>
<td>123.64</td>
</tr>
</tbody>
</table>

**Table 6.10 Comparison of Interactive Buckling Stress Results for Lightly Profiled Panels with Rib Depth of 1.0 mm**

<table>
<thead>
<tr>
<th>Series Number</th>
<th>Width of Flat Plate $b$ (mm)</th>
<th>Thickness $t$ (mm)</th>
<th>$b/t$ Ratio</th>
<th>Interactive Buckling Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Theory</td>
</tr>
<tr>
<td>2a</td>
<td>16.0</td>
<td>0.6</td>
<td>26.67</td>
<td>180.89</td>
</tr>
<tr>
<td>2b</td>
<td>28.5</td>
<td>0.6</td>
<td>47.50</td>
<td>178.21</td>
</tr>
<tr>
<td>2c</td>
<td>38.5</td>
<td>0.6</td>
<td>64.17</td>
<td>174.10</td>
</tr>
<tr>
<td>2d</td>
<td>53.5</td>
<td>0.6</td>
<td>89.17</td>
<td>167.78</td>
</tr>
<tr>
<td>2e</td>
<td>78.5</td>
<td>0.6</td>
<td>130.83</td>
<td>158.71</td>
</tr>
<tr>
<td>2f</td>
<td>128.5</td>
<td>0.6</td>
<td>214.17</td>
<td>145.79</td>
</tr>
</tbody>
</table>
Table 6.11 Comparison of Interactive Buckling Stress Results for Lightly Profiled Panels with Rib Depth of 1.3 mm

<table>
<thead>
<tr>
<th>Series Number</th>
<th>Width of Flat Plate b (mm)</th>
<th>Thickness t (mm)</th>
<th>b/t Ratio</th>
<th>Interactive Buckling Stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Theory</td>
</tr>
<tr>
<td>3a</td>
<td>16.0</td>
<td>0.6</td>
<td>26.67</td>
<td>211.75</td>
</tr>
<tr>
<td>3b</td>
<td>28.5</td>
<td>0.6</td>
<td>47.50</td>
<td>208.45</td>
</tr>
<tr>
<td>3c</td>
<td>38.5</td>
<td>0.6</td>
<td>64.17</td>
<td>203.39</td>
</tr>
<tr>
<td>3d</td>
<td>53.5</td>
<td>0.6</td>
<td>89.17</td>
<td>195.59</td>
</tr>
<tr>
<td>3e</td>
<td>78.5</td>
<td>0.6</td>
<td>130.83</td>
<td>184.26</td>
</tr>
<tr>
<td>3f</td>
<td>128.5</td>
<td>0.6</td>
<td>214.17</td>
<td>167.89</td>
</tr>
</tbody>
</table>

As seen from the results, the interactive buckling stress obtained from both theory and FEA increase with increasing rib depth $h_c$. Since the bending stiffness of the lightly profiled faces increases with increasing rib depth, it improves the strength of the compressive face thus raising the interactive buckling capacity. As sandwich panels are often subjected to wind pressure loading, a small increase in their rib depth can sufficiently enhance the interactive buckling strength making the panel capable of resisting greater wind pressures.

It is important to note that FEA results for interactive buckling stresses were compared with those predicted by the theory and the CIB (2000) design rule in order to investigate the accuracy of the theoretical equation and the reliability of the current design method. From the results it can be seen that the theoretical interactive buckling stresses do not agree with the FEA results. Instead the theoretical predictions always overestimate the wrinkling stress of lightly profiled panels for all rib depths $h_c$ considered in this study. When $h_c$ is considerably low, the difference between theory and FEA results is minimal. For the panel with $h_c$ equal to 0.7 mm and $b$ equal to 78.5 mm, theoretical prediction is 20% higher than the FEA result (Theory = 132.42 MPa and FEA = 109.76 MPa) as shown in Table 6.9. However, with the increase in $h_c$, the differences between theory and FEA results increase rapidly. When the rib depth $h_c$ is increased from 0.7 mm to 1.3 mm for the same $b$ of 78.5 mm, the theoretical equation estimates are 46% higher than the FEA results (Theory = 184.26 MPa and FEA = 125.68 MPa) as shown in Table 6.11.
When compared with the FEA results the current design predictions (CIB, 2000) are always low. Tables 6.9 to 6.11 show the interactive buckling stress results obtained from the design equation (CIB 2000) to be significantly lower than the FEA results for all the rib depths considered in this study. This indicates that the design equation included in the current European Standards (CIB, 2000) always underestimates the true strength of lightly profiled sandwich panels and its use in design makes the sandwich panel very uneconomical.

Figures 6.27 to 6.29 also show the interactive buckling stress results as a function of rib depth $h_c$. Although theoretical predictions are consistently higher than the FEA results, they agree reasonably well when the rib depth $h_c$ is very small i.e. close to a flat face. The figures show that the difference between the theory and FEA results increase if the flat plate width $b$ between the ribs also increases along with rib depth $h_c$. For example, Figure 6.27, which is a graph for a panel with $b = 16.0$ mm (small width), shows that the differences between theory and FEA results are consistently increasing, but the increase in differences is considerably low. On the other hand, Figure 6.29, which is a graph for a panel with $b = 128.5$ mm (large width), theoretical and FEA results are increasing rapidly with increasing rib depth $h_c$.

![Figure 6.27 Interactive Buckling Stress versus Rib Depth for Lightly Profiled Panel with Flat Plate Width of 16.0 mm](image-url)
Figure 6.28 Interactive Buckling Stress versus Rib Depth for Lightly Profiled Panel with Flat Plate Width of 53.5 mm

Figure 6.29 Interactive Buckling Stress versus Rib Depth for Lightly Profiled Panel with Flat Plate Width of 128.5 mm
Further it can be observed from the FEA results that very little increase in interactive buckling stress occurs with an increase in rib depth if the width of the flat plate is extremely large as seen in Figure 6.29. In all the cases, the interactive buckling stress obtained from the current design rule (CIB, 2000) is very low and highly conservative compared with FEA results. In Figures 6.30 to 6.32, the same results are plotted as interactive buckling stress versus $b/t$ ratio of a flat plate between ribs for lightly profiled sandwich panels with rib depths of 0.7 mm, 1.0 mm and 1.3 mm, respectively. As seen from the figures, the disagreement in wrinkling stresses obtained from theory and FEA increases consistently with increasing $b/t$. This disagreement is comparatively low for the panel with low rib depths as seen in Figure 6.30. However, the disagreement between FEA and theoretical predictions is very high for the panels with high rib depth as seen in Figure 6.32. Further, very little change in design prediction can be seen with an increasing $b/t$ ratio of the flat plate between ribs in contrast to FEA results as shown in all three figures.

![Figure 6.30 Interactive Buckling Stress versus $b/t$ Ratio of Flat Plate between Ribs for Lightly Profiled Panel with Rib Depth of 0.7 mm](image-url)
Figure 6.31 Interactive Buckling Stress versus $b/t$ Ratio of Flat Plate between Ribs for Lightly Profiled Panel with Rib Depth of 1.0 mm

Figure 6.32 Interactive Buckling Stress versus $b/t$ Ratio of Flat Plate between Ribs for Lightly Profiled Panel with Rib Depth of 1.3 mm
All the comparisons and discussions above indicate that the theoretical equation developed based on the elastic half-space method is inadequate to determine the value of interactive buckling stress of lightly profiled sandwich panels. Further, the current design method included in “European Standard for Sandwich Panels (CIB 2000)” is highly conservative, as it underestimates the true strength of the panels making the sandwich panel structure uneconomical. If both the rib depth $h_c$ and the flat plate width $b$ are very low, the theoretical predictions tend to be closer to FEA results. As already stated, the current theoretical equation for flat faced sandwich panels is well developed and fairly accurate. This rule was extended to lightly profiled sandwich panels with a simple modification. Therefore the interactive buckling stress obtained from theoretical and FEA results for a panel with low $h_c$ and $b$ are close to each other, as this type of panel is very close to a flat panel. However, theoretical and FEA results do not agree for the panels with large $h_c$ and $b$, as the theoretical equation is inadequate for these panels.

The above discussions and comparisons confirm that as the depth or spacing of the ribs increase, the possibility of pure wrinkling will be eliminated and flat plate buckling will occur along with wrinkling. Because of this mixed mode type complicated buckling behaviour occurring in lightly profile sandwich panels, the theoretical formula derived using the energy method which takes into consideration the pure wrinkling of the panel can not predict the true strength of lightly profiled sandwich panels which fail due to the interaction of local buckling and wrinkling. Because of this interaction of two the buckling modes, the ultimate strength of the panel decreases further due to the additional local buckling effect. Reduction in wrinkling stress due to the local buckling effect has to be considered in developing a wrinkling formula for lightly profiled sandwich panels.

The current theoretical approach has not taken into account this important aspect of structural behaviour of lightly profiled sandwich panels in its formulation. Therefore this formula is not adequate to predict the actual interactive buckling stress of the panels. The current European recommendation for the design of sandwich panels (CIB, 2000) also recommends that the condition $b/t < 100$ needs to be satisfied for the wrinkling formula to be applicable for the lightly profiled faces. This limitation recommended by CIB (2000) suggests that flat plate buckling should be avoided by
using smaller spacing of ribs in order to use the current wrinkling formula for lightly profiled sandwich panels. Current ribbed profiles used in the manufacture of lightly profiled sandwich panels do not meet the requirement outlined by CIB (2000). Kech (1991) had developed an alternative formula to deal with the mixed mode buckling phenomenon of lightly profiled sandwich panels. To prove its reliability and accuracy, a more extensive comparison with test results or exact analysis is required (Davies et al., 1991). Moreover, his equation is very complicated to use in any practical design. As lightly profiled sandwich panels are increasingly used in many building structures, a new or improved and easy-to-use design equation has to be developed based on the finite element analysis results in order to develop the confidence among designers and manufacturers. The new equation should take into account all aspects including the interaction of the two buckling modes.

6.8 Example Calculations

Equation 6.2.3 based on the elastic half-space method was used to predict the theoretical interactive buckling stress of lightly profiled sandwich panel. Theoretical values of half-wave buckle length \( a \) were determined using Equation 6.2.4. Measured material properties of polystyrene foam as used in finite element analysis were used in all of these theoretical calculations. Theoretical values of interactive buckling stress and half-wave buckle length were then compared with finite element analysis results in order to investigate the reliability of theoretical equations. A typical example of calculating theoretical interactive buckling stress and half-wave buckle length along with the interactive buckling stress predicted from design equation (CIB 2000) are given next. This example is for the ribbed faced lightly profiled sandwich panels with rib depth of 1.3 mm, flat plate width of 53.5 mm, steel face thickness of 0.6 mm and steel grade G550 as shown in Figure 6.33.

\[
\begin{align*}
\text{Young’s modulus of foam core } &= E_c = 3.80 \text{ MPa} \\
\text{Shear modulus of foam core } &= G_c = 1.76 \text{ MPa} \\
\text{Young’s modulus of steel face } &= E_f = 235000 \text{ MPa} \\
\text{Length of inclined element } &= \sqrt{4.25^2 + 1.3^2} = 4.44 \text{ mm}
\end{align*}
\]

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Slope of inclined element = \( \theta = \tan^{-1}\left(\frac{1.3}{4.25}\right) = 17.01^\circ \)

Centroid = \( \frac{53.5 \times 1.3 + 4.44 \times 1.3/2 \times 2}{53.5 + 4.44 \times 2 + 13} = 1.00 \text{ mm} \) from bottom steel

Consider this length = 75 mm

\( b = 53.5 \)
\( 21.5 \)
\( 13 \)
\( 4.25 \)
\( 4.25 \)

Figure 6.33 Typical Profile For Example Calculation

Second moment of area of steel face about centroidal axis:

\[
I_f = \frac{53.5 \times 0.6^3}{12} + 53.5 \times 0.6 \times 0.3^2 + 2 \left[ \frac{4.44^3 \times 0.6}{12} \sin^2 17.01 + 4.44 \times 0.6 \times 0.35^2 \right] \\
+ \frac{13 \times 0.6^3}{12} + 13 \times 0.6 \times 1.00^2
\]

\( I_f = 3.852 + 1.402 + 8.034 = 13.29 \text{ mm}^4 \)

\[
B_f = \frac{E_f I_f}{b} = \frac{235000 \times 13.29}{75} = 41642 \text{ N-mm}
\]

\[
A_f = \frac{(53.5 + 4.44 \times 2 + 13.0) \times 0.6}{75} = 0.6030 \text{ mm}^2 / \text{mm}
\]

\[
R = \frac{2(1 - \nu_c)E_c}{(1 + \nu_c)(3 - 4\nu_c)} = \frac{2 \times (1 - 0.08) \times 3.80}{(1 + 0.08) \times (3 - 4 \times 0.08)} = 2.4157
\]

Theoretical buckling stress,

\[
\sigma_{wr} = \frac{1.89}{A_f} \left[ \frac{8(1 - \nu_c)^2 E_c G_f B_f}{(1 + \nu_c)(3 - 4\nu_c)^2} \right]^{1/3} = \frac{1.89}{0.6030} \left[ \frac{8(1 - 0.08)^2 \times 3.8 \times 1.76 \times 41642}{(1 + 0.08)(3 - 4 \times 0.08)^2} \right]^{1/3}
\]
Theoretical half-wave buckle length,

\[ a = \pi \left[ \frac{2B_f}{R} \right]^{\frac{1}{3}} = \pi \left[ \frac{2 \times 41642}{2.4157} \right]^{\frac{1}{3}} = 102.25 \text{ mm} \]

\[ \frac{a}{2} = 51.13 \text{ mm} \]

Buckling stress from the current design equation (CIB 2000)

\[ \sigma_{wr} = \frac{0.95}{A_f} \left[ E_c G_c B_f \right]^{\frac{1}{3}} = \frac{0.95}{0.6030} \left[ 3.8 \times 1.76 \times 41642 \right]^{\frac{1}{3}} = 102.88 \text{ MPa} \]

6.9 Modification of Design Rule for Interactive Buckling

To develop an adequate and acceptable interactive buckling stress formula, it is necessary to identify all the parameters that affect the interactive buckling capacity of the lightly profiled sandwich panels. The current theoretical approach (Equation 6.2.3) based on the elastic half-space method indicates that the interactive buckling capacity of lightly profiled sandwich panels mainly depends on the mechanical properties \((E_c, G_c)\) of foam core and flexural rigidity \((B_f)\) of the steel face. However, the detailed finite element analyses conducted in this chapter have clearly indicated that the rib depth and \(b/t\) ratio of flat plates between the ribs have a great influence on the interactive buckling capacity. As the rib depth and \(b/t\) ratio of the flat plate between the ribs increase, flat plate buckling between the ribs occurs along with overall wrinkling of the panels. The simultaneous occurrence of these two buckling modes, namely local buckling and wrinkling, causes further reduction to the wrinkling capacity of lightly profiled sandwich panels. For smaller depth and spacing of ribs, the interaction is minimal and the reduction in strength is comparatively low. However, for larger depth and spacing of ribs, the interaction between the two buckling modes is severe and the reduction in strength is high. Hence, the buckling
behaviour of lightly profiled sandwich panels not only depends on the mechanical properties of foam core and steel faces, but also on the structural parameters such as rib depth and $b/t$ ratio of flat plates between the ribs. Therefore the effect of rib depth, width and thickness of flat plate must be included in any interactive buckling equation in order to calculate the true interactive buckling capacity of lightly profiled sandwich panels. Such inclusion will address the failures caused by the interaction of the two buckling modes. It should be noted that the more appropriate $b/t$ ratio is used now to allow for the variation of plate thickness $t$.

In fact, the simultaneous occurrence of wrinkling and local buckling modes makes the structural behaviour rather complex and hence it is difficult to develop a theoretical formulation to describe this complicated phenomenon. In the absence of any theoretical formulation, it is important that the existing wrinkling formula is improved or modified based on the results obtained from the extensive series of finite element analysis as a semi-empirical approach.

To improve the current wrinkling formula, the parametric study reported in Section 6.7 was extended to develop a larger database. The half-wave buckle length model was used to determine the interactive buckling capacity of lightly profiled sandwich panels with any possible geometry. In this parametric study, the rib depths of 0.7, 1.0, 1.3 and 1.6 mm were investigated. The flat plate widths chosen were 16, 28.5, 38.5, 53.5, 78.5, 98.5 and 128.5 mm. G550 steel grade with thicknesses 0.42, 0.60, 0.95 mm were used. For every rib depth, all seven flat plate widths ($b$) were considered in the finite element analysis with three different thicknesses resulting in 21 different cases. Hence with four different rib depths (0.7, 1.0, 1.3 and 1.6 mm), a total of 84 different types of foam-supported lightly profiled steel plates were modelled and analysed in this parametric study. Steel plates with different thickness and spacing of ribs enabled the inclusion of a wider range of $b/t$ ratio of flat plates. For the foam and different thicknesses of steel, measured mechanical properties were used in the analysis.

As explained already, the interactive buckling capacity of the lightly profiled sandwich panels depends on many parameters including mechanical properties of foam cores and steel faces, structural parameters such as bending rigidity, depth and
spacing of the ribs, and thickness of steel. The following functional relationship can be deduced from the above mentioned variables.

\[ \phi(\sigma_{wr}, E_c, G_c, B_f, A_f, v_c, h_c, b, t) = 0 \quad (6.9.1) \]

It was mentioned earlier that the wrinkling formula based on the elastic half-space method was inadequate as it does not consider the strength reduction due to the interaction of the two buckling modes resulting from increasing depth and spacing of ribs (rib depth \( h_c \) and flat plate width \( b \)). This formula can not predict the true interactive buckling strength of lightly profiled sandwich panels in its present form and hence it must be improved or modified to include the effect of interaction. Therefore the structural parameters such as depth of ribs \( h_c \) and \( b/t \) ratio of flat plates must be included in the interactive buckling equation. Hence, the functional relation to determine the interactive buckling capacity can be expressed as below.

\[ \sigma_{wr} = \frac{1.89}{A_f} \left[ \frac{8(1-v_c)^2 E_c G_c B_f}{(1+v_c)(3-4v_c)^2} \right]^{1/3} f(h_c, b, t) \quad (6.9.2) \]

The first part of Equation 6.9.2 is the wrinkling formula derived using the elastic half-space method, and the second part is the function added to account for the effect of depth and spacing of ribs, and thickness of steel on the interactive buckling capacity of lightly profiled sandwich panels. The first part is the dimensional formula that gives the wrinkling stress in N/mm\(^2\). It is necessary that the added second part be expressed as dimensionless. Therefore Equation 6.9.2 can be expressed in the following form including the dimensionless terms \( h_c/t \) and \( b/t \).

\[ \sigma_{wr} = \frac{1.89}{A_f} \left[ \frac{8(1-v_c)^2 E_c G_c B_f}{(1+v_c)(3-4v_c)^2} \right]^{1/3} f \left( \frac{h_c}{t} \right) \quad (6.9.3) \]

Results from the finite element analysis have shown that the relationship of the above parameters is not linear. Therefore attempts were made to combine the above dimensionless quantities with nonlinear interaction as shown next.
The various coefficient such as \( \mu, \phi \) and \( \delta \) in the above equation are determined by considering all the parameters simultaneously. The “Solver” in Microsoft Excel, which is based on the method of least squares and linear programming, was used to obtain the best equation that fits the FEA results. After substituting the values of the coefficient obtained from the “Solver” into Equation 6.9.4, the final equation to determine the interactive buckling capacity of lightly profiled sandwich panels can be expressed in the following form.

\[
\sigma_{wr} = \frac{1.89}{A_f} \left[ \frac{8(1-\nu_c)^2 E_c G_f B_f}{(1+\nu_c)(3-4\nu_c)^2} \right]^{1/3} \left( \mu + \phi \frac{h_c b}{t^2} \right)^\delta
\]

As seen in Equation 6.9.5, all the possible parameters that are deemed important in determining the interactive buckling stress of lightly profiled sandwich panels are included in the formulation. As stated earlier, the interactive buckling capacity of lightly profiled sandwich panels depends on both mechanical properties and structural parameters. The proposed formula has therefore incorporated both these aspects affecting the interactive buckling capacity.

This interactive buckling formula can be applied to lightly profiled sandwich panels with any practical rib depths (< 1.6 mm) and flat plate widths between the ribs. Lightly profiled sandwich panels currently used in Australia have a rib depth generally less than 1.0 mm and flat plate widths between the ribs less than 100 mm. Hence Equation 6.9.5 can be used successfully for the Australian ribbed profiles. As the rib depth or flat plate width is very small, the second part of Equation 6.9.5 reduces to one, resulting in the pure wrinkling stress equation. Therefore this formula can also be applied to the ribbed profiles with very small rib depths and flat plate widths, whose failure behaviour is mainly dominated by wrinkling with minimal flat plate buckling between the ribs. It must be noted here that the ductility problem associated with thinner G550 steels does not affect the final interactive buckling.
stress formula (Equation 6.9.5) as it gives only the elastic buckling stresses. Since the interactive buckling always occurs in the elastic region well below the yield stress of steel and there is very little postbuckling strength, Equation 6.9.5 is adequate for design.

The interactive buckling stress predicted by this new buckling formula was compared with the results obtained from the finite element analysis. The mean value of the ratio of the interactive buckling stresses obtained from the FEA and predicted by the new improved formula was found to be 1.01 and the corresponding coefficient of variation (COV) was 0.07. The maximum error in the predicted stresses was found to be less than 10%. This excellent correlation with the FEA results confirms that the improved formula can predict reasonable values of interactive buckling stress of lightly profiled sandwich panels. Figures 6.34 to 6.36 show the graphs of interactive buckling capacity plotted against the \(\frac{b}{t}\) ratio of flat plates between the ribs (or rib spacing). It can be observed that the interactive buckling capacity predicted by the new formula is close to the FEA results. This comparison confirms that the new formula can predict the interactive buckling stress accurately and hence can be used in determining the capacity of lightly profiled sandwich panels with greater confidence.

![Graph of Interactive Buckling Stress vs. \(\frac{b}{t}\) Ratio](image)

**Figure 6.34 Comparison of Interactive Buckling Capacity of lightly Profiled Panels with 0.6 mm Thick G550 Steel Face \((h_c = 0.7 \text{ mm})\)**
Figure 6.35 Comparison of Interactive Buckling Capacity of Lightly Profiled Panels with 0.6 mm Thick G550 Steel Face ($h_c = 1.0$ mm)

Figure 6.36 Comparison of Interactive Buckling Capacity of Lightly Profiled Panels with 0.6 mm Thick G550 Steel Face ($h_c = 1.3$ mm)
6.10 Summary

For the lightly profiled sandwich panels, flexural wrinkling is an extremely important design criterion as the behaviour of these panels is governed mainly by flexural wrinkling and its interaction with local buckling. A well established analytical solution exists for the pure flexural wrinkling behaviour of flat panels, however, the analytical solution for the interactive buckling of lightly profiled sandwich panels has been less well developed. The current design method recommended by the European design standard (CIB 2000) for the interactive buckling stress of lightly profiled panels is based on simple modifications of the methods utilized for flat faces to take into account the flexural stiffness of the lightly profiled faces. This semi-empirical approach of design does not consider the possible interaction between flexural wrinkling and local buckling modes. When the depth or spacing of the ribs increases in lightly profiled panels, flat plate buckling between the ribs can occur leading to the failure of the entire panel due to the interaction between local buckling and wrinkling modes.

In this study, the buckling behaviour of lightly profiled sandwich panels with varying depth and spacing of ribs was investigated extensively using detailed experimental and finite element analysis. The finite element models were validated using the results obtained from experiments and available current theoretical wrinkling formulae. The results from both experiments and finite element analyses confirmed that the wrinkling formula for lightly profiled sandwich panels based on the elastic half-space method is inadequate in its present form. Based on the finite element analysis results, an improved interactive buckling formula was developed by including the appropriate structural parameters such as depth and spacing of ribs (rib depth $h_c$ and flat plate width $b$), and thickness of steel face ($t$) to take into account the interaction of local buckling and wrinkling modes. The new interactive buckling formula is recommended for use in determining the true value of interactive buckling stress for the safe and economical design of lightly profiled sandwich panels.
CHAPTER 7.0 CONCLUSIONS AND RECOMMENDATIONS

7.1 Conclusions

The use of sandwich panels, in civil engineering construction, is an efficient and economic way of material utilization. The buckling strength of thin steel plate elements in sandwich panels is significantly improved by the presence of a foam core. However, sandwich panels exhibit various types of buckling failure modes depending on the types of steel faces used. These failure modes include local buckling of plate elements of profiled faces, flexural wrinkling of flat and lightly profiled faces and flexural wrinkling of lightly profiled faces interacting with local buckling. In order to achieve a safe and economical use of sandwich panels, it is important that accurate design rules are available that are developed based on a thorough understanding of the behaviour of sandwich panels subject to these complex modes of buckling. In this research project, the local buckling behaviour of plate elements in profiled sandwich panels and the flexural wrinkling behaviour of lightly profiled sandwich panels were investigated using a series of experimental studies and finite element analyses and appropriate design rules were developed. This thesis has presented the details of this research project and the results.

In the first phase of this research project on the local buckling behaviour of profiled sandwich panels, a series of experiments and finite element analyses were conducted on polystyrene foam-supported cold-formed steel plate elements as used in profiled sandwich panels to investigate their local buckling and ultimate strength behaviour. The study included foam-supported steel plate elements with a wide range of b/t ratios from 50 to 500 so that all the practical profiled sandwich panels were included. Both low and high strength steel grades (G250 and G550) were included. A total of 50 compression tests were conducted using a specially constructed test rig in this phase of experimental investigation. In the finite element analyses, MSC/PATRAN was used as a pre-processor (model generation) and post-processor (results visualization) and the finite element code ABAQUS was used for the analysis. Two different finite element models, a half-length model to simulate the experimental sandwich panels, and a half-wave buckle length model to simulate the realistic
conditions in sandwich panels used in building structures, were used. These models were validated by comparing the buckling and ultimate stresses and the load-deflection curves with the corresponding results from the experiments and available theoretical predictions. Using the results obtained from the half-wave buckle length model, the current effective width design rule was reviewed and a new improved design rule was developed for the safe design of sandwich panels subjected to local buckling effects with the $h/t$ ratio of plate elements up to 600.

Full-scale experiments of six profiled sandwich panels were undertaken to examine the accuracy of the new improved design rule. Test panels were subjected to a uniformly distributed wind pressure loading until failure using a large vacuum chamber (air box). Two different types of full-scale profiled sandwich panels (Type A and Type B) were used in this study in order to examine the applicability of the new design rule to any types of panels. Experimental failure pressures agreed well with the predictions from the new improved design rules.

A well established analytical solution exists for the design of flat faced sandwich panels, however, the analytical solution for the wrinkling of lightly profiled sandwich panels has been less well developed. The current design method recommended by the European design standard for the flexural wrinkling stress of lightly profiled panels is based on simple modifications of the methods utilized for flat faces to take into account the flexural stiffness of the lightly profiled faces. This semi-empirical approach of design does not consider the possible interaction between the buckling modes, namely local buckling and flexural wrinkling. As the depth or spacing of the ribs/profiles increases, flat plate buckling between the ribs can occur leading to the failure of the entire panel due to the interaction between local buckling and wrinkling modes. Therefore the second phase of this research investigated this complicated buckling behaviour associated with lightly profiled sandwich panels using a series of experiments and finite element analyses of lightly profiled sandwich panels with varying profile/rib depth and spacing. Finite element models of lightly profiled sandwich panels were calibrated using corresponding experimental results and available theoretical predictions. The results showed the inadequacy of current flexural wrinkling formulae for lightly profiled sandwich panels based on the elastic half-space method. Therefore it was modified to take into account all the practical
limitations including the effects of interaction between the two buckling modes, flexural wrinkling and local buckling.

A summary of the most significant findings arising from this research project on the investigation of buckling behaviour of sandwich panels is given next.

### 7.1.1 Local Buckling Behaviour of Fully Profiled Sandwich Panels

- The compression test set-up simulating simply supported longitudinal edges of the foam-supported steel plate elements as described in Chapter 3 can be successfully used to study the local buckling behaviour of fully profiled sandwich panels. It is considered as a suitable simple test model to investigate the local buckling behaviour of profiled sandwich panels.

- The stiffening effect of the foam core reduces the half-wave buckle length \((a < b)\) and produces many half-wave buckles within the plate length thus increasing the buckling strength considerably. The behaviour of foam supported plate elements was similar to that of flat plate elements and included elastic buckling, postbuckling and collapse (unloading) phases with the main differences being considerably improved buckling and ultimate strengths.

- There are many mathematical formulae to determine the enhanced buckling coefficient \(K\) of the foam-supported steel plate elements. For plate elements with low \(b/t\) ratios \(<200\), the equations proposed by Davies and Hakmi (1990) and Mahendran and Jeevaharan (1999) can be used successfully to determine the \(K\) values. However, for slender plates (high \(b/t\) ratios), the buckling equation proposed by Mahendran and Jeevaharan (1999) is more suitable in comparison with that proposed by Davies and Hakmi (1990).

- The results from experiments and finite element analyses (FEA) confirmed that for low \(b/t\) ratios \(<100\), the current effective width design rule can be used with an acceptable level of accuracy. However, it overestimates the strength and is unsafe for high \(b/t\) ratios (slender plates).
• Based on the results from experimental investigations, an interim design formula was developed by using an empirical reduction factor of 0.1\(R\) in the enhanced buckling coefficient (\(K\)) equation developed by Davies and Hakmi (1990).

• The numerical study described in Chapter 4 confirmed that the structural behaviour of sandwich panels including the local buckling, postbuckling and ultimate strength and collapse phases observed in the experiments can be simulated well by the half length finite element models developed in this research. This reduces the dependency on time consuming and expensive experimental methods.

• A half-wave buckle length FEA model, which is very close to the theoretical model based on elastic half-space method, can be used to investigate the local buckling and ultimate strength behaviour of practical sandwich panels used in common building systems.

• Both experimental and finite element analysis results showed that the postbuckling strength of foam supported plate elements gradually decreased with increasing \(b/t\) ratios. Very slender plates with very high \(b/t\) ratios failed due to flexural wrinkling and the failure stress can be determined using the available wrinkling formula. However, when the \(b/t\) ratio of the plate element lies in the intermediate region between the Winter and wrinkling regions (100 to 600), neither the wrinkling formula nor the current effective width formula can be used. It was found that the wrinkling formula underestimated the strength while the effective width formula overestimated the strength. The \(b/t\) ratio of plate elements in practical fully profiled sandwich panels is usually in the intermediate region, and none of the above design strength formulae can be used for this region.

• An improved design formula has been developed for the foam supported steel plate elements as used in fully profiled sandwich panels. The results from experiments and the validated finite element model were used for this purpose. In
this way, the complex local buckling and ultimate strength behavioural effects and the composite action of two dissimilar materials, namely foam core and steel, were included. This new design rule can be used for profiled sandwich panels with practical $b/t$ ratios up to 600.

- Full-scale tests of profiled sandwich panels confirmed that the current effective width approach is unsafe to use in design whereas the new design rule can be used safely in design as it predicts accurately the true strength of fully profiled sandwich panels.

7.1.2 Local Buckling and Flexural Wrinkling Behaviour of Lightly Profiled Sandwich Panels

- Finite element analyses and theoretical predictions show that a small increase in the profile/rib depth of lightly profiled sandwich panels can significantly improve their flexural wrinkling capacity.

- Flat faced sandwich panels are always subject to pure flexural wrinkling failures and the theoretical wrinkling stress equation for flat faced sandwich panels developed based on the energy method predicts accurate results that are in good agreement with experimental and finite element analysis results. The theoretical wrinkling stress equation for flat faced sandwich panels was extended to lightly profiled sandwich panels by assuming that lightly profiled sandwich panels are also subjected to pure wrinkling failures. However, the FEA and experimental results show that the compression face of lightly profiled sandwich panel does not fail by pure wrinkling, but instead by interactive wrinkling and local buckling. Current theoretical and design equations are unable to predict this complex behaviour of lightly profiled sandwich panels.

- A series of finite element analyses of lightly profiled sandwich panels with varying depth and spacing of ribs indicated that as the depth or spacing of ribs increases, pure wrinkling mode is eliminated and flat plate buckling between the ribs occurs along with the wrinkling phenomenon. This mixed mode type
complicated structural behaviour arising from the interaction of these two buckling modes results in considerable reduction of wrinkling strength of these panels.

- This reduction in wrinkling stress due to the interaction of local buckling and wrinkling modes must be considered in developing suitable design wrinkling formula for lightly profiled sandwich panels. Currently used theoretical methods do not take into account this important aspect of mixed mode buckling behaviour of lightly profiled sandwich panels in its formulation. Therefore the available wrinkling formula is not adequate to predict the actual wrinkling stress and cannot be used in the wrinkling capacity evaluation of lightly profiled sandwich panels.

- The current European standard for the design of sandwich panels (CIB 2000) recommends that the condition $b/t < 100$ needs to be satisfied for the wrinkling formula to be applicable for the lightly profiled faces. This limitation recommended by CIB (2000) suggests that flat plate buckling should be avoided by using smaller spacing of ribs in order to use the current wrinkling formula for lightly profiled sandwich panels. Current ribbed profiles used in the manufacture of lightly profiled sandwich panels do not meet the requirements outlined by CIB (2000).

- Finite element analysis results show that the current theoretical equation cannot predict the true value of half-wave buckling length of lightly profiled sandwich panels as it always overestimates these values. For low depth and spacing of ribs, theoretical predictions are close to FEA results, but as the depth and spacing of ribs increase, theoretical predictions are so high that they can not be compared with FEA results.

- The results from FEA and experiments indicated that the wrinkling failure behaviour of lightly profiled sandwich panels not only depends on the mechanical properties of foam core and steel faces, but also on the structural
parameters such as profile/rib depth and $b/t$ ratio of flat plates between the profiles/ribs.

- Based on the finite element analysis results, an improved wrinkling formula has been developed by modifying the current wrinkling formula based on the energy method through the inclusion of relevant structural parameters such as the ratios of depth and spacing of ribs to thickness to take into account the interaction of the two buckling modes, local buckling and flexural wrinkling.

- Comparison of finite element analysis and experimental results indicates that the buckling behaviour of lightly profiled sandwich panels can be simulated well using appropriate finite element models as developed in this thesis. This confirms that finite element analysis is an excellent tool for use in the investigations of sandwich panel behaviour and can be successfully used to contribute towards the design of sandwich panels. This will reduce the reliance on time consuming and expensive testing.

### 7.2 Recommendations

In the design of sandwich panels subjected to local buckling effects, it is recommended that the current effective width approach be limited to sandwich panels with $b/t$ ratios less than 100. If it is to be used for sandwich panels with larger $b/t$ ratios greater than 100, the enhanced buckling coefficient $K$ of the sandwich panels must be determined by replacing $R$ in Davies and Hakmi’s (1990) formula with $0.1R$. Alternatively, the new improved effective width design rule developed in this thesis is recommended for the design of profiled sandwich panels subjected to local buckling effects. This design rule can be used for the profiled sandwich panels with any practical plate slenderness ratio ($b/t$) up to 600.

It is recommended that the half-wave buckle length model developed in this thesis be used in any future research projects on the buckling behaviour of profiled or lightly profiled sandwich panels.
If the $b/t$ ratio of plate elements in fully profiled sandwich panels is greater than 600, it is recommended that the flexural wrinkling formula for flat faced sandwich panels be used to determine the failure strength of the plate element in the panels although it is slightly conservative.

Currently available theoretical wrinkling formula for lightly profiled sandwich panels is unconservative while the current design method included in CIB (2000) is conservative. Therefore it is recommended that the modified wrinkling formula developed in this thesis is used to determine the wrinkling capacity of lightly profiled sandwich panels.

7.3 Future Work

Since the use of sandwich panels as a mainstream product in buildings is relatively new in Australia, further research should be undertaken to improve the understanding of the various behavioural aspects of sandwich panels under specific Australian conditions.

Although the sandwich panel offers a wide range of advantages, its application is sometimes limited due to its poor fire performance. Fire performance of sandwich panels especially those with polystyrene foam core should be investigated to improve the confidence of manufacturers and designers. Some non-combustible core materials such as mineral wool are also available, but the fire performance of panels made of such materials has not been investigated adequately. Research should be directed to confirm their suitability to replace combustible core materials.

In this research project, sandwich panels with cold-formed steel faces and polystyrene foam core were investigated. The formulation of new design rules was based on the results of these particular types of materials. The new design rules can be applied to any types of material as they are a function of the mechanical properties of face and core and relevant geometrical parameters such as width, depth and thickness of faces. However, it is necessary to conduct further research using other materials such as aluminium, hardboard, gypsum plasterboard, etc. as face materials.
and polyurethane, polyisocyanurate, mineral wool, phenolic resin etc. as core materials to confirm the applicability of new design rules in order to develop confidence among sandwich panel manufacturers.

The wrinkling capacity formula developed in this thesis for lightly profiled sandwich panels is semi-empirical in nature. This formula takes into account the interaction of flexural wrinkling and local buckling modes that can occur in these panels. However, further research should be undertaken to develop a theoretical design equation.

It is believed that with further theoretical, numerical and experimental investigations, the new design equations developed in this thesis for profiled and lightly profiled sandwich panels can be further simplified and/or improved to make the sandwich panel designs more cost-efficient and safe.
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